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Progress in Ground-source Heat Pumps Using Natural Refrigerants

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Abstract

Natural refrigerants are attractive candidates for replacing the high Global-Warming- Potential fluorinated refrigerants used in ground-source heat pumps (GSHPs). This paper presents a comprehensive survey on GSHPs using CO₂, NH₃, water, and hydrocarbons. We compared the refrigerants' thermodynamic properties, analyzed their performance in brine-to-air and brine-to-water GSHPs, and discussed recent progress in their use in GSHPs. Studies of CO₂ were the most common due to its favorable properties, covering advanced cycles, direct-expansion, secondary fluid, and hybrid GSHPs. Though with toxicity concerns, NH₃ was the second most studied, including vapor-compression GSHPs for heating, absorption-type GSHPs to eliminate ground imbalance, and hybrid compression-absorption GSHPs to widen the operating temperature range. A few studies evaluated water as a refrigerant for absorption-type GSHPs, including applications for solar cooling, ground imbalance, and district heating. Propane was the only hydrocarbon considered for GSHPs, including analyses on refrigerant charge, performance analysis, and propane as a secondary fluid.

Keywords

Ground-source heat pump; Natural refrigerant; Carbon dioxide; Ammonia; Water; Hydrocarbon

1 Introduction

The building sector consumes 30 % to 40 % of the annual primary energy in developed countries, and about 15 % to 25 % in developing countries (Li et al., 2015). In the United States, the building sector accounts for about 40 % of primary energy consumption, with space heating, water heating, and space cooling respectively representing 37 %, 12 %, and 10 % of all the building energy consumption (EIA, 2017). Building energy use accounts for 40 % of the total U.S. CO₂ emissions and 7.4 % of the total global CO₂ emissions (EIA, 2017). Therefore, building energy efficiency plays a significant role in reducing energy demand and mitigating greenhouse gas emissions.

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There are many technologies being applied to improve building energy efficiency including enhanced thermal insulation (Papadopoulos, 2005), changes to occupant behavior (Hong et al., 2016), renewable energy integration (Yuan et al., 2013), and efficient heating, ventilating, and air-conditioning (HVAC) technologies (Huang et al., 2016). Ground-source heat pumps (GSHPs) are a particularly efficient HVAC technology that utilizes the ground as a heat sink/source with more favorable temperatures compared to outdoor air temperatures (Sarbu and Sebarchievici, 2014; You et al., 2015, Wei et al., 2017).

The typical working fluids for GSHPs, hydrochlorofluorocarbons (HCFCs, e.g., R22) and hydrofluorocarbons (HFCs, e.g., R134a, R410A), are scheduled to be phased out/down. HCFCs have both ozone depletion potential (ODP) and high Global Warming Potential (GWP), and may not be produced or imported after 2030 in developed nations and 2040 in developing nations. The Kigali Amendment to Montreal Protocol (UNEP, 2016) requires the participating parties to gradually reduce HFCs use by 80 % to 85 % by the late 2040s. Most developed countries are expected to begin reducing HFCs use in 2019. Most developing countries will follow with a freeze of HFCs consumption levels in 2024 (in 2028 for some developing countries). It is therefore necessary to look for long-term substitutes for HVAC equipment (Sarbu, 2014), and a group of natural refrigerants and low-GWP fluorinated refrigerants (i.e., hydrofluoroolefins, HFOs) are among the favorable candidates (McLinden et al., 2014; McLinden et al., 2017; Bamigbetan et al., 2017).

This paper provides a comprehensive literature survey on the progress in using natural refrigerants in GSHPs. First, we compared the thermodynamic characteristics of natural refrigerants, as well as their operation in vapor-compression brine-to-air and brine-to-water GSHPs. The remainder of the report discusses the status of efforts to apply each natural refrigerant, including CO₂ (i.e., carbon dioxide, R744), NH₃ (i.e., ammonia, R717), water (i.e., R718), and hydrocarbons, as applied to GSHPs in vapor-compression, absorption, and hybrid cycles. This literature survey provides a reference for future developments and applications of GSHPs using natural refrigerants.

2 Characteristics of natural refrigerants and their operation in basic vapor-compression GSHPs

Table 1 lists the basic properties of the studied natural refrigerants, all of which have zero ODP and very low GWP (compared to R22 with GWP 1760, R134a with GWP 1300, R410A with GWP 1924) over 100 years (McLinden et al., 2017). Water has a high normal boiling point of 100.0 °C, leading to sub-atmospheric operation and low compressor suction density for the temperature ranges of typical vapor-compression cycles used for space conditioning. Water also has a very high enthalpy of vaporization, but the low vapor density yields extremely low volumetric capacity. CO₂ has a low critical temperature of 31.0 °C, which causes transcritical operation and high pressure. NH₃ has favorable thermodynamic properties, including moderate pressure, high critical temperature, and high enthalpy of vaporization. However, NH₃ is toxic and requires additional controls to be used safely. Hydrocarbons propane and isobutane are moderate-pressure fluids with a high critical temperature, and enthalpy of vaporization between that of CO₂ and NH₃. The major

drawback of hydrocarbon refrigerants is their high flammability, with an “A3” safety classification (ASHRAE, 2016).

We compared these refrigerants under typical conditions for simulated vapor-compression systems applied to space cooling and heating. Each refrigerant was evaluated for a basic vapor-compression cycle consisting of a compressor, condenser, evaporator, and expansion device. Only thermodynamic properties were considered, so effects of differing transport properties were not captured. The heat exchangers were assumed to have sufficient size to achieve the target saturation temperatures. Additionally, heat exchanger and connecting tubing pressure drops were neglected. The rating test heat source/sink temperatures for brine-to-air and brine-to-water GSHPs (ISO 13256-1, 1998; ISO 13256-2, 1998) were applied, as shown in Table 2. We additionally specified the evaporating temperature, superheat, condensing temperature, and subcooling for both cooling and heating modes. This simplified analysis is useful for understanding the general characteristics of natural refrigerants operating in vapor-compression GSHPs.

To evaluate the compactness of a GSHP unit, we used the volumetric capacity, which is the cooling or heating capacity per refrigerant volumetric flow at the compressor inlet. In the cooling and heating modes, it was respectively calculated by:

$$q_{v, \text{cooling}} = \frac{Q_{\text{evap}}}{\dot{V}_r} = \frac{\dot{m}_r (h_{\text{evap}, \text{out}} - h_{\text{evap}, \text{in}})}{\dot{m}_r / \rho_{\text{com}, \text{in}}} = \rho_{\text{com}, \text{in}} (h_{\text{evap}, \text{out}} - h_{\text{evap}, \text{in}}) \quad (1)$$

$$q_{v, \text{heating}} = \frac{Q_{\text{cond}}}{\dot{V}_r} = \frac{\dot{m}_r (h_{\text{cond}, \text{in}} - h_{\text{cond}, \text{out}})}{\dot{m}_r / \rho_{\text{com}, \text{in}}} = \rho_{\text{com}, \text{in}} (h_{\text{cond}, \text{in}} - h_{\text{cond}, \text{out}}) \quad (2)$$

where $q_{v, \text{cooling}}$ and $q_{v, \text{heating}}$ are the cooling and heating volumetric capacities, $\text{kJ} \cdot \text{m}^{-3}$; Q_{evap} and Q_{cond} are the capacities of the evaporator and condenser, kW; $\dot{V}_r$ is the refrigerant volumetric flow rate at the compressor inlet, $\text{m}^3 \cdot \text{s}^{-1}$; $\dot{m}_r$ is the refrigerant mass flow rate, $\text{kg} \cdot \text{s}^{-1}$; $h_{\text{evap}, \text{in}}$ and $h_{\text{evap}, \text{out}}$ are the refrigerant specific enthalpies at the evaporator inlet and outlet, $\text{kJ} \cdot \text{kg}^{-1}$; $h_{\text{cond}, \text{in}}$ and $h_{\text{cond}, \text{out}}$ are the refrigerant specific enthalpies at the condenser inlet and outlet, $\text{kJ} \cdot \text{kg}^{-1}$; $\rho_{\text{com}, \text{in}}$ is the refrigerant density at the compressor inlet, $\text{kg} \cdot \text{m}^{-3}$.

The coefficients of performance (COPs) in the cooling and heating modes were respectively calculated by:

$$\text{COP}_{\text{cooling}} = \frac{Q_{\text{evap}}}{W_{\text{comp}}} = \frac{\dot{m}_r (h_{\text{evap}, \text{out}} - h_{\text{evap}, \text{in}})}{\dot{m}_r (h_{\text{comp}, \text{out}} - h_{\text{comp}, \text{in}})} = \frac{h_{\text{evap}, \text{out}} - h_{\text{evap}, \text{in}}}{(h_{\text{comp}, \text{out}} - h_{\text{comp}, \text{in}}) / \eta_i} \quad (3)$$

$$\text{COP}_{\text{heating}} = \frac{Q_{\text{cond}}}{W_{\text{comp}}} = \frac{\dot{m}_r(h_{\text{cond},\text{in}} - h_{\text{cond},\text{out}})}{\dot{m}_r(h_{\text{comp},\text{out}} - h_{\text{comp},\text{in}})} = \frac{h_{\text{cond},\text{in}} - h_{\text{cond},\text{out}}}{(h_{\text{comp},\text{out},i} - h_{\text{comp},\text{in}})/\eta_i} \quad (4)$$

where W_{comp} is the compressor power, kW; $h_{\text{comp},\text{in}}$ and $h_{\text{comp},\text{out}}$ are the refrigerant specific enthalpies at compressor inlet and outlet, $\text{kJ}\cdot\text{kg}^{-1}$; $h_{\text{comp},\text{out},i}$ is the refrigerant specific enthalpy at compressor outlet for isentropic compression, $\text{kJ}\cdot\text{kg}^{-1}$; η_i is the compressor isentropic efficiency, which was set to 0.70 for simplification (McLinden et al., 2017):

To evaluate the closeness of the actual COP to the Carnot COP, the COP ratio was calculated for cooling and heating modes:

$$\phi_{\text{cooling}} = \frac{\text{COP}_{\text{cooling}}}{\text{COP}_{\text{cooling, Carnot}}} \quad (5)$$

$$\phi_{\text{heating}} = \frac{\text{COP}_{\text{heating}}}{\text{COP}_{\text{heating, Carnot}}} \quad (6)$$

The COP of reversed Carnot cycle was defined in terms of the temperatures of heat source, T_{source} , °C, and heat sink T_{sink} , °C:

$$\text{COP}_{\text{cooling, Carnot}} = \frac{T_{\text{source}} + 273.15 \text{ K}}{T_{\text{sink}} - T_{\text{source}}} \quad (7)$$

$$\text{COP}_{\text{heating, Carnot}} = \frac{T_{\text{sink}} + 273.15 \text{ K}}{T_{\text{sink}} - T_{\text{source}}} \quad (8)$$

We used the Engineering Equation Solver (EES) (Klein, 2017) and its built-in refrigerant properties for the performance analyses. Table 3 and Table 4 present the characteristics of brine-to-air and brine-to-water GSHP using different natural refrigerants. Fig. 1 to Fig. 5 illustrate the pressure-enthalpy (p - h) diagrams for cooling and heating modes of the brine-to-water GSHP using the different natural refrigerants.

It is worth noting that the actual system efficiency and volumetric capacity will be less than the values shown in Tables 3 and 4. Pressure drop and, limited heat transfer area will reduce the system performance. Additionally, operation at higher temperature lift than values considered here (Table 2) will reduce performance. This is especially true for CO_2 in cooling mode; if the heat sink temperature is much higher than the standard 25 °C (Table 2) and the condensing temperature rises above the critical point (31.0 °C), the system transitions to a

less-efficient transcritical cycle. Nevertheless, the present analysis is useful for broad comparison of the refrigerants in terms of relative efficiencies, volumetric capacities, saturation pressures, and discharge temperatures.

Table 3, Table 4, and Fig. 1 to Fig. 5 show that CO₂ has the highest operation pressures, while water has the lowest and sub-atmospheric operation pressures. Additionally, water has very high discharge temperatures, especially in the heating mode, reaching 259.9 °C for the brine-to-air GSHP and 355.2 °C for the brine-to-water GSHP. NH₃ also has a high discharge temperature of 171.6 °C in the heating mode for the brine-to-water GSHP. In terms of the volumetric capacities, CO₂ is much better than other refrigerants, reaching 16119.0 kJ·m⁻³ in the cooling mode and 9277.0 kJ·m⁻³ in the heating mode for the brine-to-water GSHP. NH₃ has the second highest volumetric capacities, while water has extremely low volumetric capacities, which are 15.0 kJ·m⁻³ in the cooling mode and 12.8 kJ·m⁻³ in the heating mode for the brine-to-water GSHP.

Isobutane has the highest COP in most cases, while NH₃ has the highest COP in the heating mode for the brine-to-water GSHPs. The differences in COP between NH₃ and hydrocarbons are generally not significant. Water has a slightly lower COP in the cooling mode, and must operate with evaporating temperatures above 0 °C to avoid freezing, in both cooling and heating modes. CO₂ has much lower COPs than other refrigerants operating with the same evaporating and condensing (or gas cooler outlet) temperatures. Note that the values of COP/COP_{carnot} are N/A for cooling mode of the brine-to-air GSHP because the heat-source temperature (27 °C, air) is higher than the heat-sink temperature (25 °C, brine), which theoretically leads to an infinite COP_{carnot}. The values of COP/COP_{carnot} are much higher for NH₃ and hydrocarbons than for CO₂. Compared to the brine-to-air GSHPs, the values of COP/COP_{carnot} are much higher for the brine-to-water GSHPs due to the smaller temperature differences of the refrigerant-water heat exchangers (the thermodynamic losses associated with heat transfer across a finite temperature difference are smaller, so the cycle operation is closer to the ideal Carnot cycle).

Unlike the other refrigerants (Figs. 2 through 5) considered here, CO₂ operates near and partially above the critical point (i.e. transcritical cycle, Fig. 1). In the heating mode, the GSHP typically operates in a transcritical cycle. In contrast, the cooling mode operates in both a subcritical and transcritical cycle, depending on the ground loop temperature; this is an advantage over CO₂ air-source heat pumps (ASHPs) that typically provide cooling while operating in the less-efficient transcritical cycle. In this regard, CO₂ may be more promising when applied in GSHPs rather than in ASHPs. This is especially true in the regions with low ground temperatures that help the system operate more often in a subcritical cycle during the cooling season.

Each natural refrigerant has advantages and disadvantages for application with vapor-compression GSHPs. To generally compare the fluids:

- CO₂ has high operation pressure, low discharge temperature, high volumetric capacity, and low COP, and is non-flammable/non-toxic;

- NH_3 has moderate operation pressure, moderate discharge temperature, moderate volumetric capacity, and high COP, and is mildly flammable and toxic;
- water has extremely low operation pressure, high discharge temperature, extremely low volumetric capacity, and high COP, and is non-flammable/non-toxic;
- hydrocarbons have moderate operation pressure, low discharge temperature, moderate volumetric capacity, and high COP, and are highly flammable but non-toxic.

3 Progress in GSHPs using CO_2

The remainder of this paper discusses research that has been conducted for each natural refrigerant used in GSHPs, beginning in this section with CO_2 . We classified the progress of GSHPs using CO_2 into six applications: (1) the basic vapor-compression cycle; (2) advanced vapor-compression cycles; (3) direct-expansion systems; (4) systems with CO_2 as a secondary fluid; (5) multi-source hybrid CO_2 GSHPs; and (6) hybrid CO_2 GSHPs for lower ground thermal imbalance.

3.1 Basic vapor-compression cycle

The basic CO_2 GSHP includes a compressor, a condenser (or gas cooler), an expansion valve, and an evaporator. When operating as a transcritical cycle, the CO_2 GSHP is advantageous for water heating due to the ability to match the temperature profile of the supercritical refrigerant in the gas cooler with the temperature profile of the water (Kim et al., 2004). Different from subcritical cycles, the transcritical cycle has a high-side pressure independent of the temperature, and the pressure can be adjusted to optimize the COP (Pettersen, 1994; Hwang and Radermacher, 1998).

Jiang et al. (2009) studied a CO_2 GSHP water heating and air-conditioning system, and found that the COP of the GSHP was much higher than that of an ASHP. For evaporator temperatures of $-2\text{ }^\circ\text{C}$ (ASHP) and $10\text{ }^\circ\text{C}$ (GSHP), gas cooler outlet temperature of $22\text{ }^\circ\text{C}$, and water outlet temperature of $75\text{ }^\circ\text{C}$, the COP of the GSHP was 5.5, while that of the ASHP was 4.2. Therefore, CO_2 could be more promising to be used in GSHPs due to the favorable ground temperatures. Lin et al. (2011) investigated the experimental performance of a CO_2 GSHP in the heating mode. They adjusted the expansion valve opening to obtain different compressor discharge pressures. As the high-side pressure increased so did the heating capacity, compressor power, cooling water temperature, and gas cooler temperature. The heating COP reached the highest value of 3.4 when the high-side pressure was in the range of 9 MPa to 10 MPa.

It is suggested to adjust the operating pressure in transcritical CO_2 cycles for each operating condition to achieve optimal efficiency. The system controls can automatically set the expansion valve restriction, circulating refrigerant charge, and compressor speed, based on pre-determined optimal settings. The sizing of expansion valve, the charge of refrigerant, and the sizing of reservoir should be related to these controls, especially when the operating conditions vary greatly.

3.2 Advanced vapor-compression cycles

Advanced vapor-compression cycles can reduce the thermodynamic losses and improve the performance of the CO₂ GSHP (especially for transcritical operation). Many advanced CO₂ refrigeration and heat pump cycles have been proposed, which use one or more of: suction-line heat exchangers (SLHXs), ejectors, expanders, and multiple gas coolers (Groll and Kim, 2007; Ma et al., 2013).

Kim and Chang (2013) developed and validated a thermodynamic model for a CO₂ GSHP with a SLHX. They simulated the cooling and heating performance under various expansion valve openings, compressor speeds, and superheat, given a water temperature of 17 °C at the evaporator inlet and 25 °C at the gas cooler inlet in cooling mode (12 °C and 30 °C, respectively for heating). The added SLHX improved the COP by 2 % to 6 % in the cooling mode, while it slightly reduced the COP in the heating mode. Ma et al. (2003) compared conventional R22 and R134a GSHPs with two transcritical CO₂ GSHPs, where one CO₂ system used an expansion valve and the other used an expander. They applied an evaporating temperature of 0 °C in winter and 5 °C in summer, and varied the gas cooler outlet temperature. The CO₂ systems had comparable (to R22 and R134a) cooling and heating COPs when the gas cooler outlet temperature was higher than 55 °C, especially for the cycle with an expander. Although there are many advanced cycles for potential performance improvement, they are not always beneficial and no one could always outperform others, depending on the GSHP configuration, component performance, and operating modes and conditions. It's better to compare various GSHP cycles in terms of both energy and economic performance under the specific situation to make a choice.

In addition, a multi-gas-cooler configuration can also increase the performance of transcritical CO₂ GSHP by reducing the gas cooler outlet temperature. Jin et al. (2014) proposed an improved CO₂ GSHP system with an air-cooled gas cooler and a water-cooled gas cooler (Fig. 6(a)); this system increased the cooling performance by bringing down the gas cooler outlet temperature and thus reducing the expansion losses. Their simulation showed that by varying the fraction of the heat rejected by the air-cooled gas cooler, the cooling COP could be increased by 8 % to 20 % compared to a basic CO₂ GSHP. The system obtained a maximum average COP of 3.45 when the air-cooled gas cooler accounted for 30 % to 40 % of the total gas cooler capacity. The optimal operating pressure in the gas cooler was around 8 MPa for the investigated system. Morshed (2015) also studied the CO₂ GSHP system with an air-cooled and water-cooled gas cooler and an ejector (Fig. 6(b)). They computed the cooling COP and required borehole length under varied high-side pressures, assuming a cooling load of 10 kW, an evaporating temperature of 0 °C, a water-cooled gas cooler outlet temperature of 30 °C, and a borehole heat transfer rate of 40 W/m. Compared to the CO₂ GSHP without an ejector, the cycle increased the COP by 8 % on an average, with high-side pressure varying from 8.1 MPa to 11.5 MPa. Also, the author found that the operating conditions of the ejector cycle lowered the fraction of heat rejected in the air-cooled gas cooler, so longer boreholes were required.

Compared to an all-borehole (water-cooled gas cooler) GSHP, the multi-gas-cooler configuration can reduce the borehole cost by replacing some boreholes with refrigerant-to-

air heat exchangers. This is an effective approach for transcritical cooling where large temperature differences can drive much of the large temperature glide in an air-cooled heat exchanger. However, for GSHPs operating in a subcritical mode, a phase-change process with negligible temperature glide occurs in the condenser (instead of gas cooler), the refrigerant-to-air heat exchangers may cause a high condensing temperature due to the higher air temperature.

3.3 Direct-expansion systems

Direct-expansion GSHPs circulate the refrigerant, rather than a secondary fluid, in the ground heat exchanger acting as an evaporator (or condenser, gas cooler). Eliminating the secondary loop increases system efficiency, by removing the temperature lift required for a secondary fluid, and lowers the installation cost (Guo et al., 2012).

The heat transfer characteristics of boreholes are critical to the performance of the direct-expansion GSHPs. Eslami-Nejad et al. (2014) developed a numerical model to study the thermal behavior of a vertical geothermal borehole in a direct-expansion CO₂ GSHP in heating mode. The model accounted for combinations of single-phase and two-phase flow in the ground heat exchanger, and considered the borehole wall temperature variations as well as the thermal interaction between pipe sections. They assessed the pressure, temperature, and quality variations of the CO₂ along the U-pipe. Gao et al. (2017) established a borehole model to analyze the heat transfer between the refrigerant and the soil for a direct-expansion CO₂ GSHP. They studied the influence of shank spacing (center-to-center U-pipe distance), soil and backfill thermal conductivity, CO₂ inlet temperature, and CO₂ mass flow rate. They showed that for transcritical cooling, where the supercritical CO₂ has a large temperature change in the ground loop, thermal short circuiting exists between the shanks of the CO₂ gas cooler if the spacing is less than 120 mm. The refrigerant temperatures at the pipe outlet were approximately 2 °C and 9 °C higher than that at the pipe bottom when the spacing was 100 and 70 mm, respectively. The borehole field should be arranged to avoid the thermal short circuiting while occupying reasonable land areas. To reduce the borehole cost, the configuration of the borehole field should also be optimized to maximize the heat transfer under a given total borehole length.

Besides the borehole heat transfer, the system performance of direct-expansion GSHPs have also been studied experimentally and in simulation. Austin and Sumathy (2011) analyzed the performance of a direct-expansion CO₂ GSHP in a transcritical cycle for domestic hot water (DHW). Given a soil temperature of 6 °C and a cold-water inlet temperature of 7 °C, their simulation showed the baseline system had a COP of 2.18 and a heating capacity of 10.5 kW. For each evaporator (ground heat exchanger) length, there is an optimal gas cooler size that produces the highest COP; a longer ground heat exchanger should be paired with a larger gas cooler. Their optimization of ground heat exchanger length increased the COP to 2.58, an improvement of 18 % over the baseline. Eslami-Nejad et al. (2015) developed a quasi-transient model to simulate a direct-expansion CO₂ GSHP supplying space heating and DHW for a single-family house in a cold climate. The CO₂ GSHP offered efficient and stable performance, with an annual average COP of 2.8. They showed that increasing the total borehole length of a reference case by 25 % only decreased the annual energy

consumption by 6 %, while reducing the borehole length by 25 % increased the annual energy consumption by 10 %. Ghazizade-Ahsaei and Ameri (2017) analyzed the energy and exergy performance of a transcritical direct-expansion CO₂ GSHP for space-heating and DHW applications. They studied the system performance with variations in evaporator temperature difference, compressor speed, gas cooler inlet water temperature and mass flow rate, gas cooler length, number of ground loops, and soil temperature. The results can be used for design and optimization of direct-expansion CO₂ GSHPs. Eslami-Nejad et al. (2017) built an experimental test apparatus and developed a model to study direct-expansion transcritical CO₂ GSHP performance (Fig. 7). Their model predicted the experiment results within the measurement uncertainty. Parametric analysis indicated that improper control of gas cooler outlet temperature and compressor discharge pressure could degrade the COP up to 25 % and the heating capacity up to 7.5 %.

Without the temperature lift required for a secondary fluid used in conventional GSHPs, direct-expansion GSHPs operate in a subcritical cycle over a wider range of operating conditions. However, the existing studies mainly focused on the transcritical operation of the direct-expansion GSHPs, leading to the low efficiencies and losing the advantages over ASHPs. More efforts are suggested to apply direct-expansion GSHPs such that they operate in both subcritical and transcritical cycles during cooling season.

3.4 CO₂ as a secondary fluid in thermosyphons

Compared to conventional GSHPs using water or antifreeze as a secondary fluid, systems using CO₂ as a secondary fluid in the form of a closed two-phase thermosyphon have many advantages: pump-free operation enabled by the two-phase natural circulation inside the pipe, good heat-transfer performance during flow boiling, and no contamination of the ground in case of leakage (Mastrullo et al. 2014). Note that the system can be employed only in heating mode, constrained by the heat flow direction from the bottom to the top (Lim et al., 2017). Fig. 8(a) illustrates the principle of a closed two-phase thermosyphon connected to a GSHP. The CO₂ is condensed at the top of the thermosyphon due to the heat extraction by the heat pump. The condensate runs down on the inner walls and collects heat from the earth while evaporating. The density difference drives the vapor to rise in the center of the thermosyphon and is available for condensation at the top again. The CO₂ thermosyphon can be installed either as a single pipe or as a double pipe (Fig. 8(b)). In a double-pipe system, the liquid and vapor phases are separated, and the vaporous CO₂ ascends in the outer pipe, condenses and runs down in the inner pipe.

The heat transfer characteristics of CO₂ thermosyphons are critical to the performance of the GSHPs. Mastrullo et al. (2014) presented a numerical borehole heat exchanger model for a GSHP using CO₂ as the secondary fluid. They used the model to study the effect of CO₂ mass flow rate and inlet temperature on the subcooling length, outlet quality, pressure drop, and heat capacity of the U-pipe loop. Ebeling et al. (2017) investigated the thermodynamic behavior of CO₂ thermosyphons connected to a GSHP. They installed two geothermal thermosyphons with a length of 400 m and a design heat extraction of about 25 kW each. They used the measurements of heat and mass transfer within the pipes to validate the model. Similar to the direct-expansion GSHPs, geothermal thermosyphons should be

arranged and configured to avoid the thermal short circuiting and maximize the heat transfer while occupying reasonable land areas and using reasonable total borehole length.

Besides the geothermal thermosyphon heat transfer, the system performance of GSHPs have also been studied, mainly by measurement. Ochsner (2008) installed a 7.8 kW GSHP that provided in-floor heating for a single-family house. The GSHP was connected to a CO₂ thermosyphon that reached a depth of approximately 100 m, with a design heat extraction rate of 50 W/m. They measured the performance of the GSHP in the year 2006 to 2007 heating season, with the monthly COP in the range of 3.9 to 4.8 and the seasonal performance factor (SPF, defined as the ratio of the total delivered capacity to the total energy consumption over the heating or cooling season) of 4.1. Rieberer (2005) installed an experimental 9.5 kW GSHP equipped with two 65 m vertical boreholes, each with four CO₂ thermosyphons with a diameter of 15 mm. The heat extraction rate from the borehole was about 58 W/m. They also tested a 45 m horizontal CO₂ thermosyphon (i.e., collector) heated by the ambient air in the laboratory, which emulated thermal interaction of buried tubing with the ground. Heat extraction rate from the ambient to the evaporating CO₂ was in the range of 13 W/m to 17 W/m. Acuña et al. (2010) installed a CO₂ thermosyphon in a 70 m deep groundwater-filled borehole, instrumented with a fiber optic cable for distributed temperature measurements along the depth. The heat exchanger consisted of a straight insulated downcomer (i.e. a pipe that carried the CO₂ to the bottom of the borehole) and a helical riser designed to locate the circulating CO₂ as close as possible to the rock wall. The average temperature change of the riser was about 0.01 °C per meter of borehole. The average heat extraction rate was 71 W/m, significantly higher than for conventional ground loops (typically about 30 W/m).

The CO₂ geothermal thermosyphons feature high heat extraction rate from the ground and contribute to high efficiency of the GSHPs. A major shortcoming is that the system can only operate in heating mode, constrained by the heat flow direction from the bottom to the top. To make use of the heat pump unit in summer, additional cooling devices (e.g., liquid-to-air heat exchangers, cooling towers, etc.) are required; otherwise, another borehole field with secondary fluid should be configured for cooling mode.

3.5 Multi-source hybrid CO₂ GSHPs

In heating-dominated CO₂ GSHP applications, the heat source temperature can decrease year by year after continuous heat extraction, causing deteriorated GSHP performance. Several studies in the literature mitigated the heat source temperature change with additional heat sources like ambient air and solar energy.

Chargui et al. (2012) studied a hybrid CO₂ GSHP with dual sources and multiple modes for a greenhouse. The heat pump had two evaporators: a refrigerant-to-liquid heat exchanger to utilize heat from the surface water, and a refrigerant-to-air heat exchanger for use when the surface-water temperature was low (Fig. 9(a)). The system also allowed a direct heating mode for summer when the surface water could be used directly. Numerical results showed that the average heating COP was about 3.8. Alternating use of the additional ambient heat source can help recover the ground or water temperature. Moreover, the multi-source hybrid

GSHP can also reduce the cost by replacing some boreholes with refrigerant-to-air or liquid-to-air heat exchangers.

Kim et al. (2013) simulated a hybrid solar-assisted CO₂ GSHP for residential hydronic heating (Fig. 9(b)). They parametrically varied the undisturbed ground temperature from 11 °C to 19 °C, and the heating COP correspondingly changed from 2.13 to 2.81. The increasing compressor suction pressure led to the reduction of pressure ratio, increased compressor efficiency, and thus decreased compressor power. In addition, the fraction of the system heating load provided by the heat pump increased from 88.4 % to 93.5 %, with a maximum heating COP of 2.81. Choi et al. (2014) compared this solar-assisted CO₂ GSHP system to a conventional R22 GSHP system under different operating temperatures, solar radiation, indoor design temperatures, and borehole outlet temperatures. Under basic conditions (hot water temperature of 44 °C, indoor design temperature of 22 °C, daily solar radiation of 10 MJ/m², and evaporator inlet temperature of 15 °C), the heating COPs of the R22 and CO₂ GSHPs were 3.15 and 2.24, respectively. Further, the heating capacity of the R22 GSHP was 10 % higher than that of the CO₂ GSHP.

Similarly, the additional solar source can help maintain a favorable ground or water temperature by extracting less heat from the soil. The boreholes can be sized on cooling load, while the deficient heating capacity is complemented by the solar collector. System using CO₂ currently have much lower COPs than the conventional refrigerants applied in GSHPs, but it can be improved by adopting advanced CO₂ cycles. Combining GSHPs with DHW is also beneficial, since the temperature glide of CO₂ enables efficient heat exchange in water heaters.

3.6 Hybrid CO₂ GSHPs for lower ground thermal imbalance

The difference between the annual heat rejection and extraction by the ground, i.e., the ground thermal imbalance, must be considered in the design of CO₂ GSHPs to maintain long-term high efficiency and sufficient capacity. The ground thermal imbalance is especially important when the loads are heating or cooling dominated (You et al., 2016). Mitigating ground thermal imbalance with oversized ground heat exchangers is often cost prohibitive. Hybrid systems, which provide supplementary capacity during the dominant season, are an attractive and often more cost-effective alternative. The discussion here focuses on CO₂ GSHPs with the multiple gas coolers (for cooling-dominated applications) and solar-assisted systems (for heating-dominated applications).

For cooling-dominated applications, the heat rejection into the ground in summer is much higher than the heat extraction from the ground in winter, leading to ground temperature increase year after year. Hu et al. (2016) suggested using the CO₂ GSHP system with an auxiliary gas cooler to reject a portion of heat to the ambient air. They simulated an 8000 m² archives building and showed that the soil thermal imbalance was effectively eliminated by optimizing the fraction of heat rejected by the air-cooled gas cooler. Jin et al. (2017) analyzed a similar CO₂ GSHP with a SLHX in warm climates, using the ambient air and ground boreholes as heat sinks in the cooling mode, with only the ground boreholes as a heat source in the heating mode. The steady-state analysis suggested that the optimal control strategy of gas cooler pressure was important for decreasing the annual thermal imbalance.

The hourly simulation showed that the total cooling COP varied from 2.2 to 4.1, and the total heating COP varied from 2.53 to 3.15, while the ground had a low thermal imbalance.

Integrating DHW can significantly increase the system efficiency of CO₂ GSHPs. Jin et al. (2016) introduced a CO₂ GSHP system with heating, cooling, and DHW for warm climates. The system combined city water, the ground, and the ambient air as the heat sinks in the cooling mode, while only the ground was used as the heat source in the heating mode (Fig. 10(a)). During the cooling mode, the hybrid system recovered part of the rejected heat to produce DHW. The high-pressure CO₂ rejected heat in four sequential gas coolers; in order, the CO₂ was cooled by the DHW (at high temperature), the air, the ground, and the DHW (at low temperature). This heat exchanger configuration leveraged the large temperature change of the high-pressure CO₂ in the gas coolers (as opposed to most refrigerants, which reject heat in a nearly constant-temperature condensation process) to maximize the cycle efficiency and the exergy of the recovered heat. The simulation showed that the COP of the combined DHW and cooling mode varied from 3.0 to 5.5, and the annual ground thermal imbalance could be 0 % for the reference building.

In residential applications, the air-cooled gas cooler is suitable due to the simple configuration. In commercial applications, an evaporative gas cooler or a cooling tower may be used to increase the system efficiency.

For heating-dominated applications, the heat rejection into the ground in summer is much lower than the heat extraction from the ground in winter, leading to ground temperature decrease year after year. Ye et al. (2014) designed a solar-assisted CO₂ GSHP to maintain a low thermal imbalance for heating-dominated applications in several different cities (Fig. 10(b)). This hybrid system contained a solar collector, an air-cooled gas cooler, and a water-cooled gas cooler connected to the boreholes. They used the simulation to find the solar collector area that eliminated the thermal imbalance, for each city.

For a solar-assisted CO₂ GSHP, the solar energy can be used in a variety of ways: space heating, DHW, heat source of heat pump, heat storage underground. Energy and economic performance need to be compared to determine the most appropriate configuration for the specific application.

4 Progress in GSHPs using NH₃

NH₃ has long been used as a natural refrigerant with great thermodynamic performance, though the toxicity and mild flammability have been a barrier to broad implementation. Different from CO₂, NH₃ can be applied in both compression cycles and absorption cycles. We classified the progress of GSHPs using NH₃ into three cycle types: (1) vapor-compression cycles; (2) absorption cycles; and (3) hybrid compression-absorption cycles.

4.1 Vapor-compression cycle

A decade ago, Palm (2008) designed a prototype NH₃ water-to-water heating-only heat pump that was similar to typical GSHPs sold in Sweden. The tests showed that the relatively high discharge temperature allowed heating of DHW to temperatures well above the NH₃

condensing temperature. They also found that 100 g of NH_3 was sufficient for the 9 kW heat pump using mini-channel heat exchangers, and 120 g with plate heat exchangers. The measured heating COP was 3.8 to 4.8 for evaporating temperatures ranging from -13°C to 2°C and a condensing temperature of 48°C . Zajacs et al. (2017) designed an 8.4 kW NH_3 GSHP to provide space heating and DHW. They chose the main components and materials to avoid corrosion, and optimized the tank volume to minimize heating demand during the peak hours with high electricity price. Under an evaporating temperature of -3°C and a condensing temperature of 40°C , the calculated COP was 4.0.

Antonijevic and Komatina (2011) analyzed a two-stage cascade geothermal heat pump for heating. They compared several combinations of refrigerants, concluding that the combination with the highest overall COP was NH_3 in the low-temperature stage and R134a in the high-temperature stage. With geothermal source temperatures of 10°C to 30°C and a high-stage condensing temperature of 75°C , the heating COP ranged from 3.1 to 4.3.

Jensen et al. (2017) investigated the design of two heat pumps connected in series for a district heating network. The liquid sides of the evaporators for each heat pump were connected in series to enable large temperature change of the fluid used as a heat source; the condensers were similarly connected for the heat sink fluid. The GSHP system extracted geothermal heat at 73°C and supplied a heating capacity of 7.2 MW at a temperature of 85°C . They compared different refrigerants, including CO_2 , propane, NH_3 , isobutane, and R134a for vapor-compression heat pumps, and found that NH_3 was the best for many operating conditions (Ommen et al., 2015).

Although with safety concern, NH_3 is attractive because of its high efficiency. The high volumetric capacities of NH_3 (Table 3) also increase the heat pump compactness and reduce the refrigerant charge, which can lower the safety risk. Actually, the irritant odor of NH_3 is a natural detector of leakage, which is conducive to the safety control.

4.2 Absorption cycles

Conventional vapor-compression GSHPs used in cold regions suffer from ground thermal imbalance, which leads to decreased efficiency and reduced capacity after years of operation (You et al., 2016). Wu et al. (2013) proposed using an absorption GSHP to mitigate this problem; the system had a lower heating COP but a higher cooling COP, so it extracted less heat from the soil in winter and rejected more heat into the soil in summer. They conducted a multi-year simulation for the absorption GSHP system and found that it effectively reduced the year-round ground thermal imbalance in heating-dominant areas. Additionally, the primary energy efficiency of the absorption GSHP was competitive with the conventional electrically-driven GSHP. Experimental testing of a prototype confirmed the energy saving potential of the absorption GSHP using $\text{NH}_3\text{-H}_2\text{O}$ as the working fluid (Wu et al., 2016a). For heating-dominated buildings currently not equipped with cooling systems, or where the cooling is supplied by independent air conditioners (very common in China), the GSHP will have significant ground thermal imbalance. Wu et al. (2014a) proposed a hybrid system combining an absorption GSHP and borehole free-cooling, providing cooling with insignificant modification (only valves) and at very low operation cost (only the pump electricity). Simulations indicated that after 10 years, the heat rejected by borehole free-

cooling reduced the soil temperature drop from 7 °C to 3 °C, and the primary energy efficiency increased from 95 % to 111 % in an extremely cold city.

Some applications of absorption GSHPs exhibit long-term ground temperature increase; Wu et al. (2014b) mitigated the temperature increase using a “combined-appliance” absorption GSHP that recovered the heat rejection in summer to produce DHW. Case studies on the “combined” system applied in Beijing showed that the thermal imbalance ratio was only –15 % for the absorption GSHP, while the imbalance ratio was as large as –77 % for the conventional electrically-driven GSHP. The annual soil temperature change was only 0.5 °C for the absorption GSHP, while it could reach 3.0 °C for the electrical GSHP. Moreover, the heat rate recovered for DHW, during operation of the GSHP in the summer, was in the range of 30.0 kW to 36.8 kW. Overall, the annual primary energy efficiency increased from 105.0 %, for the absorption GSHP without heat recovery, to 151.3 %, for the absorption GSHP with recovery.

To reduce the required number of boreholes while maintaining a low ground thermal imbalance in cooling-dominated climates, Wu et al. (2015a) suggested using a hybrid absorption GSHP integrated with a cooling tower. They conducted hourly simulations for a single-effect cycle and a generator-absorber heat exchange (GAX) cycle. Compared to the conventional electrical GSHP, the required number of boreholes and occupied land area were reduced by 37 % to 52 % using the single-effect cycle, and reduced by 20 % to 38 % using the GAX cycle.

The absorption-type GSHPs are potential alternatives in heating-dominant regions in terms of primary energy efficiency, ground thermal imbalance, and required boreholes. The cooling efficiency is relatively low, but the system can directly use a heating source (fossil fuels, renewable energy, waste heat, etc.) thereby reducing electrical power demand during peak times in summer.

4.3 Hybrid compression-absorption cycles

Hybrid NH₃ cycles herein refer to systems that combine elements of vapor-compression and absorption cycles (rather than using an alternative heat source/sink, as is the case with the hybrid CO₂ systems discussed in Section 3). The compression-absorption cycle (Fig. 11(a)) replaces the evaporator and the condenser in a vapor-compressor cycle with a generator and an absorber. Since the working fluids in the heat exchangers are mixtures, the temperature glides facilitate applications with wide temperature variations (Jensen et al., 2015). Additionally, this hybrid cycle can produce higher-temperature hot water efficiently since the pressures are reasonably low; this stands in contrast to conventional subcritical vapor-compression cycles requiring very high pressures to produce hot water (Gudjonsdottir et al., 2017). Kim et al. (2010) suggested using a hybrid NH₃-H₂O compression-absorption cycle to utilize geothermal heat at moderate temperatures of 30 °C to 50 °C to produce hot water above 90 °C. Simulation results showed that the hybrid system had low irreversibility during heat-exchange processes and subsequently had 10 % to 20 % higher COP than the R134a vapor-compression cycle. The authors built a lab-scale prototype, which they used to verify the simulation and to refine techniques for gas separation and the chemical mixing process. Jensen et al. (2017) also suggested using the hybrid compression-absorption heat pump to

replace the vapor-compression heat pump in a serially-connected GSHP system (the heat pumps sequentially extracted/added heat to the heat source/sink fluids, similar to the system discussed in Section 4.1 by the same authors) for a district heating network. They analyzed the exergetic efficiency, technical feasibility, and the economic viability for a range of configurations. This hybrid heat pump applies when the required temperature is high and the available heat source temperature is not low. It is typically suitable for industrial and district heating applications.

Another hybrid compression-absorption cycle is a compression-assisted absorption cycle (Fig. 11(b)), which can increase the absorption pressure and strengthen the vapor absorption process in the absorber (the solution can absorb more refrigerant under increased pressures) (Ventas et al., 2012). This improves the system efficiency and widens the effective operating temperature range. Experimental comparisons revealed that the hybrid compression-assisted technique could extend the lower limit of driving source temperature (generator input) from 125 °C to 110 °C, and increased the heating capacity by 96.4 % (Wu et al., 2016a). Another experimental comparison indicated that the hybrid heat pump extended the lower limit of evaporator inlet temperature from -10 °C to -25 °C, and increased the heating capacity by approximately 56 % to 85 % (Wu et al., 2016b). This hybrid heat pump typically requires NH₃ compressors (or pressure boosters) with relatively low compression ratios, since the absorption pressure is only moderately higher than the evaporation pressure.

Another type of hybrid compression-absorption GSHP combines two standalone GSHPs, one vapor-compression and one absorption, in a parallel configuration that shares the ground loop and water pipelines. Wu et al. (2015b) proposed this hybrid system to balance the positive and negative attributes of the vapor-compression GSHP (higher cooling primary energy efficiency, higher cooling capacity, and decreasing soil temperature) and the absorption GSHP (higher heating primary energy efficiency, higher heating capacity, and increasing soil temperature). This hybrid GSHP maintained low ground thermal imbalance with annual soil temperature variations smaller than 0.2 °C/yr. Compared to the conventional electrical GSHP, the hybrid GSHP saved 9.8 % to 25.7 % in primary energy for different cities. The design and control of these two GSHPs might be a challenge to maximize the annual efficiency. Combining two standalone existing GSHPs is easy to apply, but the cost can be reduced by developing a new cycle, in which the vapor-compression cycle and the absorption cycle share the same evaporator and condenser.

5 Progress in GSHPs using water

Compared to CO₂ and NH₃, water has been infrequently considered as a working fluid for GSHPs. The limited studies covered absorption systems applied to: (1) solar cooling; (2) ground thermal imbalance; and (3) district heating.

Absorption cycles are attractive options for solar cooling, but are rarely integrated with a ground loop. Macía et al. (2013) investigated an experimental H₂O-LiBr absorption GSHP driven thermally by solar energy and by a gas boiler (in times of low solar radiation). For the cooling operation in summer, the evaporator was connected to a radiant floor system and the condenser exchanged heat with the ground. The unit was a single-effect cycle with a

nominal cooling capacity of 35 kW and COP of 0.7. Results showed that if the condenser inlet temperature decreased from 28 °C to 25 °C, the heat pump COP increased by 30 %.

To alleviate the ground thermal imbalance in heating-dominated regions, Wu et al. (2013) proposed to apply an absorption GSHP, adopting both H₂O-based and NH₃-based working fluids. Since H₂O cannot be used under low evaporating temperatures due to the freezing point limitation, it's better to use H₂O-based fluids where the soil temperature is relatively high and NH₃-based fluids where the soil temperature is low.

To improve the efficiency of district heating systems, Li et al. (2011) proposed a novel H₂O-LiBr absorption heat pump to scavenge low-grade thermal energy from renewable sources, like subsurface underground water, geothermal fluid, municipal sewage, and surface water. The absorption system was driven by the high-temperature district heat network (130 °C in Fig. 12), extracted further heat from the low-grade source (72 °C in Fig. 12), and delivered the water at temperatures useful for hydronic heating coils (50 °C in Fig. 12). Compared to the conventional district heating system, the new heating system saved energy by 23 % to 46 %. Compared to the system with a vapor-compression GSHP (Fig. 12(b)), the new system used more low-grade thermal energy and less electricity. This system is suitable when the available geothermal temperature is relatively high. The solution temperature in the absorber is high in heating mode, which requires the working fluid to have a high mass fraction of LiBr for a strong vapor absorption process. With high LiBr mass fraction crystallization-prevention methods must be incorporated.

6 Progress in GSHPs using hydrocarbons

The existing research on GSHPs using hydrocarbons is quite limited, and mainly focuses on propane. Existing GSHP studies investigated: (1) refrigerant charge; (2) system performance; (3) comparison with ASHPs; and (4) hydrocarbons as a secondary fluid of GSHPs.

Fernando et al. (2004) built and tested a prototype brine-to-water heat pump with a heating capacity of 5 kW using propane as refrigerant. They minimized the refrigerant charge by using mini-channel aluminum heat exchangers, which featured high heat-transfer coefficients and could reduce the charge without lowering the COP. For the minimum borehole brine temperatures of -2.2 °C to 3.8 °C, a charge of about 200 g was the best choice for the heat pump and resulted in a COP of 3.5 to 4.0.

Corberán et al. (2011) examined how the control logic affected the overall performance of a propane GSHP with nominal heating and cooling capacities of 18 kW and 14 kW, respectively. They analyzed the set-point temperatures and temperature bandwidths of the building hydronic circuit and building space, and found that the set-point temperature was the dominant factor. As the cooling set-point temperature increased from 21 °C to 25 °C, the daily system power consumption decreased significantly from 67.5 kWh to 48.2 kWh. After this initial parametric testing, they ran the system to provide space conditioning for five years. During this time, the supply water temperature was controlled to 45 °C for heating and 10 °C for cooling, while the borehole outlet water temperature was around 17.4 °C for

heating and 26 °C for cooling. The daily heat pump COP on a typical heating day was in the range of 3.5 to 4.2, while that on a typical cooling day was in the range of 4.2 to 5.1 (Montagud et al., 2011). Recently, they presented reference data sets of the propane GSHP system based on an eleven-year operation period for validation and analysis (Ruiz-Calvo et al., 2016).

Urchueguía et al. (2008) showed an experimental comparison between a propane GSHP system and a conventional ASHP system. The test data indicated that the GSHP saved 43 % in energy over a whole heating season, and 37 % over a whole cooling season. The heating SPF was 3.5 for the GSHP and 2.0 for the ASHP, while the cooling SPF was 4.3 for the GSHP and 2.7 for the ASHP. Tamaro et al. (2016) compared a GSHP with an ASHP using propane through a transient building simulation. They considered office and residential buildings in mild, warm, and cold climates of Europe. The heating SPF of both systems was above 3.75, with an advantage of around 5 % for the GSHP. In addition, the percentage of hours with thermal discomfort remained below 10 % for all the cases.

Similar to CO₂, propane has also been investigated as a secondary fluid of GSHP systems using a two-phase thermosyphon for pump-free operation. Grab et al. (2011) reported the measured results of a GSHP system integrating seven propane geothermal thermosyphons, with a borehole spacing of 6.0 m and depths between 85.0 m and 95.5 m. They used fiber optic cables to measure the temperature along the whole thermosyphon. They also measured the heat extraction through the thermosyphons and the power consumption of the system. Hantsch et al. (2013) and Hartmann et al. (2015) used a numerical model to investigate how the heat transfer of geothermal thermosyphons varied with pipe material, wetting ratio, thermal conductivities of soil and grout, and borehole diameter. They found that the pipe and grout material became increasingly important for lower wetting ratios and shorter extraction times; wetting ratios of at least 80 % were recommended.

Section 2 showed that hydrocarbons have excellent thermodynamic properties and cycle performance. Their main drawback is their flammability, where a reduction in charge can improve safety. GSHP systems are good candidates for applying hydrocarbons, since the charge can be small (there is no outdoor refrigerant-to-air heat exchanger or long line sets) and the refrigerant circuit is factory-sealed.

7 Summary and conclusions

Concerns about global warming and the need to comply with related regulation have invigorated interest in using natural refrigerants in GSHPs. We first compared the basic thermodynamic properties of different natural refrigerants, including CO₂, NH₃, water and hydrocarbons, and their performance in a basic vapor-compression cycle incorporated in brine-to-air and brine-to-water GSHPs. Each natural refrigerant has advantages and disadvantages: CO₂ has high operation pressure, low discharge temperature, high volumetric capacity, low COP, and is non-flammable/non-toxic; NH₃ has moderate operation pressure, moderate discharge temperature, moderate volumetric capacity, high COP, mild flammability, and is toxic; water has extremely low operation pressure, high discharge temperature, extremely low volumetric capacity, high COP, and is non-flammable/non-toxic;

the hydrocarbons examined have moderate operation pressure, low discharge temperature, moderate volumetric capacity, high COP, high flammability, and are non-toxic.

Table 5 summarizes the studies discussed here regarding GSHPs using natural refrigerants. CO₂ was the most popular natural refrigerant for the experimental GSHP systems. The existing publications covered the studies on the basic vapor-compression cycle, advanced vapor-compression cycles, direct-expansion systems, CO₂ as a secondary fluid of GSHPs, multi-source hybrid CO₂ GSHPs, and hybrid CO₂ GSHPs for lower ground thermal imbalance.

Based on the simplified thermodynamic analysis in Section 2, the low COPs for CO₂, compared to NH₃ and hydrocarbons, make it typically an inferior refrigerant. However, this analysis used fixed values for evaporator and condenser saturation temperatures. CO₂ would show more competitive results when evaluated using a more detailed analysis that considers the effects of pressure drop and refrigerant-side heat transfer coefficients, and computes (rather than specifies) the saturation temperatures; this improved performance would be especially true if the analysis includes optimized heat exchanger designs that can exploit the favorable thermodynamic and transport properties of CO₂ (Brignoli et al. 2017). These optimized systems can operate with higher evaporator saturation temperature (and pressure) and lower condenser saturation temperature (and pressure), resulting in higher compressor efficiency and system efficiency. Further, by using direct-expansion ground loops or high-efficiency heat exchangers in the secondary-loop, CO₂ GSHPs can operate as a subcritical cycle over a wider ground temperature range, and thus yield more competitive efficiencies. Finally, CO₂ is non-flammable and non-toxic, and systems using the fluid can be made compact because of the high volumetric capacity. Considering these combined potentials and merits there has been much interest in exploring the use of CO₂ for GSHP applications.

NH₃ is the second most studied natural refrigerant for GSHP systems. However, the fluid is more attractive in absorption cycles, which are good options to utilize renewable energy, waste heat, and fossil fuel. For vapor-compression cycles, the existing NH₃ GSHP studies mainly focused on heating applications where these cycles show the best efficiencies. The absorption-type NH₃ GSHPs were competitive with the conventional vapor-compression GSHPs, in terms of the ground thermal imbalance, primary energy efficiency, and number of boreholes, especially in cold regions. Additionally, hybrid compression-absorption NH₃ GSHPs have been proposed for higher-temperature hot water production, wider operation temperature ranges, and higher efficiencies. The toxicity concern is the main barrier for wider applications of NH₃, so advanced technologies for charge reduction, leakage reduction, and safety control are required.

Water has attracted much less attention for use in GSHPs; it is constrained by its extremely low operation pressure, high discharge temperature, and extremely low volumetric capacity, as well as high freezing point. The limited studies all focused on absorption-type GSHPs, covering solar cooling, ground thermal balance, and district heating.

Hydrocarbons have also been rarely investigated for use with GSHPs, because the flammability risk restricts widespread adoption. Propane was studied with a focus on

minimizing refrigerant charge, GSHP system performance, comparison with the ASHP, and propane as a secondary fluid of GSHPs. GSHP systems are good candidates for applying hydrocarbons, since the charge can be small (there is no outdoor refrigerant-to-air heat exchanger or long line sets) and the refrigerant circuit is factory-sealed.

To date, natural refrigerants have been explored less for GSHPs than for ASHPs. However, a new paradigm is emerging with restrictions on HFC and HCFC fluids, as well as continued need to reduce total energy consumption and related CO₂ emissions. As a result, GSHPs using natural refrigerants might experience a rapid expansion in research and application.

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Nomenclature

h	specific enthalpy, kJ·kg ⁻¹
$\dot{m}$	mass flow rate, kg·s ⁻¹
p	pressure, kPa
Q	thermal capacity, kW
q_v	volumetric capacity, kJ·m ⁻³
T	temperature, °C
$\dot{V}$	volumetric flow rate, m ³ ·s ⁻¹
W	power, kW
ρ	density, kg·m ⁻³
η_i	isentropic efficiency
ϕ	ratio of real COP to Carnot COP
H_{VAP}	vap enthalpy of vaporization, kJ·kg ⁻¹

Abbreviations

ASHP	air-source heat pump
CFC	chlorofluorocarbon
COP	coefficient of performance
DHW	domestic hot water
EES	Engineering Equation Solver software

GAX	generator-absorber heat exchange
GSHP	ground-source heat pump
GWP	global warming potential over 100 years
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
HVAC	heating, ventilating, and air conditioning
ODP	ozone depletion potential
SLHX	suction line heat exchanger
SPF	seasonal performance factor
TRNSYS	Transient System Simulation Tool

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Fig. 1.
P-h diagram of the brine-to-water GSHP using CO₂

(c) cooling mode

(d) heating mode

Fig. 2.
P-h diagram of the brine-to-water GSHP using NH_3

(a) cooling mode

(b) heating mode

Fig. 3.
P-h diagram of the brine-to-water GSHP using water

Fig. 4.
P-h diagram of the brine-to-water GSHP using propane

Fig. 5.
P-h diagram of the brine-to-water GSHP using isobutane

Fig. 6.
CO₂ GSHP with an air-cooled and a water-cooled gas cooler

(a) schematic of the test apparatus

(b) photograph of the test apparatus

Fig. 7.

Experimental test apparatus for the direct-expansion transcritical CO₂ GSHP (Eslami-Nejad et al., 2017)

(a) thermosyphon principle (Ebeling et al., 2017)

(b) single and double pipe (Ochsner, 2008)

Fig. 8.
Geothermal thermosyphon for GSHPs

(a) integration with ambient air (Chargui et al., 2012)

(b) integration with solar thermal energy (Kim et al., 2013)

Fig. 9.
Hybrid CO₂ GSHP with multiple heat sources

(a) multiple gas coolers for cooling-dominated climates (Jin et al., 2016)

(b) solar-assisted for heating-dominated climates (Ye et al., 2014)

Fig. 10.
Hybrid CO₂ GSHPs to reduce the ground thermal imbalance

(a) compression-absorption (Kim et al., 2010) (b) compression-assisted absorption (Wu et al., 2016a)

Fig. 11.
Hybrid compression-absorption NH₃ GSHP

Fig. 12.
District heating combined with different GSHPs (Li et al., 2011)

Table 1

Properties of the natural refrigerants (Lemmon et al., 2013; Bolaji and Huan, 2013)

Refrigerant	R744	R717	R718	R290	R600a
Name	Carbon dioxide	Ammonia	Water	Propane	Isobutane
Molecular formula	CO ₂	NH ₃	H ₂ O	C ₃ H ₈	C ₄ H ₁₀
Molecular weight	44.01	17.03	18.02	44.10	58.12
ODP	0	0	0	0	0
GWP	1	0	0	3	3
Normal boiling point (°C)	−87.8	−33.3	100.0	−42.1	−11.7
Critical temperature (°C)	31.0	132.3	374.0	96.7	134.7
Critical pressure (kPa)	7377	11 333	22 064	4247	3640
<i>H</i> _{vap} at 5 °C (kJ·kg ^{−1})	215	1244	2489	368	350
Safety classification (ASHRAE, 2016)	A1	B2	A1	A3	A3

Table 2

Typical working conditions for GSHPs

Mode	Condition	Brine-to-air	Brine-to-water
Cooling	Heat source temperature (°C)	27	12
	Evaporating temperature (°C)	10	4
	Superheat (°C)	5	5
	Heat sink temperature (°C)	25	25
	Condensing temperature (°C)	30	30
	Subcooling (°C)	2	2
Heating	Heat source temperature (°C) ^a	0	0
	Evaporating temperature (°C) ^a	−8	−8
	Superheat (°C)	5	5
	Heat sink temperature (°C)	20	40
	Condensing temperature (°C)	34	45
	Subcooling (°C)	2	2

^a 10 °C higher for water to stay above freezing point

Table 3

Performance of the brine-to-air GSHP using natural refrigerants

Mode	Refrigerant	CO ₂	NH ₃	water	propane	isobutane
Cooling	Evaporator pressure (kPa)	4502.0	615.3	1.2	636.7	220.3
	Condenser pressure (kPa)	7214.0	1167.0	4.2	1079.0	404.5
	Discharge temperature (°C)	54.9	74.7	160.8	41.1	38.3
	Volumetric capacity (kJ·m ⁻³)	18896.0	5482.0	22.3	4309.0	1778.0
	COP	5.91	9.07	8.75	8.90	9.20
	COP/COP _{carnot}	N/A	N/A	N/A	N/A	N/A
Heating	Evaporator pressure (kPa)	2803.0	315.2	0.7	368.8	116.8
	Condenser pressure (kPa)	8000.0 ^a	1312.0	5.3	1189.0	452.0
	Discharge temperature (°C)	90.8	136.9	259.9	55.4	47.6
	Volumetric capacity (kJ·m ⁻³)	10502.0	2808.0	13.0	2294.0	869.8
	COP	3.23	4.71	5.93 ^b	4.60	4.72
	COP/COP _{carnot}	0.237	0.345	0.209	0.337	0.346

^aCondenser (or gas cooler) pressure was optimized for CO₂ transcritical cycle^bhigher COP due to higher heat-source temperature to stay above freezing point of water

Table 4

Performance of the brine-to-water GSHP using natural refrigerants

Mode	Refrigerant	CO ₂	NH ₃	water	propane	isobutane
Cooling	Evaporator pressure (kPa)	3869.0	497.7	0.8	535.2	180.2
	Condenser pressure (kPa)	7214.0	1167.0	4.2	1079.0	404.5
	Discharge temperature (°C)	62.7	89.6	208.7	44.6	39.8
	Volumetric capacity (kJ·m ⁻³)	16119.0	4455.0	15.0	3563.0	1431.0
	COP	4.371	6.67	6.33	6.55	6.76
	COP/COP _{carnot}	0.199	0.304	0.289	0.299	0.308
Heating	Evaporator pressure (kPa)	2803.0	315.2	0.7	368.8	116.8
	Condenser pressure (kPa)	11200.0 ^a	1782.0	9.6	1534.0	604.0
	Discharge temperature (°C)	124.8	171.6	355.2	68.7	58.8
	Volumetric capacity (kJ·m ⁻³)	9277.0	2672.0	12.8	2043.0	781.6
	COP	2.43	3.80	4.46 ^b	3.63	3.74
	COP/COP _{carnot}	0.356	0.557	0.473	0.531	0.548

^aCondenser (or gas cooler) pressure was optimized for CO₂ transcritical cycle^bhigher COP due to higher heat-source temperature to stay above the freezing point of water

Table 5

Summary of studies on GSHPs using natural refrigerants

Refrigerant	Topic	Remark
CO ₂	basic cycle	<ul style="list-style-type: none"> heating COP of GSHP was 31 % higher than that of ASHP (Jiang et al., 2009) heating COP peaked at 3.4 with high pressures of 9 MPa to 10 MPa (Lin et al., 2011)
	advanced cycle	<ul style="list-style-type: none"> SLHX improved cooling COP by 2 % to 6 % (Kim and Chang, 2013) expander improved cooling and heating performance (Ma et al., 2003) cooling COP was improved to 3.45 by two gas coolers (Jin et al., 2014) cooling COP was increased by 8 % using ejector (Morshed, 2015)
	direct-expansion system	<ul style="list-style-type: none"> modeled CO₂ geothermal borehole (Eslami-Nejad et al., 2014; Gao et al., 2017) heating COP was 2.58 after heat exchanger optimization (Austin and Sumathy, 2011) 25 % less borehole length increased the annual energy by 10 % (Eslami-Nejad et al., 2015) studied effect of different parameters on energy and exergy (Ghazizade-Ahsaei and Ameri, 2017)
	CO ₂ as a secondary fluid	<ul style="list-style-type: none"> modeling, validation, and heat transfer (Mastrullo et al., 2014; Ebeling et al., 2017) vertical CO₂ thermosyphon with heat extraction of 50 W/m to 71 W/m (Ochsner, 2008; Rieberer, 2005; Acuña et al., 2010) horizontal CO₂ thermosyphon with heat extraction of 13 W/m to 17 W/m (Rieberer, 2005)
	multi-source hybrid GSHP	<ul style="list-style-type: none"> heating COP was 3.8 for GSHP integrated with ambient air (Chargui et al., 2012) heating COP was 2.81 for solar-assisted GSHP (Kim et al., 2013) heating COP of CO₂ was 28.8 % lower compared with R22 (Choi et al., 2014)
	GSHP thermal imbalance	<ul style="list-style-type: none"> two gas coolers and rejected a part of heat to ambient (Hu et al., 2016; Jin et al., 2017) four gas coolers and rejected a part of heat to ambient and DHW (Jin et al., 2016) solar collector supplemented the heat supply (Ye et al., 2014)
NH ₃	vapor-compression cycle	<ul style="list-style-type: none"> 120 g NH₃ for 9 kW GSHP, heating COP was 3.8 to 4.8 (Palm, 2008) 8.4 kW GSHP for both space heating and DHW with a COP of 4 (Zajacs et al., 2017) two-stage cascade for high temperature, heating COP was 3.1 to 4.3 (Antonijevic and Komatina, 2011) two heat pumps connected in series for 85 °C district heating (Jensen et al., 2017)
	absorption cycle	<ul style="list-style-type: none"> low thermal imbalance and high primary energy efficiency in cold regions (Wu et al., 2013) borehole free cooling for low thermal imbalance and higher efficiency (Wu et al., 2014a) heat recovery to produce DHW and reduce thermal imbalance (Wu et al., 2014b) integrated with cooling tower to reduce thermal imbalance and borehole number (Wu et al., 2015a)
	hybrid compression and absorption cycle	<ul style="list-style-type: none"> high-temperature hot water, COP 10 % to 20 % higher than R134a (Kim et al. 2010) performance improvement under low driving temperatures (Wu et al., 2016a) and low evaporating temperatures (Wu et al., 2016b) combining compression and absorption to compensate for each other (Wu et al., 2015b) improvement for serially-connected GSHP for district heating (Jensen et al., 2017)
Water	solar cooling	<ul style="list-style-type: none"> single-effect H₂O-LiBr absorption unit with a cooling COP of 0.7 (María et al., 2013)
	thermal balance	<ul style="list-style-type: none"> H₂O-LiBr was suitable for higher soil temperature to avoid freezing (Wu et al., 2013)
	district heating	<ul style="list-style-type: none"> saving energy by 23 % to 46 % compared to conventional district heating (Li et al., 2011)
Propane	refrigerant charge	<ul style="list-style-type: none"> for borehole brine outlet of -2.2 °C to 3.8 °C, 200 g charge was the best with COP of 3.5 to 4.0 (Fernando et al., 2004)
	system performance	<ul style="list-style-type: none"> daily COP was 3.5 to 4.2 for heating and 4.2 to 5.1 for cooling during long-term operation (Corberan et al., 2011; Montagud et al., 2011; Ruiz-Calvo et al., 2016)
	comparison with ASHP	<ul style="list-style-type: none"> energy saving was 43 % in heating season and 37 % in cooling season (Urchueguía et al., 2008; Tammaro et al., 2016)
	propane as a secondary fluid	<ul style="list-style-type: none"> measurement, modeling, and parameter studies of thermosyphons (Grab et al., 2011; Hantsch et al., 2013; Hartmann et al., 2015)