ORNL/TM-2024/3521

CRADA NFE-23-09890 Final Report - Scoping Study of Integrated Geothermal CO_2 Heat Pump and Water Well System in Cold Climate Regions

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October 2025

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ORNL/TM-2024/3521

Building and Transportation Division

CRADA REPORT: SCOPING STUDY OF INTEGRATED GEOTHERMAL CO_2 HEAT PUMP AND WATER WELL SYSTEM IN COLD CLIMATE REGIONS

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UT-BATTELLE LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

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ABBREVIATIONS

ASHP	air-source heat pump
CCHP	cold-climate heat pump
COP	coefficient of performance
DOE	US Department of Energy
GPM	gallons per minute
GSHP	ground-source heat pump
HP	heat pump
HX	heat exchanger
SPF	seasonal performance factor
WSHP	water-source heat pump

ACKNOWLEDGMENTS

The authors acknowledge support provided by Samuel Petty, Nathaniel Allen, Michael Blunschi, Michael Sheppy, Marc Lafrance, and Hayes Jones from the Commercial Building Integration Program at US Department of Energy's Building Technologies Office.

ABSTRACT

This study evaluates the feasibility of using ground-source heat pumps (GSHPs) integrated with water wells for residential heating demands in cold regions of the United States. Four different heat pump systems were evaluated and compared: a CO_2 air-source heat pump (ASHP), a R-410A ASHP, a CO_2 GSHP integrated with well water, and a R-410A GSHP integrated with well water. Simulations were conducted at both equipment and building integration levels, followed by a nationwide analysis across 10 selected cities. Results indicate that integrating GSHP technology with existing water wells (e.g., artesian, bedrock, drilled, and collector wells) is a feasible heating solution because of their high flow rates, good water quality, stable temperatures, and low environmental impact. Although R-410A systems demonstrate higher efficiency, CO_2 systems are more suitable for heating from contamination-free and energy security perspective. The CO_2 GSHPs maintain better performance in colder climates compared with that of ASHPs, providing higher coefficient of performance and environmental benefits because CO_2 refrigerant leakage would not contaminate well water. A national analysis confirms that CO_2 GSHPs outperform furnaces, especially in cold climates, making them a promising solution for residential heating.

1. INTRODUCTION AND OBJECTIVE

According to data on new single-family homes [1], of the approximately 3,584,000 new homes built in the last 5 years (2019–2023), forced-air furnace and heat pump (HP) technology are the main heating methods used in US homes. Forced-air furnace heating is still the most popular heating method as it is used in 58.1% of new homes, as shown in Table 1. Heat pump technology is used by 41.1% of homes (or around 1,191,000). Other heating methods make up less than 1%. As for fuels powering home heating systems, gas is still the primary fuel used for over 85% of forced-air furnaces. Electricity powers around 89% of heat pumps. Overall, 54.7% of new home heating systems are powered by gas, and 45.1% are powered by electricity.

Table 1. Heating systems and fuels breakdown for single-family homes built in the last 5 years.

System used	Houses built	Powered by gas (%)	Powered by electricity (%)	Powered by other (%)
Forced-air furnace	2,096,000	85.1	14.8	<0.5
Heat pump technology	1,457,000	11.3	88.7	0
Other/none	31,000	32.3	41.9	25.8

Forced-air furnaces remain the dominant heating system in the US, with 2,096,000 installations compared with 1,457,000 HPs. However, the South stands out as an exception, where HPs are more prevalent, far outpacing forced-air furnaces with 1,235,000 installations, as shown in Figure 1. This trend aligns with the region's milder winters, making HPs a more viable and efficient option. Gas is still the preferred heating fuel in most regions (Figure 2), particularly in the Northeast and Midwest, where it powers the majority of heating systems. Electricity is more commonly used in the South, supporting the widespread adoption of HPs.

The transition to efficient heating in the US faces several challenges, particularly in cold regions such as the Northeast and the Midwest. These areas are heavily reliant on forced-air furnaces and gas heating because of harsh winter climates, where consistent and reliable heating is critical. The performance of current HP technologies in extremely cold temperatures is a major barrier because these systems may not efficiently meet heating demands. The minimal adoption of electric heating in these regions—less than 1% in the Northeast and less than 2.5% in the Midwest—also underscores consumer skepticism and the perceived risks associated with switching from traditional gas-fired heating systems to new HP technologies.

To overcome the challenges in heating electrification, technological advancements are crucial, particularly the development of various cold-climate heat pump (CCHP) technologies that can operate efficiently in extreme temperatures.

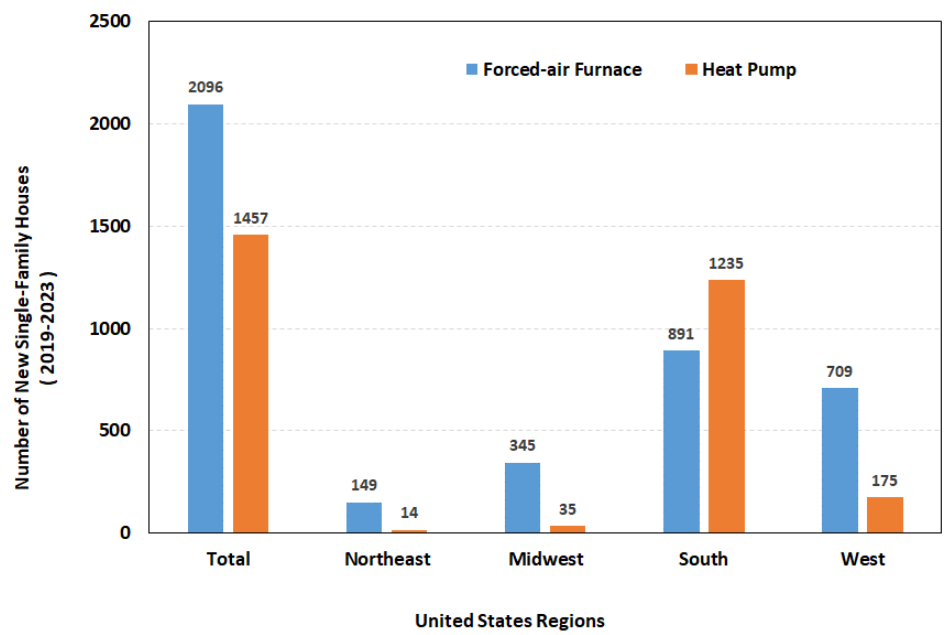

Figure 1. Types of heating systems used in new single-family houses in the last 5 years (2019–2023).

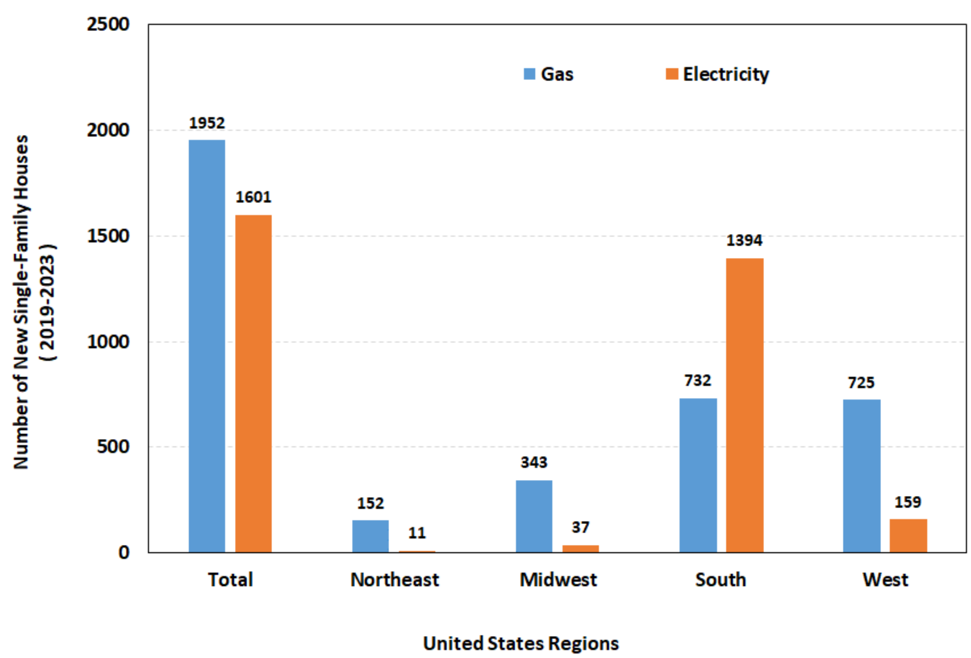

Figure 2. Types of heating power sources used in new single-family houses in the last 5 years (2019–2023).

1.1 HEAT PUMP TECHNOLOGIES FOR COLD CLIMATE REGION

In cold climates, heating demands are high in the heating season. Although forced-air furnace heating dominates these regions, finding an efficient solution that also reduces energy consumption is desired. In general, three types of HP technologies can be adopted in cold climates [2,3]: air-source heat pump (ASHP), water-source heat pump (WSHP), and ground source heat pump (GSHP).

1.1.1 Air-Source Heat Pump (ASHP)

ASHPs have become an increasingly popular choice for heating and cooling in residential buildings owing to their energy efficiency and environmental benefits. ASHPs transfer heat from the outside air into the home, even at lower outdoor temperatures. This makes them a more sustainable and often cost-effective option compared with conventional systems like gas furnaces or electric resistance heaters.

However, drawing heat from outside air also presents a great challenge for operating in a cold climate: the efficiency and reliability of standard ASHPs can be compromised. At low temperatures, the HP must work harder to extract heat from the outside air, which can lead to decreased performance, higher energy consumption, and in some cases, the need for auxiliary heating to achieve indoor comfort. A study in Canada [2] shows the seasonal performance factor (SPF) is around 2.3–3.5 in cold seasons. Another study shows that the ASHP system can reach an SPF of 3.43, with its lowest coefficient of performance (COP) of 2.6 during the coldest month of January in a single-family building in Latvia [4]. The cold-climate ASHP is an active topic in academic research. The latest review study identified the following potential challenges and risks of ASHP in cold climates [5]:

- Reduced COP and heating capacity
- Increased compressor discharge temperature because of working at full capacity to maintain the heating temperature setpoint
- Risk of compressor refrigerant slugging and flooding
- Nonoptimal refrigerant charge
- High defrosting needs
- Excessive cycling during partial load conditions
- Increased heat exchanger (HX) area leading to higher cost (This conclusion pointed out a potential area for study which is varying HX surface area in different climates)

Recognizing these challenges, manufacturers and researchers have developed CCHPs specifically designed to operate efficiently in lower temperature conditions. These systems often incorporate advanced features such as enhanced compressor technologies, variable-speed motors, and defrost mechanisms that allow them to maintain higher efficiency and heating capacity, even when outdoor temperatures fall well below freezing. The continued development of CCHPs is critical for advancing the use of ASHPs in cold climates and plays a significant role in the transition to more efficient heating solutions in cold climate regions.

1.1.2 Ground-Source Heat Pump (GSHP)

A GSHP, also called a geothermal HP, is a type of heating and cooling system that uses the ground as a heat source in winter and a heat sink in summer. It is a renewable energy technology that leverages the relatively constant temperatures found beneath the earth's surface to provide efficient space heating, cooling, and hot water heating for residential and commercial buildings. GSHP systems use a series of underground pipes, called a ground loop, that circulates heat-transfer fluid (usually a mixture of water and antifreeze). This ground loop is buried horizontally in trenches or vertically in boreholes, depending on available space and soil conditions.

In cold climates, where outdoor air temperatures are very low, GSHPs offer distinct advantages. The ground temperature at a certain depth remains relatively stable throughout the year, typically between 10°C and 15°C, even in winter. This stable thermal environment allows GSHPs to operate more efficiently and consistently compared with ASHPs, which may struggle as outdoor air temperatures drop. As a result, GSHPs can provide steady and efficient heating, reducing the need for auxiliary heating systems and lowering energy consumption. One study shows that typical SPF values for GSHPs are 2.6–3.6, and the equivalent COP of GSHPs for various heating systems can vary between 3 and 5 in Finland and Canada [2].

However, the biggest challenge for GSHPs is the initial installation cost, especially high drilling costs, which is expected to be offset by long-term savings in energy costs and the extended lifespan of the system. Additionally, concerns about potential environmental impacts, such as refrigerant leakage into groundwater, necessitate the implementation of robust, leak-free system designs. These drawbacks are the main barriers to positioning GSHPs as an energy-efficient heating solution in cold climates.

1.1.3 Water-Source Heat Pump (WSHP)

A WSHP draws heat from groundwater or from a body of surface water (e.g., lakes and rivers) that maintains relatively stable temperatures throughout the year. This stable thermal environment makes WSHPs highly effective, even in regions where winter temperatures can be extremely low.

The efficiency of WSHPs in cold climates is large because extreme temperatures do not affect water bodies to the same extent that they affect the air. This means that WSHPs can operate more reliably and with higher efficiency than ASHPs, which often struggle to extract sufficient heat from frigid air. Studies show that WSHPs have a COP similar to that of GSHPs [6], with the following considerations:

- Water quality is extremely important because water pollution might cause system fouling, corrosion, and blockage.
- Available water must have adequate volume and flow rate. This is key to ensuring a stable heating supply to the residents.
- Two options are available for handling discharged water: the groundwater can be reinjected into the ground through separate wells, or it can be discharged into surface water basins (i.e., rivers and lakes).
- The feasibility of open-loop systems depends on local codes and regulations [3].

Barriers to the widespread adoption of WSHPs in cold climates include high initial installation costs and environmental considerations. Despite these challenges, WSHPs are becoming an increasingly attractive option for cold-climate heating because of their efficiency, reliability, and potential for long-term cost savings.

1.1.4 Summary

As described in Table 2, the comparison of ASHP, GSHP, and WSHP systems in cold climates highlights the unique advantages and disadvantages of each. ASHPs offer the advantage of lower initial costs and simpler installation, making them accessible for many homeowners. However, their efficiency can decrease significantly in extremely cold weather, potentially leading to higher operating costs and the need for auxiliary heating. To address these limitations, manufacturers have developed cold-climate ASHPs to improve performance, but they still may not match the efficiency of GSHPs and WSHPs.

On the other hand, GSHPs and WSHPs provide more reliable and efficient heating in cold climates because of the stable temperatures of the ground and water sources on which they rely. GSHPs are particularly well-suited for cold regions, offering low operating costs and high efficiency, though they require a significant initial investment and extensive land for installation. WSHPs also deliver high efficiency and reliability, especially when located near a suitable water source. However, the installation of WSHPs can be complex and costly, with additional environmental considerations to manage. Overall, ASHPs are more affordable and easier to install, whereas GSHPs and WSHPs are superior in terms of efficiency and reliability in cold climates but come with higher installation costs and complexity.

Table 2. Comparison of ASHP, GSHP, and WSHP heating systems in cold climate applications.

Cold climate application	ASHP	GSHP	WSHP
Efficiency	**Moderate:** efficiency reduction in low ambient air temperatures	**High:** stable performance	**High:** stable performance
Initial cost	**Low:** least expensive option	**High:** expensive in-ground loop installation	**Moderate:** cost to access the water source
Operating cost	**Moderate:** efficiency declines, leading to high energy consumption	**Low:** lower long-term energy consumption	**Low:** lower long-term energy consumption
Environmental impact	**Low:** relatively low emissions	**Moderate:** potential risk of refrigerant leakage	**Moderate:** potential risk of refrigerant leakage
Reliability	**Moderate:** might need supplemental heating	**High:** stable ground temperature	**High:** stable water temperature
Space requirement	**Low:** minimal land area	**High:** land for the ground loop	**Moderate:** land to access the water source

Considering the analysis in Table 2, integrating GSHP technology with existing water wells in cold climates presents a promising heating solution. This approach leverages the natural thermal stability of groundwater, which is consistently warmer than the air during cold winters, thereby enhancing the efficiency and reliability of the HP system. By using existing water well, homeowners can avoid one of the most significant barriers to GSHP adoption: the high initial installation cost of ground loops. Ground loop installation, which often involves extensive drilling or excavation, represents a substantial portion of the upfront investment for a traditional GSHP system. By integrating the GSHP with an existing water well, this costly and land-intensive component can be eliminated, making the system more accessible and economically feasible for a wider range of homeowners.

Overall, integrating GSHP technology with existing water wells presents a highly effective strategy for addressing the unique challenges of heating in cold climates. The efficiency and reliability of GSHPs combined with the practicality and cost-effectiveness of using existing water infrastructures offers a powerful heating solution in these regions.

1.2 TYPES OF WATER WELLS

An estimated 15.1 million housing units and more than 43 million individuals—approximately 15% of the US population—depend on private domestic wells for their drinking water [7]. These water wells provide around 3.6 billion gal of freshwater for domestic water usage every day [8]. Figure 3 shows the estimated number of households using domestic water wells in each state [9].

In contrast to public water supplies, domestic wells typically tap into shallow groundwater sources and do not fall under the regulatory purview of the Safe Drinking Water Act [10].

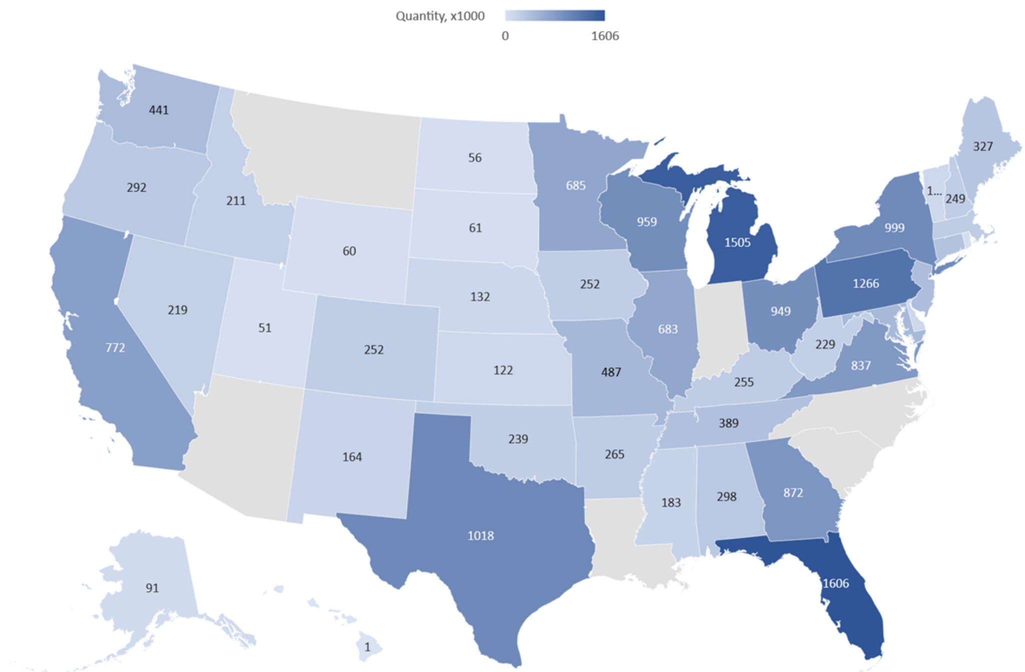

Figure 3. US households using private water wells [3].

Many types of domestic water wells are available, each of which draws water from different sources and employs varying construction methods. The following are common types of domestic water wells:

- Dug wells (hand dug or machine dug): Dug wells are among the oldest and simplest types of wells. They are manually or mechanically excavated by digging or drilling into the ground until the water table is reached. Dug wells are typically lined with materials such as brick, stone, or concrete to prevent collapse (Figure 4).

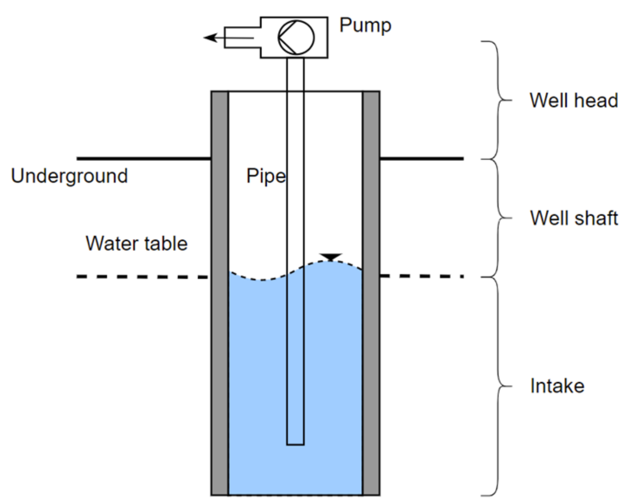

Figure 4. Schematic of a dug well [11].

- Bored wells: These wells are created by drilling a hole into the ground using a specialized auger or boring machine. They are typically larger in diameter than driven wells and are suitable for accessing water at greater depths (Figure 5).

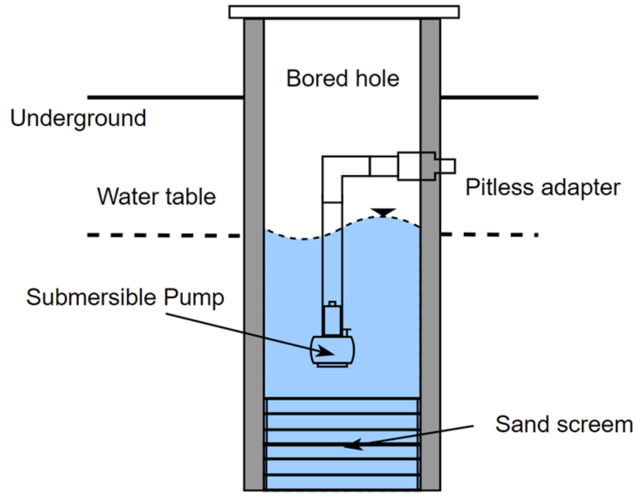

Figure 5. Schematic of a bored well (redrawn from [12]).

- Driven wells: Also known as sand-point or sand-jacked wells, driven wells are constructed by driving a narrow pipe or casing into the ground using a heavy weight or mechanical driver. These wells are suitable for shallow water tables and loose, sandy soils (Figure 6).

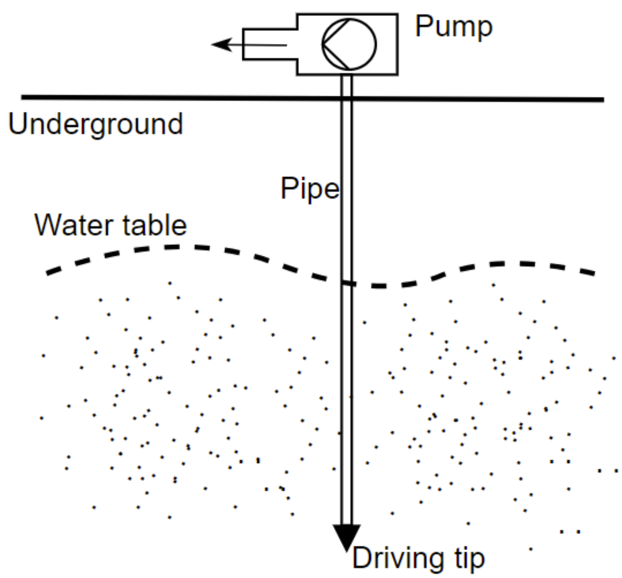

Figure 6. Schematic of a driven well (redrawn from [13]).

- Drilled wells (rotary or percussion drilled): Drilled wells are constructed using drilling rigs and can reach greater depths than those of other well types. The two primary types are rotary and percussion drilled. Rotary-drilled wells use a rotating drill bit to bore through the earth, and percussion-drilled wells use a hammering action to break through rock formations (Figure 7).

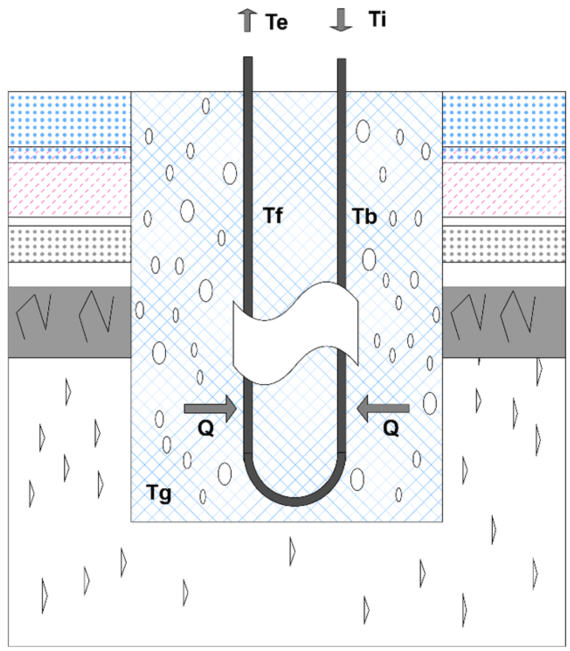

Figure 7. Schematic of a drilled well (redrawn from [14]).

- Bedrock wells: These wells are drilled into solid bedrock formations, which are common in some regions. Bedrock wells are used when the water table is deep, and the well must reach below loose, unconsolidated material (Figure 8).

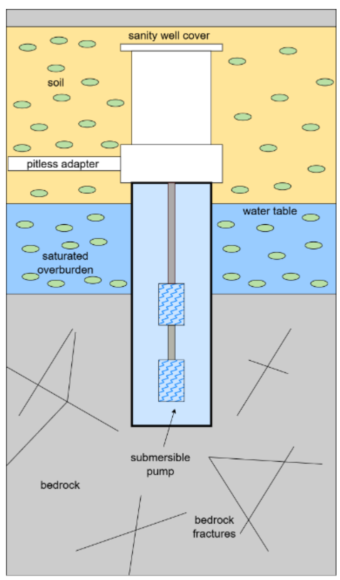

Figure 8. Schematic of a bedrock well (redrawn from [15]).

- Artesian wells (flowing wells): Artesian wells tap into confined aquifers in which the water is under pressure. When the well is drilled into the aquifer, the water may rise to the surface without the need for a pump, creating a flowing well (Figure 9).

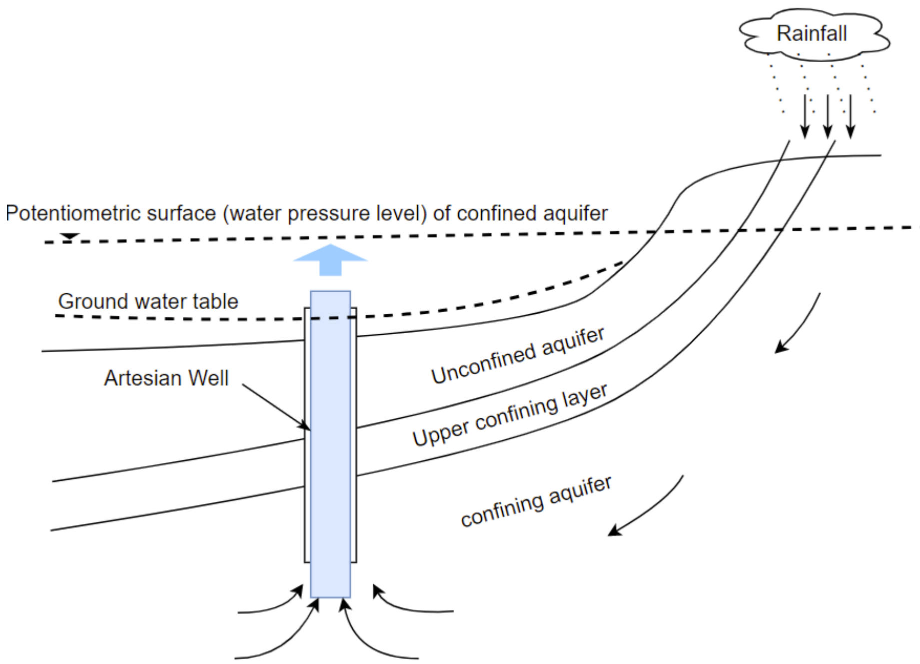

Figure 9. Schematic of an artesian well.

- Collector or infiltration wells: These wells, designed to collect surface water and direct it into an aquifer for storage or recharge, are used in areas with specific water management needs (Figure 10).

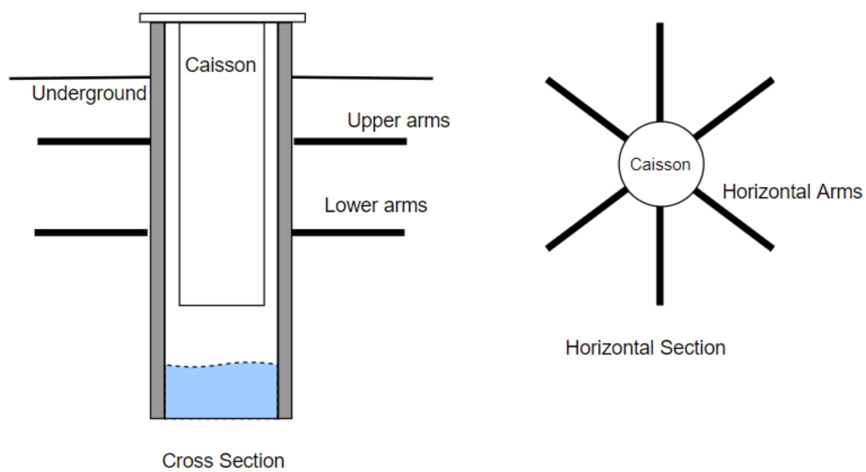

Figure 10. Schematic of a radial collector well (redrawn from [16]).

When integrating a domestic water well with an HP system, ensuring the well meets specific criteria to support efficient and sustainable operation is crucial (Table 3). The well must provide a consistent and adequate flow rate to match the HP's demand, with most residential systems requiring between 1.5 and 2.5 gal/min per ton of capacity. Water quality is also vital; the water should have low levels of minerals and contaminants to prevent scaling, fouling, and corrosion of the HXs. Additionally, the well-water temperature should remain relatively stable throughout the year because HPs are designed to operate efficiently within a specific temperature range, usually between 4°C and 15°C.

Environmental and legal considerations are equally important. The system must be designed to return water to the aquifer or a separate discharge location without causing thermal pollution or affecting local groundwater levels. Proper well construction is essential to prevent surface water contamination, and regular maintenance is necessary to ensure long-term performance. The well type and its compatibility with the HP system should be carefully evaluated.

Table 3. Water well comparison.

Well type	Water flow rate	Water quality	Temperature stability	Environmental impact	Maintenance cost
Dug well	Low	Poor	Less stable	Risk	High
Driven well	Moderate	Fair	Less stable	Moderate risk	Moderate
Bored well	Moderate	Fair	Unstable	Risk	High
Drilled well	High	Good	Stable	Low risk	Low
Artesian well	High	Very good	Very stable	Minimal risk	Very low
Bedrock well	High	Good	Very stable	Low risk	Low
Collector well	High	Good	Stable	Low risk	Moderate

The following is a summary of water well suitability for HP integration:

- Highly suitable: artesian, bedrock, drilled, and collector wells. These typically provide high flow rates, good water quality, stable temperatures, and low environmental impact, making them excellent choices for integration with HPs.

- Moderately suitable: driven wells. These may be suitable for smaller systems or low-demand applications but require careful design and regular monitoring.

- Not recommended: dug and bored wells. These typically have lower flow rates, higher risk of contamination, and less stable temperatures, making them less ideal for HP integration.

1.3 PROBLEM AND OBJECTIVE

In cold regions of the United States, residential homes can use GSHPs to provide efficient heating during the winter. These systems typically use well water as a heat source, extracting heat from groundwater to warm the home. GSHPs are known for their high efficiency and reduced energy consumption compared with that of conventional gas-fired heating systems. However, they present a potential environmental risk: the possibility of refrigerant leakage into the well water.

Refrigerants are crucial for the operation of HPs because they undergo phase changes that facilitate the transfer of heat from one place to another. However, if a refrigerant leak occurs in a GSHP system, the refrigerant could escape into the well water and contaminate the groundwater supply. This is a particularly concerning issue in cold climates, where reliance on well water is common because other viable water sources are limited. Contamination of well water by refrigerants poses significant environmental and health risks. Many refrigerants are not biodegradable and can persist in the environment for long periods, potentially leading to the degradation of water quality. Depending on the type and concentration of the refrigerant, the contaminated water could pose health hazards to residents who rely on the well for drinking water. Moreover, the environmental impact extends beyond human health because contaminated groundwater can affect local ecosystems, harming aquatic life and disrupting natural water cycles.

Addressing refrigerant leaks can be challenging and costly, especially in cold climates where GSHPs are most beneficial, making mitigation of this risk crucial from the outset. Given the serious risks associated with refrigerant leakage, exploring solutions that eliminate the potential for environmental and health hazards in GSHP systems is essential. Instead of focusing on preventing leaks—because any leakage poses an unacceptable risk—we must prioritize the adoption of technologies and system designs that inherently avoid the use of harmful refrigerants or contain them in a way that guarantees zero leakage.

The main purpose of this scoping study is to conduct detailed feasibility analysis to identify an efficient heating approach for households in cold regions of the United States to substitute conventional gas-fired heating with well water integration while avoiding well contamination through refrigerant leakage.

2. MATHEMATICAL MODELING

A steady-state simulation model for HP systems has been developed for supporting the system design and performance prediction.

The compressor model includes the refrigerant mass flow rate prediction, power consumption estimation, and energy balance calculation.

The compressor suction side mass flow rate, m_r, is

$$m_r = \rho_s \eta_v V_{disp} \tag{1}$$

$$W = \left[C_1 \left({P_d}/{P_s}\right)^{\gamma-1/\gamma} + C_2\right] \times P_s V_s + C_3 \tag{2}$$

$$W = m_s(h_d - h_s) + Q_{loss}, \tag{3}$$

where ρ_s is compressor suction side refrigerant density; V_{disp} is compressor displacement rate; η_v is compressor volumetric efficiency; P_s and P_d are compressor suction and discharge pressure; and C_1, C_2, and C_3 are constants. The terms h_s and h_d are compressor suction and discharge enthalpy, and Q_{loss} is ambient heat loss, which can be calculated as below empirical function of ambient temperature and compressor discharge temperature.

The compressor volumetric efficiency can be given by the following two-term model:

$$\eta_v = C_4 + C_5 \times (P_d/P_s)^{1/\gamma}, \tag{4}$$

where γ is the isentropic exponent, and C_4 and C_5 are constants.

In the evaporator, heat transfer between the refrigerant and fluid is balanced with the capacities on the refrigerant side and air side (Figure 11).

On the refrigeration side, heat transfer in the evaporator is divided into two zones: the superheat zone and the two-phase zone.

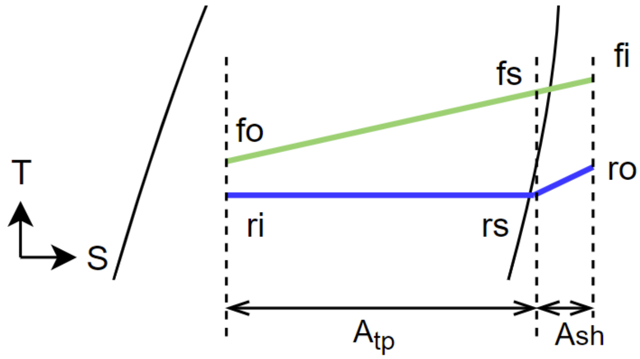

Figure 11. Schematics of evaporator zone modeling. (Abbreviations: A_{sh} = superheat zone surface area; A_{tp} = two-phase zone surface area; fi = fluid in; fo = fluid out; fs = fluid saturation point; ri = refrigerant in; ro = refrigerant out; rs = refrigerant saturation point.)

Superheat zone heat transfer can be calculated as follows:

$$Q_{ref\_sh} = m_r \times (h_{ro} - h_{rs}) \; = \; U_{sh} f_{sh} A \Delta T_{m\_sh} \, . \tag{5}$$

Two-phase zone heat transfer can be calculated as follows:

$$Q_{ref\_tp} = m_r \times (h_{rs} - h_{ri}) \; = U_{tp} f_{tp} A \Delta T_{m\_tp}, \tag{6}$$

where

- m_r is the refrigerant mass flow rate;
- h_{ri} and h_{ro} are refrigerant enthalpy at the evaporator inlet and outlet;
- h_{rs} is the refrigerant enthalpy at the saturated vapor refrigerant state (quality = 1);
- A is the evaporator surface area;
- f_{sh} and f_{tp} are the ratio of superheat zone and two-phase zone surface areas to evaporator surface area;
- ΔT_{m_sh} and ΔT_{m_tp} represent the log-mean-temperature difference at the superheat and two-phase zones; and
- U_{sh} and U_{tp} are overall heat transfer coefficients at the superheat and two-phase zones.

The fluid side heat transfer and energy balance become

$$Q_{ref\_sh} + Q_{ref\_tp} = m_f \times \left(h_{fi} - h_{fo}\right) \; - \; m_w h_w, \tag{7}$$

where

- m_f is air mass flow rate;
- h_{fi} and h_{fo} are the evaporator entering and leaving air enthalpy; and
- h_w is condensate enthalpy if the fluid is air.

When the fluid is vaporized, the condensate mass flow rate, m_w, can be calculated by

$$m_w = m_a(W_{ai} - W_{ao}), \tag{8}$$

where W_{ai} and W_{ao} are humidity ratios of air entering and leaving the evaporator, calculated based on air pressure and air dry- and wet-bulb temperatures.

Similarly, the condenser can also be modeled using the zone model approach that divides the condenser into three zones: superheat, two-phase, and subcool (Figure 12).

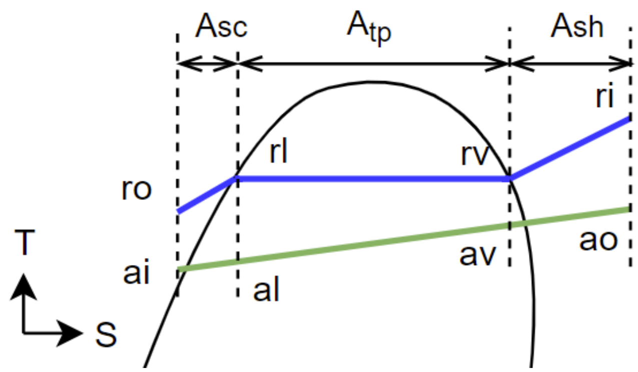

Figure 12. Schematics of condenser zone modeling. (Abbreviations: ai = air in; al = air liquid; ao = air out; av = air vapor; A_{sc} = subcool zone surface area; A_{sh} = superheat zone surface area; A_{tp} = two-phase zone surface area; ri = refrigerant in; rl = refrigerant liquid; ro = refrigerant out; rv = refrigerant vapor.)

The refrigerant side heat transfer balance equations are the following:

$$Q_{ref\_sh} = m_r \times (h_{ri} - h_{rv}) \ = \ U_{sh} f_{sh} A_{cd} \Delta T_{m\_sh} \tag{9}$$

$$Q_{ref\_tp} = m_r \times (h_{rv} - h_{rl}) \ = U_{tp} f_{tp} A_{cd} \Delta T_{m\_tp} \tag{10}$$

$$Q_{ref\_sc} = m_r \times (h_{rl} - h_{ro}) \ = U_{sc} f_{sc} A_{cd} \Delta T_{m\_sc}, \tag{11}$$

where

- m_r is refrigerant mass flow rate;
- h_{ri} and h_{ro} are refrigerant enthalpy at the evaporator inlet and outlet;
- h_{rv} and h_{rl} are the refrigerant enthalpy at the saturated vapor refrigerant state (quality = 1) and liquid refrigerant state (quality = 0);
- A_{cd} is the condense surface area;
- f_{sh}, f_{tp}, and f_{sc} are ratios of superheat and two-phase zone surface areas to condenser surface area;
- ΔT_{m_sh}, ΔT_{m_tp}, and ΔT_{m_sc} represent the log-mean-temperature difference at the superheat, two-phase, and subcool zones; and
- U_{sh}, U_{tp}, and U_{sc} are overall heat transfer coefficients at the superheat, two-phase, and subcool zones.

The air side heat transfer and energy balance become

$$Q_{ref\_sh} + Q_{ref\_tp} + Q_{ref\_sc} = m_a \times (h_{ai} - h_{ao}), \tag{12}$$

where m_a is air mass flow rate; and h_{ai} and h_{ao} are the evaporator entering and leaving air enthalpy.

Expansion valve performance is simulated using a semithermodynamic model that balances the predicted refrigerant mass flow rate with the compressor model to achieve system convergence. The expansion process is assumed to be isentropic, and the mass flow rate through the valve is a function of the pressure drop across it:

$$m_v = C_v\sqrt{\rho_{in}\Delta P_v}, \tag{13}$$

where C_v is the flow coefficient.

3. SYSTEM DESIGN

To address the challenges and environmental impacts of refrigerant leakage, one potential solution is the use of alternative heat transfer fluids that are nontoxic and environmentally benign, such as carbon dioxide (CO_2), thereby eliminating the risk of contaminating well water. These fluids would replace traditional refrigerants in HP systems, ensuring that even a system malfunction would pose no threat to the environment or human health. Another approach could involve advanced HP designs that completely isolate the refrigerant from the water source, using secondary loops or indirect heat exchange systems that provide the same efficiency without the direct risk of contamination.

The integration of CO_2 GSHPs with residential water wells not only could mitigate the risks associated with leaks but also could provide a safer, more sustainable solution for residential heating in cold climates. This new design could ensure that GSHP systems remain a safe and sustainable heating option for residential homes in cold climates, without the environmental and health risks posed by refrigerant leakage.

3.1 RESIDENTIAL BUILDINGS

The US Department of Energy (DOE) has developed a series of prototype residential buildings that serve as standardized EnergyPlus models for evaluating energy performance and efficiency across various climates in the United States. These prototypes represent typical residential structures, providing a consistent baseline for energy simulations and analyses.

A target residential building is selected from these standard prototype residential buildings [17]. The building used in this analysis (Figure 13) is representative of a single-family home—one of the most common types of residential buildings in the United States—with a total area of 2,377 square feet, one floor, and one attic area that is not thermally controlled. This prototype is designed to reflect the construction practices, materials, and equipment commonly found in homes built according to modern building codes. Key features of the prototype include standard wood-frame construction, typical insulation levels, and conventional heating/cooling systems, making it an ideal candidate for evaluating the performance of alternative heating technologies, such as HPs.

The prototype will be modeled in various climate zones across the United States, allowing the assessment of heating and cooling systems under different environmental conditions (Figure 14). This ensures that the results of the simulations are broadly applicable, providing insights into how the HP systems under study would perform in real-world residential settings. By comparing the energy performance of advanced HPs with a baseline furnace heating system in this DOE prototype, we can gain a deeper understanding of the potential energy savings, cost benefits, and environmental impacts associated with transitioning to more efficient heating technologies in the US residential sector.

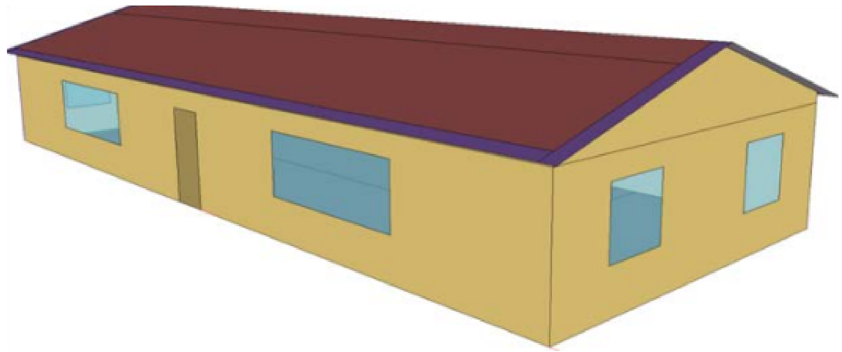

Figure 13. A typical residential building from the DOE prototype residential buildings models.

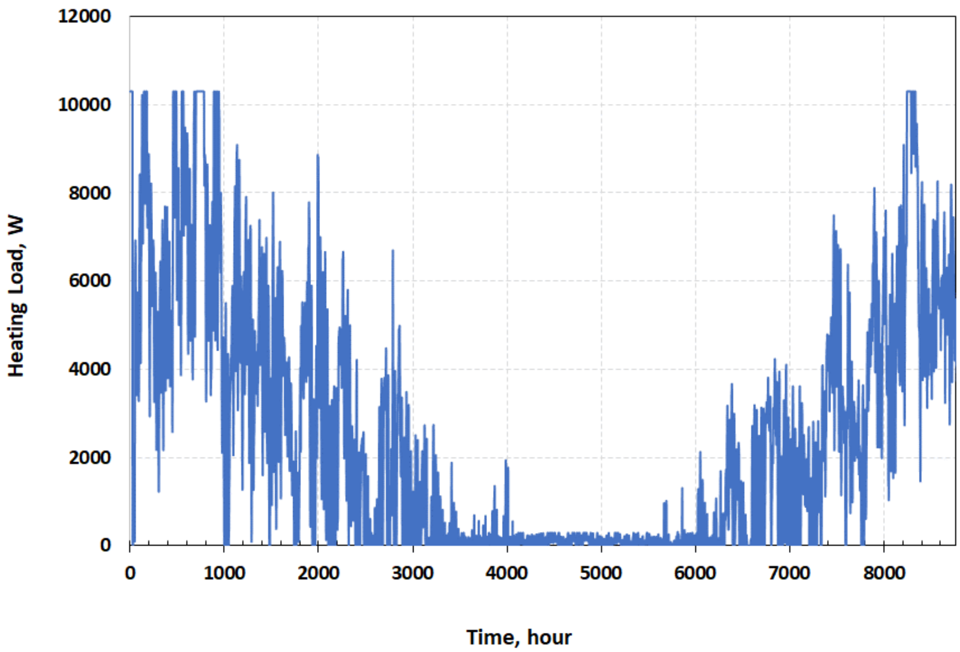

Figure 14. Simulated annual heating load profile of a typical residential building in US climate zone 6.

3.2 SYSTEM CONFIGURATION

The water well integrated CO_2 GSHP, illustrated in Figure 15, is designed for residential heating and cooling in cold climates. Key components include the following:

- A domestic water well serves as the primary source of groundwater.
- A submersible pump within the water well extracts groundwater.
- An injection well reintroduces the cooled water back into the aquifer to maintain the thermal and hydrological balance of the underground water source.
- A CO_2 HP system is composed of
  - an outdoor unit that is a water-to-refrigerant HX and a CO_2 refrigerant loop,
  - a compressor using CO_2 as refrigerant,

- an indoor unit equipped with a fin-coil HX, and
- an expansion valve.

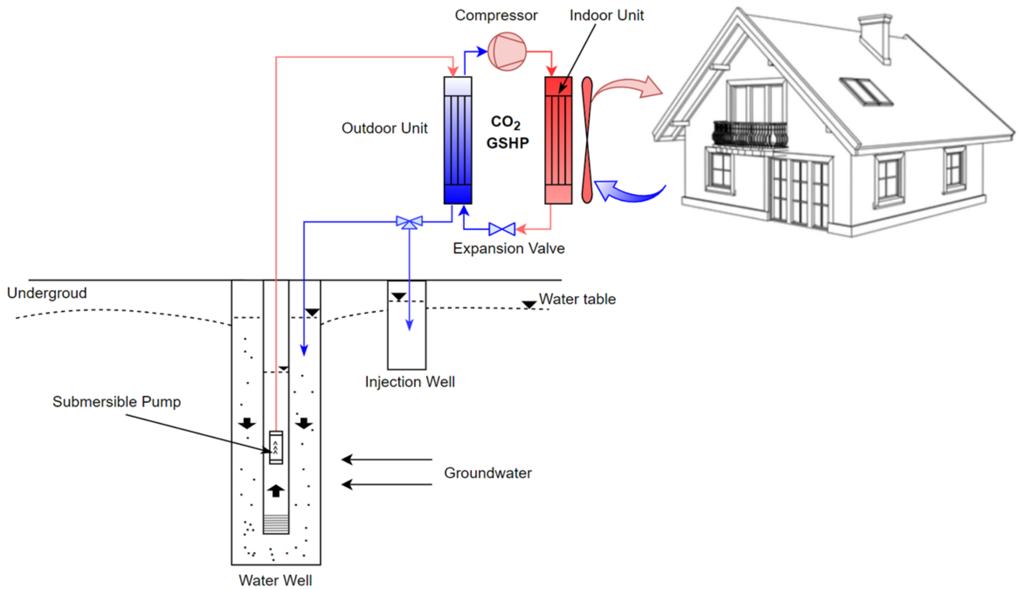

Figure 15. System configuration of a water well integrated CO_2 GSHP.

When the system operates, the submersible pump draws groundwater from the well, and the water flows into the outdoor HX unit where it transfers heat to the CO_2 refrigerant within the HX. The heated refrigerant is then compressed, increasing its temperature and pressure before circulating to the indoor fin-coil HX unit. Inside the home, the fin-coil HX releases heat, warming the indoor air. Afterward, the refrigerant is expanded, reducing its temperature and pressure, and the cycle repeats. Meanwhile, the cooled groundwater is directed either back to the water well or to an injection well, returning it to the underground aquifer.

This water well integrated CO_2 GSHP system offers significant environmental and operational benefits, particularly in cold climates. By using CO_2 as a refrigerant, the system minimizes global warming potential compared with that of traditional refrigerants. The stable groundwater temperature ensures reliable heating efficiency throughout the winter, reducing reliance on less efficient heating methods. Additionally, integrating the system with an existing water well reduces installation costs and land use. However, although CO_2 is a nontoxic and environmentally benign refrigerant that would not harm groundwater in the event of a leak, the lubricant oil used for the CO_2 compressor is mixed with CO_2 during operation and could pose environmental and health hazards. The potential solution is to identify and use a nontoxic and environmentally benign lubricant oil or use an oil-free compressor [18,19] in this proposed water well integrated CO_2 GSHP system.

3.3 DESIGN CONDITIONS

These HP systems are designed based on the winter design temperature assumption [19] that the winter temperatures are 47°F, 17°F, and 5°F, with the lowest expected winter temperature of −15°F (−26.1°C) (Table 4). After conducting a detailed heating and cooling load calculation for the residential building in climate zone 6 described in Section 2.3, the heating load is estimated at 50,000 btu/h, reflecting the need for a robust heating solution at a 47°F outdoor air temperature. These systems are designed to operate efficiently to maintain an indoor temperature of 68°F (20°C) during the heating season and can operate at temperatures as low as −15°F (−26.1°C) or lower.

Table 4. Heat pump design conditions.

Heating	Case 1	Case 2	Case 3	Case 4
Outdoor	47°F (8.3°C)	17°F (−8.3°C)	5°F (−15°C)	−15°F (−26.1°C)
Indoor	68.0°F (20°C) DB, 59.0°F(15°C) WB			

Note: DB = dry-bulb temperature; WB = wet-bulb temperature.

3.4 SYSTEM DESIGN

For comparison purposes, three additional HP systems were designed to evaluate the potential benefits of the proposed water well integrated CO_2 GSHP system. The four HP systems are R-410A ASHP, water well integrated R-410A GSHP, CO_2 ASHP, and water well integrated CO_2 GSHP (Figure 16).

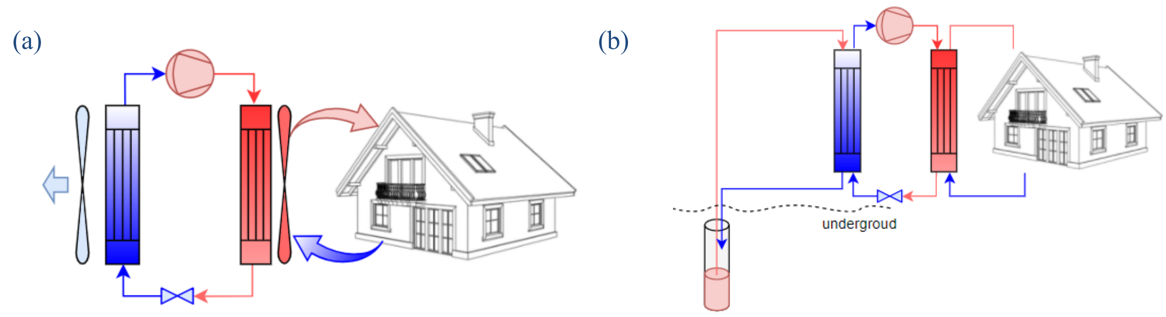

Figure 16. Heat pump system schematics of (a) ASHPs (R-410A or CO_2) and (b) water well integrated GSHPs (R-410A or CO_2).

These systems were designed to operate with indoor air temperatures of 20°C and varying outdoor or well water temperatures of 8.3°C, with performance parameters evaluated under these conditions (Table 2).

The heating capacities of the systems are maintained at 17.8 ± 0.2 kW. Notably, the CO_2 ASHP has the highest heating capacity at 18.0 kW. Under the design conditions, the R-410A ASHP and water well integrated R-410A GSHP outperformed their CO_2 counterparts, with COPs of 5.32 and 5.17, respectively, compared with COPs of 4.15 for the CO_2 ASHP and 3.96 for the water well integrated CO_2 GSHP. This suggests that CO_2 systems may achieve capacities comparable to those of R-410A systems but with lower efficiency.

Examining the compressor design, both R-410A systems have a higher displacement volume of 162 cm^3/rev compared with that of the CO_2 systems, which are designed with 61 cm^3/rev. Power consumption is notably higher for the CO_2 systems, with the water well integrated CO_2 GSHP consuming 5,041 W and the CO_2 ASHP consuming 3,818 W, compared with 2,949 W and 2,964 W for the R-410A systems. The higher power consumption in CO_2 systems is reflected in their higher pressure ratios and increased losses, with the water well integrated CO_2 GSHP experiencing the highest pressure ratio, 3.43.

The indoor units of all systems use fin-coil designs, but with varying heat transfer surface areas. The R-410A ASHP features the largest surface area at 1.77 m^2, whereas the CO_2 systems have smaller areas at 0.92 m^2. This design choice likely contributes to the differences in capacity and efficiency. The outdoor units show further differentiation, with the R-410A systems using fin-coil designs and the CO_2 GSHP systems using brazed plate HXs. The R-410A systems again demonstrate larger heat transfer surface areas and different pressure loss characteristics, particularly on the air and refrigerant sides.

In summary, the design of these HP systems illustrates a trade-off among capacity, efficiency, and component design, with R-410A systems generally offering higher COPs and lower power consumption but at the cost of larger components and surface areas. On the other hand, CO_2 systems provide comparable heating capacities with more compact designs but at the expense of higher power consumption and lower COPs. Table 5 summarizes the performance of each HP design.

Table 5. Heat pump design performance.

	R-410A ASHP	**Water well integrated R-410A GSHP**	**CO_2 ASHP**	**Water well integrated CO_2 GSHP**
Indoor air temperature (°C)	20			
Outdoor air temperature (°C)	8.3			
Water well temperature (°C)	N/A	8.3	N/A	8.3
System				
Heating capacity (kW)	17.8	17.9	18.0	17.6
Heating COP	5.32	5.17	4.15	3.96
Compressor				
Displacement volume (cm^3/rev)	162	162	61	61
Volumetric efficiency	0.60	0.60	0.55	0.55
Isentropic efficiency	0.68	0.68	0.68	0.68
Power consumption (W)	2,949	2,964	3,818	5,041
Pressure ratio	2.43	2.43	2.11	3.43
Loss (W)	236	237	305	403
Indoor unit				
	Fin-coil	Fin-coil	Fin-coil	Fin-coil
Heat transfer surface area (m^2)	1.77	1.32	0.92	0.92
Tube length (m)	0.45	0.45	0.43	0.43
Tube diameter (m)	0.01	0.01	0.01	0.01
Tube pitch (m)	0.025	0.025	0.025	0.025
Row pitch (m)	0.022	0.022	0.022	0.022
Fin frequency (fpi)	14	14	14	14
Capacity (kW)	17.6	17.6	17.6	17.6
Air side pressure loss (mm H_2O)	2.8	5.5	10.2	10.2
Refrigerant side pressure loss (kpa)	7.4	2.75	2.85	2.82
Face velocity (m/s)	1.25	1.68	2.43	2.44
Outdoor unit				
	Fin-coil	Brazed plate	Fin-coil	Brazed plate
Heat transfer surface area (m^2)	2.34	1.22	1.38	0.85
Tube length (m)	1.78	N/A	2.54	N/A
Tube diameter (m)	0.01	N/A	0.01	N/A
Tube pitch (m)	0.025	N/A	0.025	N/A
Row pitch (m)	0.022	N/A	0.022	N/A
Fin frequency (fpi)	22	N/A	22	N/A
Capacity (kW)	14.8	10.7	14.0	14.0

Air side pressure loss (mm H_2O)	2.1	N/A	1.07	N/A
Brine side pressure loss (kpa)	N/A	6.9		2.7
Refrigerant side pressure loss (kpa)	30		9.6	
Face velocity (m/s)	0.73		0.8	

4. ENERGY ANALYSIS

This section delves into the simulation and analysis of HP performance, focusing on both the efficiency of the systems and their energy consumption when applied to a typical residential building. The analysis encompasses the four previously discussed distinct HP systems and compares their performance against a baseline furnace heating system.

The primary objectives of this analysis are twofold: first, to evaluate the operational efficiency of each HP under various environmental conditions, assessing factors such as COP and heating capacity; and second, to quantify the energy consumption of these systems when used to maintain indoor comfort in a typical residential home in cold climates. By comparing the HPs with a conventional furnace system, this study aims to identify potential energy savings and environmental benefits. This comprehensive analysis will provide valuable insights into the practical implications of adopting advanced HP systems in residential buildings, guiding decisions on optimal heating solutions that balance efficiency, cost, and environmental impact.

4.1 SYSTEM PERFORMANCE

The simulation results provide a comparative analysis of the four different HP systems evaluated across a range of ambient temperatures from 47°C to −15°C, focusing on the COP and capacity ratio, as shown in Figure 17 and Figure 18, respectively.

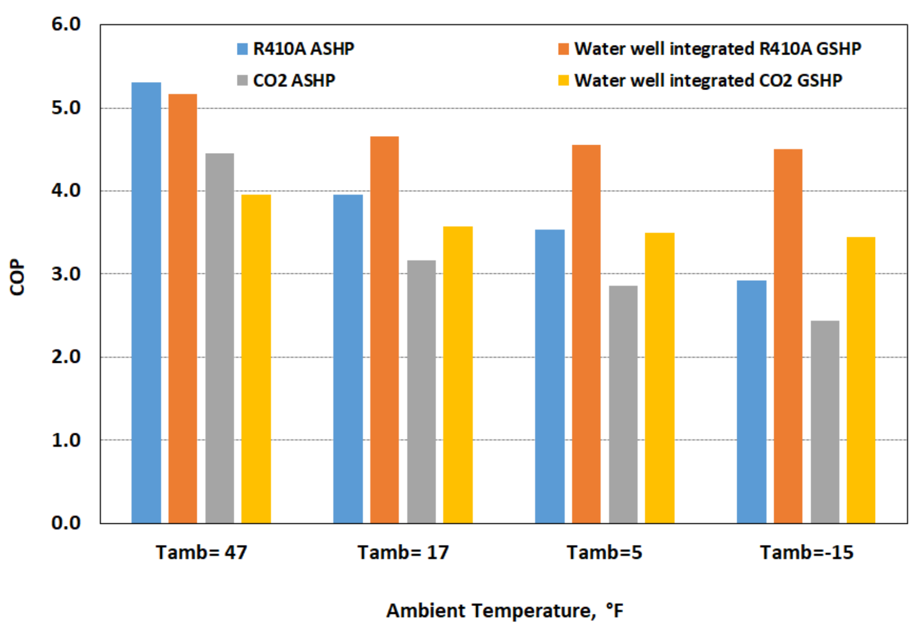

Figure 17.System efficiency (COP) of four heat pump system at design conditions

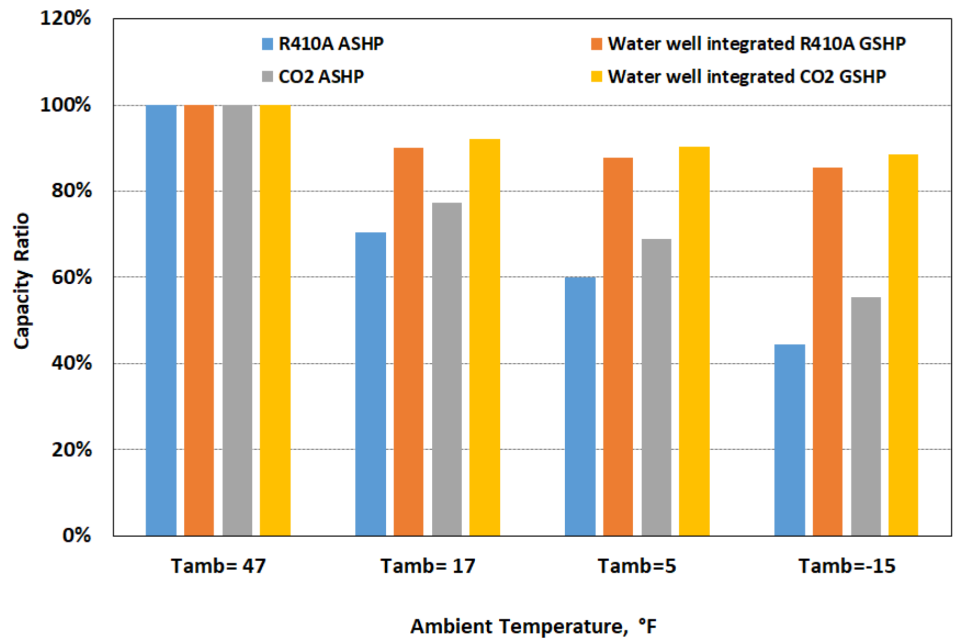

Figure 18. Capacity ratio.

The COP, which measures the efficiency of the HPs, revealed distinct trends among the systems. Both air-to-air systems exhibited a decrease in COP as the ambient temperature dropped, with the R-410A system starting at 5.31 at 47°C and falling to 2.92 at −15°C, and the CO_2 system dropping from 4.45 to 2.44 across the same temperature range. In contrast, the well water-to-air systems showed better performance in colder conditions because the well water temperature can be maintained in the range of 33°F–38°F even with ambient temperatures below 17°F. The R-410A system maintained a relatively high COP from 5.17 at 47°C to 4.5 at −15°C, whereas the CO_2 system had a more modest but stable drop from 3.96 to 3.44 across the same temperature range. This indicates that the well water-to-air systems are better suited for colder climates, particularly when using the R-410A system.

The capacity ratio, which reflects the system's ability to maintain its output relative to its rated capacity, further underscores the advantages of well water-to-air systems in colder environments. The air-to-air R-410A system's capacity ratio dropped significantly from 100% at 47°C to 44.5% at −15°C, and the air-to-air CO_2 system showed a similar decline from 100% to 55.3%. On the other hand, the well water-to-air systems demonstrated superior capacity retention, with the R-410A system decreasing from 100% to 85.5% and the CO_2 system falling from 100% to 88.6% as the temperature dropped.

In summary, the simulation results suggest that well water-to-air GSHPs, particularly those using CO_2, are more robust and maintain better performance in colder climates compared with air-to-air systems. The findings underscore the importance of selecting the appropriate HP technology based on environmental conditions, with a clear advantage for ground-source systems in cold climates.

4.2 ENERGY PROFILE

A 5 month heating season (October 1 to March 31) simulation was conducted for four HP systems and one baseline system. The results are shown in Figure 19. The figure shows that HP systems are more energy efficient than the baseline (furnace) system. Among the HP systems, the water-to-air systems are more energy efficient than the air-to-air systems. For the water-to-air systems, R-410A consumes less energy than the CO_2 type. However, from the refrigerant leaking perspective, the CO_2 HP is the recommended heating equipment. Because CO_2 leakage usually evaporates into the surrounding environment, this system offers energy efficiency without polluting the underground water, except in the case of a polluting compressor oil.

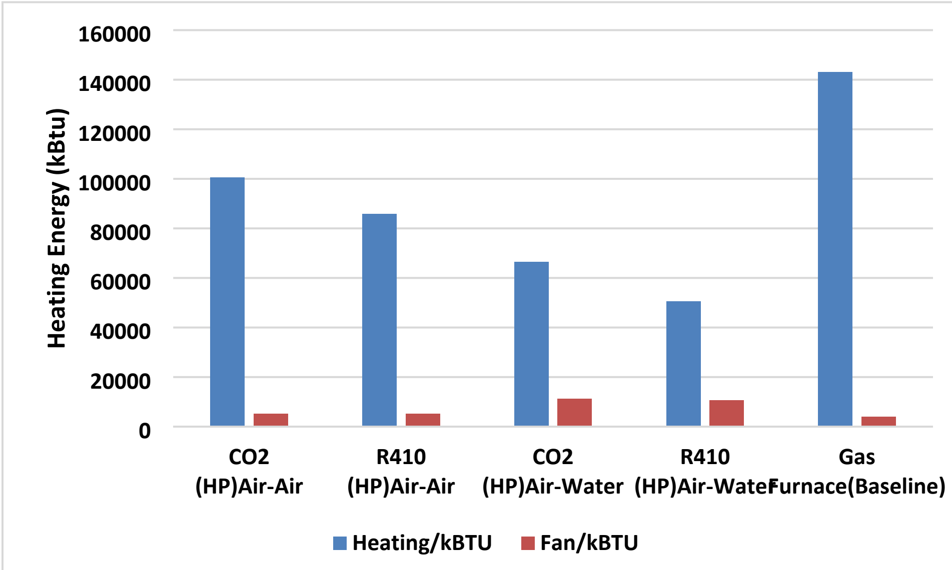

Figure 19. Heat pump energy consumption comparison.

4.3 ELECTRICITY PROFILE

The hourly profile for electricity demands is shown in Figure 20. In winter, the electricity demands seem to be similar among the four systems. During summer, electricity demand patterns for the four HP systems show that air-to-air CO_2 is the highest, air-to-air R-410A is second highest, water-to-air CO_2 is third, and water-to-air R-410A is the lowest.

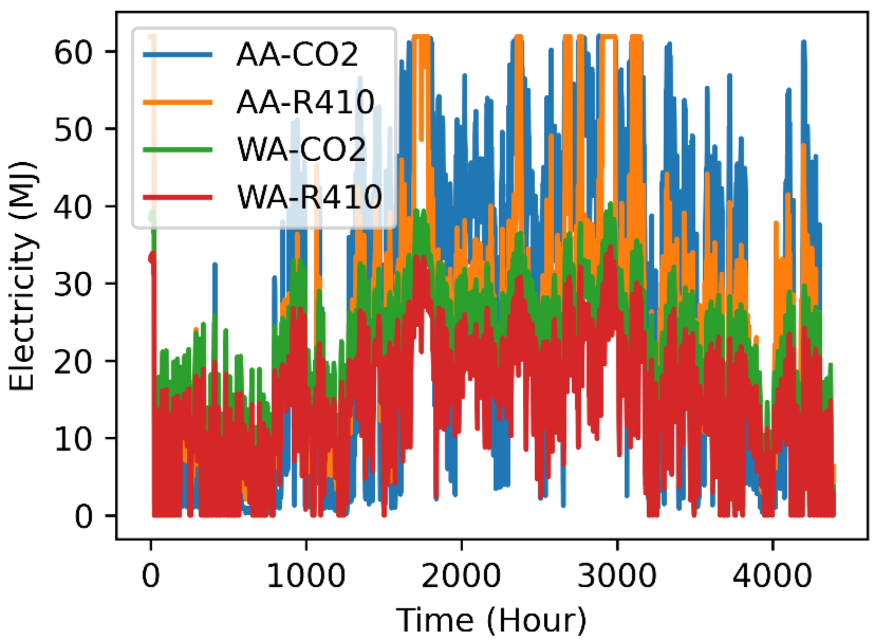

Figure 20. Heat pump electricity demands.

4.4 NATIONAL IMPACT HEATING ANALYSIS

Ten cities were selected across the United States for the national impact heating analysis. The selected cities and climate zones are shown in Table 6.

Table 6. National impact heating analysis selected cities and climates.

Climate	Cities
2A	Houston, TX
3A	Atlanta, GA
3B	Los Angeles, CA
4C	Seattle, WA
5A	Chicago, IL
5B	Boulder, CO
6A	Minneapolis, MN
6B	Helena, MT
7A	Duluth, MN
8A	Fairbanks, AK

The national impact heating analysis is summarized in Table 7. It includes the gas furnace (baseline) and water-to-air CO_2 HP systems. Heating energy, fan energy, total heating energy, and energy savings are provided for each selected city.

Table 7. National heating energy savings analysis

Climate	System	Heating (kBtu)	Fan (kBtu)	Total (kBtu)	Savings (%)
2A	Gas furnace	4,053	356	4,408	
2A	CO_2 HP	2,602	466	3,067	30.42%
3A	Gas furnace	42,890	1,972	44,862	
3A	CO_2 HP	23,518	3,945	27,462	38.78%
3B	Gas furnace	9,125	1,320	10,444	
3B	CO_2 HP	6,203	1,422	7,625	27.00%
4C	Gas furnace	60,284	2,050	62,335	
4C	CO_2 HP	39,254	6,390	45,644	26.78%
5A	Gas furnace	104,716	3,189	107,906	
5A	CO_2 HP	52,104	8,437	60,541	43.89%
5B	Gas furnace	73,091	2,807	75,899	
5B	CO_2 HP	42,420	7,267	49,687	34.54%
6A	Gas furnace	142,994	4,003	146,997	
6A	CO_2 HP	66,394	11,138	77,532	47.26%
6B	Gas furnace	98,468	3,253	101,720	
6B	CO_2 HP	59,292	9,972	69,265	31.91%
7A	Gas furnace	175,232	4,847	180,079	
7A	CO_2 HP	79,764	13,026	92,790	48.47%
8A	Gas furnace	228,290	5,914	234,204	
8A	CO_2 HP	113,445	16,009	129,454	44.73%

Figure 21 and Figure 22 show energy savings and energy consumption, respectively, for the selected 10 cities. Heating energy savings appears to be dynamic because of climate differences. Figure 21 shows that warmer climates (3B and 4C) have less heating energy savings, and colder climates (5A, 6A, 7A, and 8A) have higher heating energy savings. Climate 2A (Houston) consumes the least heating energy within the 10 selected cities because the outside air temperature in Houston is high enough to require use of less heating energy. Climate 3B (Los Angeles) consumes the second least heating energy owing to the mild weather. Climate 3A (Atlanta) consumes the third least heating energy.

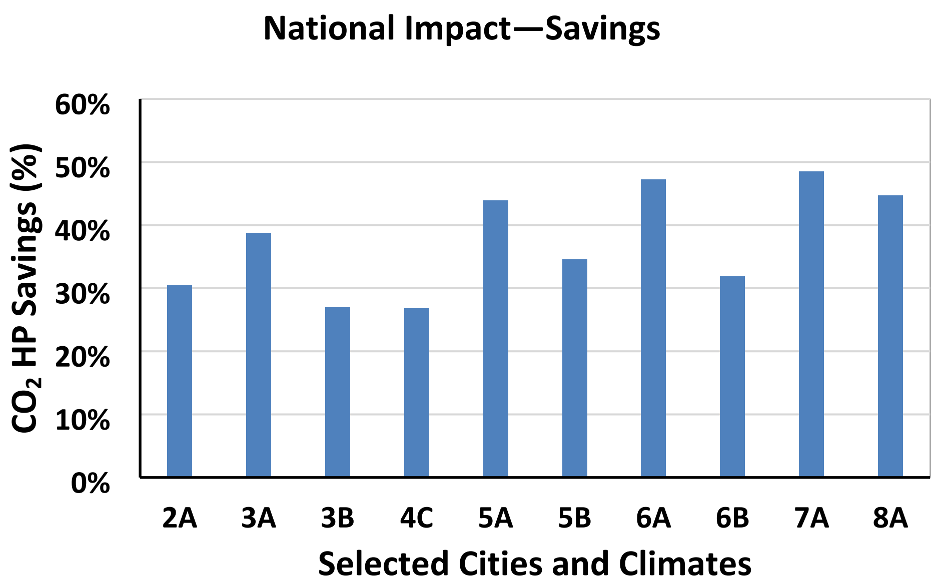

Figure 21. Energy savings for national impact heating analysis in selected cities (listed by each city's climate zone).

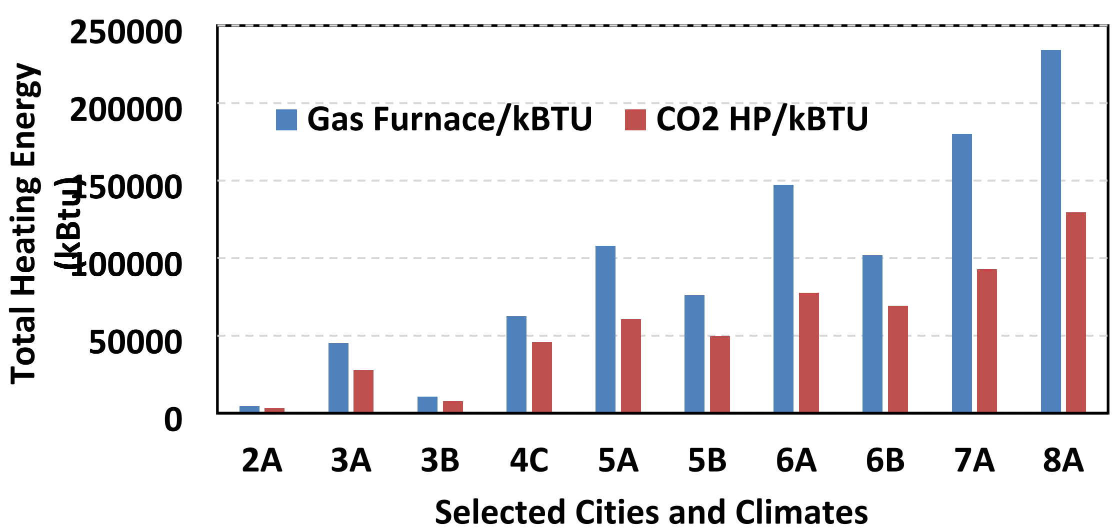

Figure 22. Energy consumption for national impact heating analysis in selected cities (listed by each city's climate zone).

5. CONCLUSION

This study conducted preliminary scoping research to assess the technical, environmental, and economic feasibility of using CO_2-based WSHPs as geothermal HPs sharing domestic water wells and to provide guidance for the potential deployment of this technology. The simulations were conducted for four different GSHP systems: air-to-air CO_2, water-to-air CO_2, air-to-air R-410A, and water-to-air R-410A. The conclusions are as follows:

- Considering system deployment, integrating GSHP technology with existing water wells in cold climates could offer a promising heating solution.

- The analysis of various types of water wells shows that artesian, bedrock, drilled, and collector wells are suitable for GSHP integrations because they can provide high flow rates, good water quality, stable temperatures, and low environmental impacts.

- This study analyzed four HPs: air-to-air R-410A, air-to-water R-410A, air-to-air CO_2, and air-to-water CO_2. The R-410A systems seem to have better efficiency than those of the CO_2 systems. However, from the contamination free and energy security perspective, CO_2 systems are more suitable for heating purposes.

- The HP equipment modeling analysis shows water-to-air HPs with CO_2 as refrigerant are more robust and maintain better performance in colder climates compared with air-to-air HP systems.

- The integrated energy system analysis with a residential model was conducted for a 5-month heating season. Results agree well with the equipment model analysis. Water-to-air systems seem to have higher COPs than those of air-to-air systems. The CO_2 refrigerant is more environmentally friendly, and it will not pollute the well water if a refrigerant leak occurs.

- Simulated power demand profiles for the four systems indicate that the air-to-air CO_2 HP is highest, the air-to-air R-410A HP is second highest, the water-to-air CO_2 HP is third, and the water-to-air R-410A HP is lowest. This indicates that the water-to-air CO_2 HP is a good choice.

- The national impact heating analysis was conducted for 10 cities across US climate zones. Results show the CO_2 HP is superior to the furnace. Heating demands are higher in cold climates than those in mild or warmer climates. Thus, installation of CO_2 HPs for cold climate heating needs is recommended.

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