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Comparison of Primary Energy Consumption of Vapor and Non-Vapor Compression Natural Refrigerant Heat Pumps for Domestic Hot Water Applications

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Abstract

The continued phase out of high global warming potential (GWP) refrigerants and the desire to reduce primary energy consumption in domestic water heating has spurred development in electric transcritical carbon dioxide (CO_2) vapor compression heat pumps and natural gas-fired ammonia-water absorption heat pumps. While both technologies are potentially more efficient than electric resistance or conventional gas-fired water heaters, there has not been an extensive comparison of the two technologies for domestic water heating applications. The objective of the present study is to assess the baseline performance of the two technologies on a primary energy consumption, emission and operation cost basis. Thermodynamic state point simulations of representative gas-fired, single-effect ammonia-water absorption system and an electric, transcritical CO_2 heat pump with suction line heat exchanger are developed. The models are then used to assess the sensitivity of system performance to inlet water temperature and ambient conditions. The results show that choosing an "optimal" system is highly dependent on the local electricity mix and fuel costs, and the relative importance of operational cost versus greenhouse gas emissions to the user. At the current U.S. average electricity mix and fuel costs, the absorption heat pump emits less CO_2 and is less expensive to operate. However, in locales with less expensive electricity and/or less carbon producing electricity generation, the electric heat pumps are at an advantage.

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Keywords: domestic hot water; natural refrigerants; system simulation; absorption; transcritical

1. Introduction

Water heating accounts for 17.7% (1.8 quads) of residential energy consumption in the United States [1]. By far, most water heating is accomplished with gas-fired or electrical resistance tank or tankless "on demand" water heaters. These systems are generally low cost, easy to install and have become very efficient from a first law perspective. However from a second law perspective, the conversion of electrical work or the combustion of energy dense fuel to provide heating of water up to ~60°C is not ideal. Heat pump water heaters (HPWH) are thermodynamically more advantageous, but have seen limited penetration into residential market due to higher capital and installation cost, and the relatively low cost of energy in the U.S [2]. However, as countries respond to various international agreements on limiting greenhouse gas emissions, HPWHs have become an attractive

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technology for reducing primary energy consumption and carbon dioxide (CO_2) emissions. Furthermore, as restriction of refrigerants with high global warming potential increases [3,4], HPWHs with low GWP working fluids are being sought as an alternative. Two technologies using natural refrigerants have gained traction in the residential sector. The first is electrically driven, vapor compression heat pump using carbon dioxide (CO_2) as the working fluid. The second is a gas-fired, ammonia-water absorption heat pump system. In both systems the working fluids have no ozone depletion potential (ODP), and a GWP of 1 for CO_2, and 0 for ammonia-water. Since the two systems operate using different cycles and different fuel sources, it can be difficult to fairly compare the performance of the two. Coefficients of performance (COP) of CO_2 heat pumps based on an electrical impact are generally > 3, while ammonia-water COPs based on gas input are typically from 1 to 2. However, it isn't useful to directly compare these numbers. In the present study, we develop simulation models of an absorption and vapor compression heat pump using natural refrigerants, and compare the performance on a basis of primary COP, CO_2 emission and operating cost.

2. Prior Work

Starting in the mid-1990s, heat pumps using CO_2 as a working fluid have experienced a resurgence [5]. Unlike a conventional heat pump, CO_2 heat pumps operate as a transcritical cycle, that is, the high-side of the system is above the vapor/liquid dome. Thus, rather than a two-phase condensation process, heat is rejected via a non-isothermal gas cooling process. This non-isothermal glide is well suited to water-heating applications, where a high temperature lift is required [6,7]. Further, by matching the thermal capacitance rates, extremely compact gas coolers can be developed, reducing system cost [8,9]. Based on these advantages, CO_2 heat pumps have been commercialized, most notably the Japanese "Eco Cute" water heating system, which has over 4 million units installed. The Japanese units generally take advantage of time varying electricity rates to generate and store hot water at night, at temperatures higher (T = 90°C) than would be utilized in the household. Carbon dioxide heat pumps have also been the subject of much academic interest, including exploration of the possibility of simultaneously production of hot water and space heating or cooling, with reviews and issues discussed in [10–12]. Despite the large body of work, CO_2 heat pumps have achieved very limited penetration in the U.S. residential market.

At the same time, there has been a renewed interest in absorption heat pumps. Investigations have ranged from fundamental studies to the development of systems for a range of applications. In the past 10 years, there has been significant developments relating to residential space (less than 25 kW) and water (less than 5kW) heating systems. The Heat4U program in the EU funded the development of an 18 kW absorption heat pump targeted for existing European residential buildings that had consortium of 14 partners that included Robur and Bosch [13]. The U.S. Department of Energy funded the development of a low-cost 23.5 kW absorption heat pump for building space heating [14]. Both projects highlighted the potential energy savings in residential applications of gas absorption heat pumps when compared to conventional furnaces and boilers. Garrabrant *et al.* [15,16] investigated and has continued the development of a residential capacity (3 kW) absorption heat pump water heater. This system is similar in look and function to that of commercially available vapor-compression heat pump water heaters and offers increased performance when compared to conventional gas-fired appliances. Results of field testing performed with the system developed by Garrabrant *et al.* was presented by Glanville [17] and showed that the systems could achieve greater than 50% energy savings compared to conventional gas water heaters.

3. Model Development

To compare the performance of the low GWP hot water heat pumps, three different simulation models were developed for a set of equivalent global system parameters (Table 1). All models were developed using the *Engineering Equation Solver* (EES) platform [18]. To facilitate comparison, all three models assumed systems with air-coupled evaporators and a hydronically coupled condenser/gas cooler. The absorption system also had a hydronically coupled absorber and flue gas heat exchanger. Models were developed for an absorption system with a constant hydronic circulation rate, a CO_2 heat pump with a constant hydronic circulation rate, and a CO_2 system with a variable hydronic circulation rate to maintain a 60°C water outlet temperature. To simplify the analysis in this investigation, we are neglecting the effect of a storage tank. It is assumed that the heated hydronic loop is coupled through a secondary heat exchanger to the domestic water, and that the hydronic loop outlet temperature is a surrogate for the domestic hot water temperature in our analysis. The developed baseline models were used to specify the heat exchanger overall conductance values (UA), pump power, and compressor power required to satisfy the desired heat duty and temperature lift from Table 1. The models were then used to evaluate and compare the performance of the systems at off-design ambient temperature and water inlet temperatures.

All three systems were designed to provide a nominal 2.9 kW of heating at an ambient temperature of 20°C. For the constant hydronic circulation systems, a temperature lift of 6.8 K with an inlet of 32.2°C was assumed for the baseline. These values were selected based on prior work performed by Garrabrant *et al.* [15] and commercially available heat pump water heating systems. The baseline water inlet temperature was selected to allow for heat exchanger UA values that were balanced for the range of water inlet temperatures experienced in a full heating cycle (15-60°C) without being oversized. The 20°C ambient was selected because that is the required temperature for the US Department of Energy UEF test procedure. The coupling water flow rate-temperature lift were selected based on coupling fluid flow rate optimization work performed by Garrabrant *et al.* [15]. For the variable flow system, a lift of 45 K with a water inlet of 15°C was specified. The constant flow systems are representative of a tank mounted heat pump water heater with recirculation pump, while the variable flow CO_2 system is representative of an Eco Cute type system with variable water pump control to maintain a constant hot water outlet temperature. Other global assumptions for the system models are as follows:

- Steady state operation
- Isenthalpic throttling processes
- Adiabatic heat exchangers (*i.e*., no heat loss/gain from ambient)
- Isobaric heat transfer/absorption/desorption processes

3.1. Ammonia-water heat pump model

The gas-fired heat pump considered was a single-effect ammonia-water absorption heat pump water heater. Figure 1 is a cycle schematic of the system. The desorber is direct natural gas-fired and the heat of combustion is the main energy input to the system. Refrigerant is generated in the desorber and flows through the rectifier where water is selectively condensed before flowing through the typical refrigerant loop of a vapor-compression heat pump (condenser, suction line heat exchanger/refrigerant heat exchanger, expansion device and evaporator). Heat is exchanged with coupling water during the condensation of the refrigerant in the condenser. Heat is exchanged with ambient air during the evaporation of the refrigerant in the evaporator. Refrigerant exiting the refrigerant loop mixes with the dilute (ammonia weak) solution before entering the absorber where the refrigerant vapor is absorbed by the solution in an exothermic process. The heat of absorption is exchanged with the coupling water. The concentrated (ammonia rich) solution exiting the absorber is pumped to the high pressure side of the system where it recuperates heat in the rectifier and solution heat exchanger before entering the desorber. Dilute solution exiting the desorber exchanges heat with the concentrated solution and the flows through an expansion device from the high to low pressure side of the system. The solution then mixes with the refrigerant from the refrigerant loop. The combusted gas products exiting the desorber exchange heat with the coupling water in addition to the heat exchanged by the absorber and condenser.

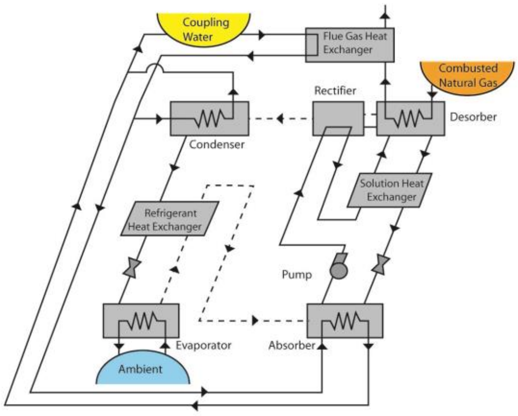

Fig. 1. Schematic of Ammonia-Water Absorption Heat Pump Water Heater

The modeling was performed in steps similar to those outlined by Herold *et al.* [19]. Mass, species and energy conservation equations were applied for each component of the model and three independent properties were determined to fix ammonia-water state points at the inlet and outlet of each component. Vapor exiting the desorber and rectifier were assumed to be at saturation (quality of 1). Solution exiting the absorber, condenser and desorber, and reflux exiting the rectifier were assumed to be at saturation (quality of 0). Closest approach temperatures (Table 1) were initially assumed. Solution and refrigerant heat exchanger effectiveness of 95% were initially assumed. These assumed values allowed for the calculation of overall heat conductance values (UA) for each component.

The coupling fluid flow rate was assumed to be 0.101 kg/s at design inlet water and ambient temperatures of 32.2°C and 20°C, respectively. The coupling fluid flow was split between the absorber and condenser with 30% of the flow going to the condenser. At the baseline conditions, the higher heating value based gas supply rate was assumed to 1.95 kW and the auxiliary support systems were estimated to require 0.044 kW of electricity during system operation.

The overall conductance UA values were then locked and performance of the system was investigated over a range of ambient (-20 to 40°C) and coupling water (15 to 60°C) temperatures. Important baseline values are summarized in Table 2 at the conclusion of this section.

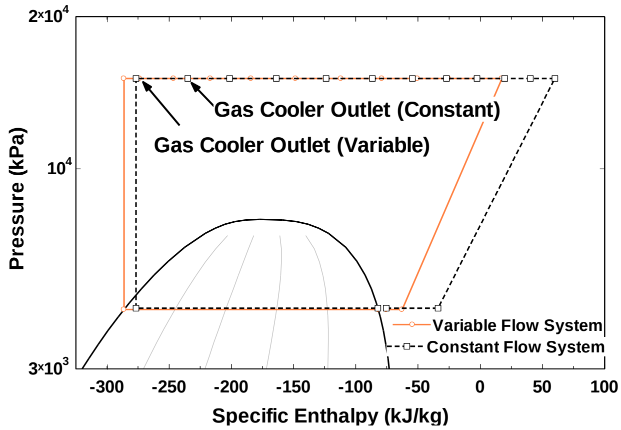

Fig. 3. Pressure-enthalpy Diagram of Baseline CO_2 Heat Pump Cycles

3.2. CO_2 heat pump model

The CO_2 system under consideration is shown in Figure 2. The cycle consists of an air-coupled evaporator, hydronic coupled gas cooler and internal recuperative suction line heat exchanger. Water is heated at either a constant or variable flow rate through the gas cooler. Energy is transferred into the system through an air-coupled evaporator operating at a constant volumetric flow rate of 0.354 m^3/s (750 ft^3/min). The CO_2 baseline simulation models for the constant and variable hydronic flow were developed similar to the approach of Goodman *et al.* [7]. In each component, energy and mass balances were formulated. A constant compressor isentropic efficiency of 80% and a suction line heat exchanger effectiveness of 80% were assumed. The CO_2 gas cooler temperature was fixed according to the specified closest approach temperature of 4 K. Unlike a subcritical heat pump, the temperature and pressure of CO_2 in the gas cooler are independent of one another. Thus, another assumption is required to provide closure to the system. A high-side pressure of 172.4 bar (2500 psia) was assumed for all

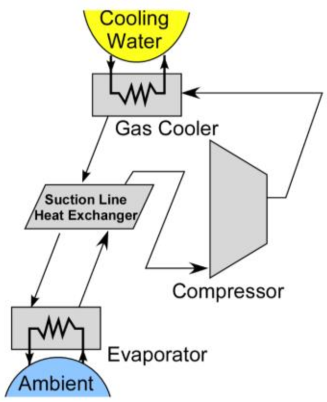

Fig. 2. Schematic of CO_2 Vapor Compression Heat Pump Water Heater

simulations in the present study. While prior work has shown that an optimal high-side pressure can be determined for different CO_2 heat pump operating conditions [7], an optimization scheme was outside of the scope of the present work. These assumptions with the baseline conditions in Table 1 allowed the thermodynamic state point at the inlet and outlet of each component within the system to be calculated. A *P-h* diagram of the constant and variable flow systems at the baseline conditions is shown in Figure 3.

3.3. Baseline System Design

A summary of the important heat exchanger UA values, and electrical and gas power input for the ammonia-water and CO_2 systems is provided in Table 2. As the heat transfer coefficients in the ammonia-water and CO_2 systems are substantially different, it is difficult draw substantive conclusions about the relative size of the heat exchangers by directly compare UA values alone. However, it is clear from a comparison of the CO_2 systems that the much larger temperature lift required in the variable flow system results in a larger required heat exchanger (*i.e*., the gas cooler UA is greater). Further, it is also immediately evident that the CO_2 systems require much more electrical power input at baseline conditions. The effect of this on primary energy consumption and greenhouse gas emissions will be explored below.

4. Results and Discussion

Using the baseline models developed in Section 3, a parametric investigation of the effect of changes in ambient temperature and water inlet temperature on primary system COP, CO_2 emissions and operational cost were considered. For this study, we considered a variation in ambient temperature from -20°C to 40°C. We also considered water inlet temperatures corresponding of (a) 15°C representative of a full tank recharge (b) 35°C corresponding to a response to a partial tank draw and (c) 50°C corresponding to recovery from standby losses. The final water temperature in all cases was 60°C, yielding lifts of 45 K, 25 K and 10 K for cases (a) – (c).

4.1. Primary COP comparison

To provide a meaningful comparison between gas-fired and electrically driven systems, a primary COP based on *source energy* must be considered. To convert from the calculated site to source energy, all electrical power inputs are multiplied by a factor of 3.14, while gas inputs are multiplied by 1.05, as shown in Eqs. (1) and (2) for the absorption and electric heat pumps respective COPs. The factors provide a conversion between site-to-source energy for the United States [20]. That is, they represent the total of the energy consumed by the heat pump, and the energy required to generate and deliver that energy to the site. For electricity this includes losses due to generation and transmission, while for natural gas this includes storage, transport and delivery of fuel. It should be noted that the gas based calculations assume the use of the higher heating value of natural gas.

$$COP_{\mathrm{NH3/H2O}} = \frac{\dot{Q}_{water}}{3.14\cdot\left(\dot{W}_{fan}+\dot{W}_{pump}+\dot{W}_{blower}\right)+1.05\cdot\left(\dot{Q}_{gas}\right)} \tag{1}$$

$$COP_{\mathrm{CO2}} = \frac{\dot{Q}_{water}}{3.14\cdot\left(\dot{W}_{fan}+\dot{W}_{comp}\right)} \tag{2}$$

For the constant flow rate absorption and CO_2 systems, the average COP to heat the water from the initial temperature to the final temperature of 60°C was calculated by averaging the performance of each system at the initial water inlet temperature with performance of the system when providing an outlet water temperature of 60°C. This approach was used to represent the overall heating performance of the constant flow rate systems for each lift

Table 2. Baseline System Results

Parameter	Absorption Constant Flow	CO_2 Constant Flow	CO_2 Variable Flow
UA Condenser/Gas Cooler	0.1765 kW/K	0.0793 kW/K	0.1117 kW/K
UA Evaporator	0.3565 kW/K	0.2168 kW/K	0.2273 kW/K
$\dot{Q}_{evap}$	1.211 kW	1.978 kW	2.10 kW
$\dot{Q}_{gas}$	1.948 kW	-	-
$\dot{W}_{fan}$	0.021 kW	0.031 kW	0.031 kW
$\dot{W}_{pump}$	0.012 kW	-	-
$\dot{W}_{blower}$	0.011 kW	-	-
$\dot{W}_{comp}$	-	0.922 kW	0.799 kW

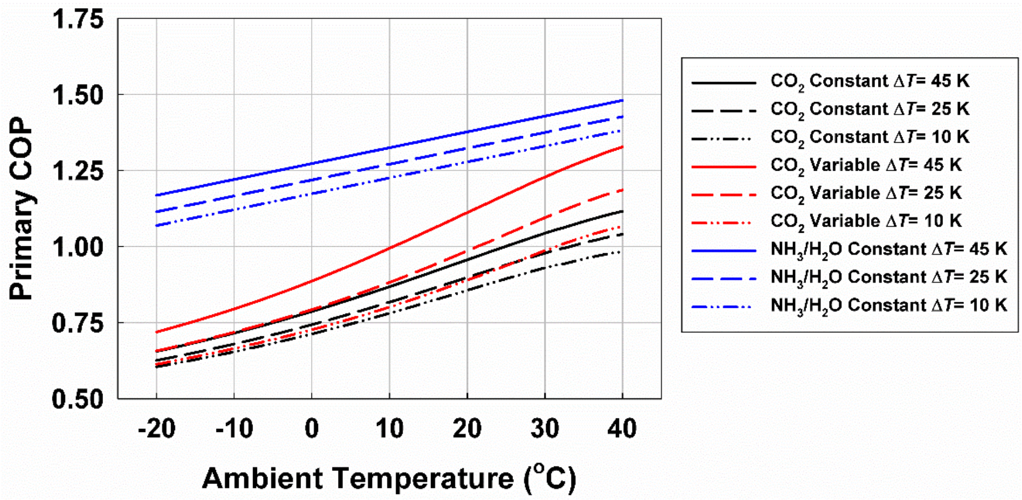

Fig. 4. Comparison of Primary COP at Varying Ambient Temperatures and Inlet Water Flow Rates for Gas-fired and Electric HPWH

because it approximates performance of the water being heated in a continuous process where the outlet water temperature in constantly increasing to 60°C. For the variable flow system, a single COP value for each water inlet temperature was calculated, assuming a fixed outlet temperature of 60°C. The results are shown in Figure 4.

Figure 4 shows that the gas-fired absorption heat pump has a higher primary COP for all conditions than either of the electric CO_2 heat pumps by approximately 35%, 44% and 52% for lifts of 10 K, 25 K and 45 K, respectively (compared to variable flow CO_2 heat pump). The absorption heat pump shows particularly superior performance at low ambient temperatures. As ambient decreases, the evaporator heat duty for both heat pump types decrease, and they approach the limit of 100% gas-fired and 100% electric resistance water heating systems.

Of the two CO_2 configurations, the variable flow configuration had higher COP than the constant flow configuration over the range of conditions. As can be concluded from the *P-h* diagram in Figure 3, CO_2 heat pumps are particularly sensitive to the gas cooler water inlet temperature. As the water inlet temperature increases, the gas cooler CO_2 outlet temperature also increases, which leads to a direct decrease in the available cooling capacity (*e.g*., energy harvested from ambient), for an equivalent compressor power. This leads to a sharp decrease in COP. Thus, it is desirable to minimize the gas cooler water inlet temperature as much as possible, as in the variable speed case where recirculation with progressively hotter water is not necessary. The variable flow system is likely to experience increased inlet water temperatures as it nears completion of its heating cycle. The impact of this will reduce overall performance but is beyond the scope of this study.

The results in Figure 4 are based on the average source-site ratios conversion factors for the United States as reported by the U.S. Environmental Protection Agency [20]. For the U.S., the source-site ratio of 3.14 for electric power is based on an electric production mix of 66% fossil fuel, 21% nuclear and 13% renewable (including hydropower). Figure 5 shows the source-site ratio for electricity that would be required for the gas absorption heat pump and electric CO_2 heat pump (variable flow) to have an equivalent primary COP. This analysis assumes a constant 1.05 source-site ratio for natural gas, and also factors in the small electrical consumption of the absorption heat pumps. Figure 5 also shows that while the electric CO_2 heat pump has worse performance on a primary COP basis in the U.S., it would fare better in Canada, which has a source-site ratio of 2.05 [20]. In fact, at ambient greater than 10°C the electric (with variable flow) would outperform the gas-fired system for all temperature lifts in Canada. For a CO_2 heat pump with constant hydronic flow, the transition ambient increases to approximately 15°C. Thus, as the electricity grid mix changes, the competitiveness of electric based systems on a primary COP basis increases with the penetration of renewables.

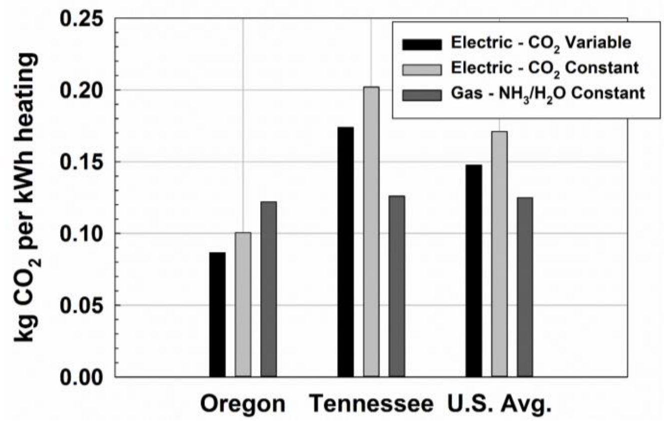

Fig. 6. Comparison of CO_2 Emissions of Gas-fired and Electric HPWHs Ambient of 20°C and a Temperature Lift of 45 K.

Fig. 5. Required Source-site Ratio for the Primary COP of the Gas-fired Absorption Heat Pump and Electric CO_2 Heat Pump with (a) Variable Hydronic Flow and (b) Constant Hydronic Flow to be Equal as a Function of Ambient Temperature and Temperature Lift.

4.2. Emission and economic comparison

Like primary COP, the greenhouse gas emissions and cost per unit of water heating is expected to vary with location. Figures 6 and 7 show the CO_2 emissions and fuel/electric cost per hour of operation, respectively, at an ambient of 20°C and a temperature lift of 45 K. Emissions (on a kg of CO_2 production per kWh of water heating) [21] and operational cost are evaluated for the U.S. states of Oregon, and Tennessee, and for the U.S. national average [22].

Oregon is located in the Pacific Northwest of the United States and obtains nearly 73% of its electricity from renewable resources [23], primarily hydropower, yielding 0.302 kg CO_2/kWh of electricity. Tennessee, located in the American Southeast, has obtains 60% of its electricity from coal and natural gas [24], resulting in an average of 0.607 kg CO_2/kWh. Thus, the electric based systems have about half the greenhouse gas emission in Oregon compared to Tennessee. The CO_2 generated per kWh of natural gas is estimated as 0.181 kg/kWh. Thus, in Oregon, the electric heat pumps have the best performance on an emissions basis, while in Tennessee and in the U.S. on average, the gas-fired absorption heat pump emits less CO_2 per unit of water heating. As was noted above, as the electrical mix continues to undergo decarbonization, the emissions of the electric based systems will drop.

Figure 7 shows the cost of electricity and natural gas per kWh of water heating for Oregon, Tennessee, and the U.S. average at an ambient of 20°C and a temperature lift of 45 K. The cost of electricity in Oregon and Tennessee (10.35 and 10.05 cents/kWh [22], respectively) are both below the U.S. average of 12.3 cents/kWh. At the same time, a recent increase in residential natural gas prices have pushed prices to an U.S. average of \$17.62 per 1000 ft^3 as of August 2016 [25]. This results in the absorption heat pump being more expensive to operate for an equivalent heating duty in Oregon and Tennessee. At the U.S. average electricity and gas prices, the absorption heat pump is cheaper to operate compared to the constant flow CO_2 heat pump. However, the variable speed system is slightly cheaper than the absorption heat pump.

4.3. Other HPWH considerations

This evaluation has highlighted the performance and potential of electric and gas fired heat pumps using natural refrigerants. Both offer substantial energy, operating cost and CO_2 emissions reductions when compared to conventional electric and gas water heating systems. However, it should be noted that the performance results of these systems are idealized. Implementation of these systems and integration with a hot water storage tank adds another layer of complexity that was not explored as part of this study. How the heat pump system is coupled to the water storage tank can have a significant impact on performance (additional thermodynamic limitations, impact of mixing within the tank, and electrical use) and could cause results to shift slightly. Capital cost and potential for consumer adoption will also impact design of these systems.

Installation location and its impact on heat pump performance were investigated by varying the ambient temperature when establishing performance curves of each heat pump. Expanding this to a seasonal performance outside of the 20°C ambient design case would be a next step and allow for refined estimations of energy use, CO2 emissions and operating cost. The installation location of water heaters can vary significantly (climate region, conditioned or semi-conditioned space and other factors). Impact on the temperature of the installed space is another consideration. The absorption heat pump will provide cooling to the surroundings at a nominal rate of 1 kW while the vapor compression heat pump will nominally provide 2.5 kW of cooling. Depending on the installation and climate locations this could have a positive or negative impact on the surrounding space.

Consumer adoption is critical to the success of the heat pump water technologies. Water heating is a challenging market because units are only replaced when they fail which is typically 10 to 15 years and energy efficiency is not always a priority. Electric heat pump water heaters (synthetic and natural refrigerant) are commercially available but adoption has been below market expectation. Absorption heat pumps are only commercially available for larger capacity applications and market adoption of a water heating specific unit is uncertain. The continued advancement of cost-effective heat pump technologies in combination with higher efficiency regulations will improve the adoption of these technologies over the next 10 to 20 years.

5. Conclusions

In this study we developed thermodynamic simulation models for a gas-fired ammonia-water single effect absorption system and an electric CO_2 heat pump for domestic hot water production. The systems both had an air-coupled evaporator and hydronically coupled condenser/gas cooler. For the CO_2 system, we considered a variable and constant water flow configuration, while a constant water flow was assumed for the absorption system. Each system was sized to provide 2.9 kW of water heating at a baseline condition with the same set of global assumptions. The performance of the baseline systems were then parametrically investigated for a range of ambient temperature and water temperature lifts. Each system was compared on a basis of primary COP, CO_2 emissions and cost per equivalent heat duty.

The results indicate that deciding which system is "best" is highly dependent on the local electricity generation mix, and electricity and fuel costs. For the current U.S. electric generation mix, the gas-fired absorption heat pump had a higher primary COP than the CO_2 heat pumps for all ambient temperatures and water temperature lifts. However, if we consider the Canadian electric mix (with a lower source-to-site ratio), the electric heat pump became more competitive. Similarly, greenhouse gas emissions are highly dependent on the local mix. In the present study, we show that in the state of Oregon (with high percentage of low carbon power generation), the electric heat pumps emit less CO_2 per equivalent water heating duty. However in Tennessee (a high fossil fuel state) and the U.S. average, the gas-fired heat pump emitted less CO_2. Finally, we also see that the cost per unit of water heating is higher for the absorption heat pump in Oregon and Tennessee due to locally cheap electricity, but

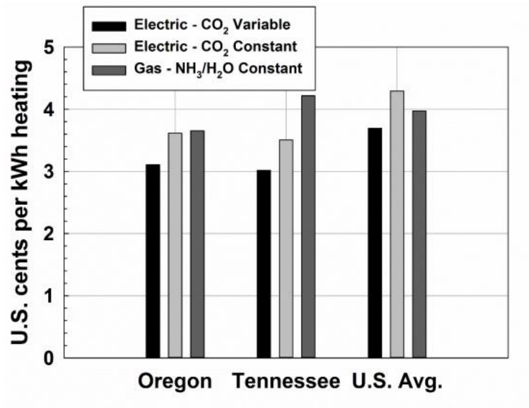

Fig. 7. Comparison of Cost of Heating of Gas-fired and Electric HPWHs for ambient of 20°C and a Temperature Lift of 45 K.

more cost effective for the U.S. on average. From a thermodynamic perspective, both heat pump technologies are preferable to pure gas-fired or electrical resistance heating. Furthermore, the use of natural refrigerants in both system is advantageous given the continued phase out of higher GWP synthetic refrigerants. The optimal system from a consumer's perspective will require consideration of capital, installation and local fuel costs. The optimal system from a greenhouse gas reduction policy perspective will require an understanding of the local electricity mix as well as nominal ambient temperatures. Finally, as the electricity grid in North America and globally continues to become more renewable, the electric based systems will become more attractive, unless there is a dramatic deviation between the cost of natural gas and electricity.

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