Development of Environmentally Benign Heat Pump Water Heaters for the US Market

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ABSTRACT

Improving energy efficiency in water heating applications is important to the nation's energy strategies. Water heating in residential and commercial buildings accounts for about 10% of U.S. buildings energy consumption. Heat pump water heating (HPWH) technology is a significant breakthrough in energy efficiency, as an alternative to electric resistance water heating. Heat pump technology has shown acceptable payback period with proper incentives and successful market penetration is emerging. However, current HPWH require the use of refrigerants with high Global Warming Potential (GWP). Furthermore, current system designs depend greatly on the backup resistance heaters when the ambient temperature is below freezing or when hot water demand increases. Finally, the performance of current HPWH technology degrades greatly as the water set point temperature exceeds 333 K.

This paper presents the potential for carbon dioxide, CO_2, as a natural, environmentally benign alternative refrigerant for HPWH technology. In this paper, we first describe the system design, implications and opportunities of operating a transcritical cycle. Next, a prototype CO_2 HPWH design featuring flexible component evaluation capability is described. The experimental setup and results are then illustrated followed by a brief discussion on the measured system performance. The paper ends with conclusions and recommendations for the development of CO_2 heat pump water heating technology suitable for the U.S. market.

Introduction

Heat Pump Water Heaters (HPWH's) provide substantial improvements in energy efficiency. Current commercially available models employ hydrofluorocarbon (HFC) refrigerants with high GWP and are limited to 333 K service temperatures. CO_2 heat pump water heating is envisioned as a technology providing a lower global warming potential (GWP) and zero ozone depletion potential (ODP) refrigerant option coupled with a potential for higher efficiency. CO_2 is a natural refrigerant without a flammability risk; however its operating parameters are different than refrigerants commonly used in heat pump applications. The CO_2 heat pump water heater cycle is transcritical, operating at much higher temperatures and pressures than conventional subcritical cycles. The transcritical cycle operation provides a large continuous temperature glide and can offer a higher service temperature with limited capacity loss. CO_2 heat pump systems are currently used in Asia and Europe for space and water heating. Development is needed to configure the technology for replacement and integration in the United States water heating market. The benign nature of the refrigerant and the high efficiency potential make CO_2 systems a likely successor to current water heating solutions. This paper discusses recent CO_2 heat pump water heater development specifically for the U.S. housing market.

Background

CO_2 HPWH technology was developed in Japan over the last decade under the brand name "EcoCute" or "Eco Cute". This type of water heater is becoming increasingly popular not only in Japan but also in other parts of the world due to the increased level of environmental awareness. The success of the Japanese CO_2 HPWH was driven by Japanese government subsidies of up to 50% of the unit cost. The Japanese government provided approximately 5 billion Yen per year in CO_2 HPWHs subsidies (Appliance Magazine 2005). Although the trademark "EcoCute" is registered under Kansai Electric Power, nine Japanese electric power companies have agreed to use it as a unified brand name.

Although "Eco Cute" water heaters cost twice as much as conventional water heating systems in Japan, their running cost is about one fifth of the standard models; and they offer 30% reduction in primary energy consumption and a 50% reduction in CO_2 emissions compared with conventional fuel fired boilers/water heaters (Kawashima 2005). These water heaters are appealing in the Japanese market context, particularly when the COPs range between 3 and 4.9, under Japanese industry rating conditions (as compared with electric resistance water heating being near 1.0 and gas heating being 0.8 including pilot light loss) and by system control schemes using lower cost, off-peak electricity. Due to the very large temperature glide for water heating, typically from 288 K up to 363 K, the continuous temperature glide of CO_2 in the supercritical region leads to minimization of entropy losses of the heat exchange process. Early experimental research in CO_2 HPWHs showed promising results (Hwang and Radermacher 1998, Neksa et al. 1998). Several cycle modifications have been introduced such as the work of Kim et al. 2005. Recent technology developments include the use of rotary compressors for higher reliability and the use of an internal heat exchanger to improve the cycle performance.

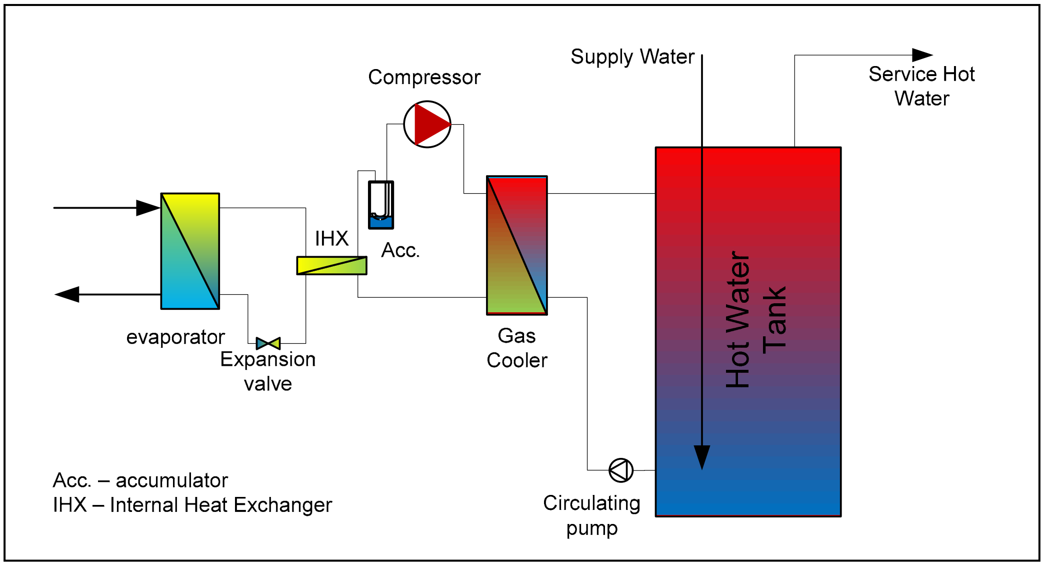
Transcritical CO_2 Vapor Compression Cycle

A typical CO_2 HPWH employs CO_2 as a refrigerant in a transcritical vapor compression cycle. CO_2 is first boiled off in an evaporator utilizing the heat from ambient air. The CO_2 gas is then compressed to supercritical pressures on the order of 10 MPa. Several types of compressors can be used, including a single rotary compressor with brushless DC motor (Maeyama and Takahashi (2007), a two-stage rotary compressor (Sanyo 2010) or a scroll compressor (Hashimoto 2006). The CO_2 gas is then cooled in a gas cooler cooled by domestic water. As the transcritical CO_2 gas cools down, water is heated up to 90°C. The schematic diagram of a baseline CO_2 HPWH is shown in Figure 1.

Thermodynamically, transcritical CO_2 cycles are inherently less efficient than conventional sub-critical vapor compression cycles at typical operating conditions. Current research is focused on improving the performance of CO_2 systems though system enhancements such as work-recovery components. Increasing the performance of compressors and heat exchangers is also vital to making CO_2 a viable heat pump technology. However, with the pending U.S. plan to phase down HFC refrigerants and the benefits of using a commonly available, non-toxic, non-flammable refrigerant, CO_2 systems remain very compelling for future applications.

Research insights and new knowledge on CO_2 cycles and components over the last decade have demonstrated that creative design solutions can, in fact, yield a dramatic performance improvement by looking beyond "conventional wisdom" about the perceived weaknesses of CO_2 compared to fluorocarbon technology's cycle requirements. Instead, the "learning curve" has focused on exploiting and utilizing the unique characteristics (strengths & opportunities) the alternative technology path offers. In the case of CO_2, these characteristics include certain thermophysical (thermodynamic and transport) properties, which increase heat transfer and boost compressor efficiency. Its high heat rejection pressure, a disadvantage for air conditioning, is a potential advantage for heat pumps, especially water heaters.

Figure 1. Schematic of Simple CO_2 HPWH



There are ways of approaching ideal cycle efficiency by exploiting the unique characteristics of CO_2, such as the slope of its vapor-pressure curve, boiling behavior near the critical point, high capacity at low temperatures, and its supercritical temperature glide. Other approaches involve ways of altering the transcritical cycle to increase the ideal efficiency, for example, through the use of expanders, liquid-to-suction line internal heat exchangers, and multistage compressors. The inherent high pressure in transcritical CO_2 cycles requires special equipment design, yet it offers many advantages over other refrigerants. The high pressure results in high gas density, which allows for a far greater refrigerating effect from a given compressor. Furthermore, the supercritical cooling of CO_2 in the gas cooler results in a temperature glide that can be matched to the water side temperature glide to improve the gas cooler effectiveness.

CO_2 HPWH Design for the U.S. Market

A CO_2 HPWH was designed as a drop-in replacement for HFC-based HPWHs available for the U.S. residential market. The system was designed with a 0.19 m^3 (50 gallon) water storage tank, energy factor greater than 2.0 and first hour rating greater than 50 (Energy Star criteria). The heat pump system was designed to provide a heating capacity of approximately 2.2 kW with 4.5 kW of backup electric resistance heaters. This is a significant difference to the "EcoCute" models which rely entirely on the heat pump to provide all heating needs, with source heat outside of conditioned space. Furthermore, the design required an incremental price premium over current HPWH technology. Hence, a fixed speed compressor, fixed speed pump and simple expansion device were used to assess performance at a minimal cost premium over existing HPWH technology in the US market. The prototype CO_2 HPWH was designed with a high degree of system flexibility and for the balance of systems required for experimental testing. The four main components of the prototype CO_2 HPWH are: compressor, gas cooler, expansion device and evaporator. The prototype was designed to test several CO_2 HPWH configurations. It was equipped with two gas coolers: a wrap-around gas cooler and an external gas cooler. The wrap around gas cooler is similar to state-of-the-art HPWHs used in the US and was designed to promote temperature stratification in the tank. The external gas cooler required the use of a water pump and might result in water scaling and reliability issues on the long run. In addition, the prototype was designed to either include or exclude an internal suction-gas heat exchanger.

The prototype schematic is illustrated in Figure 2. A three-way valve after the oil separator was used to switch the operation between the wrap-around gas cooler and the external gas cooler. Another three-way valve was used on the gas cooler outlet side to either enable or disable the internal gas-suction heat exchanger. Furthermore, the prototype was equipped with a wind tunnel to evaluate the evaporator's airside performance and provide proper control and measurement of the air flow rate. This wind tunnel is not required for the field-ready prototypes. Finally, a custom made accumulator was designed with a throttle valve at the bottom to control the liquid return to the compressor suction line. Figure 3 shows a picture of the fabricated prototype.

Figure 2. Detailed Prototype Schematic

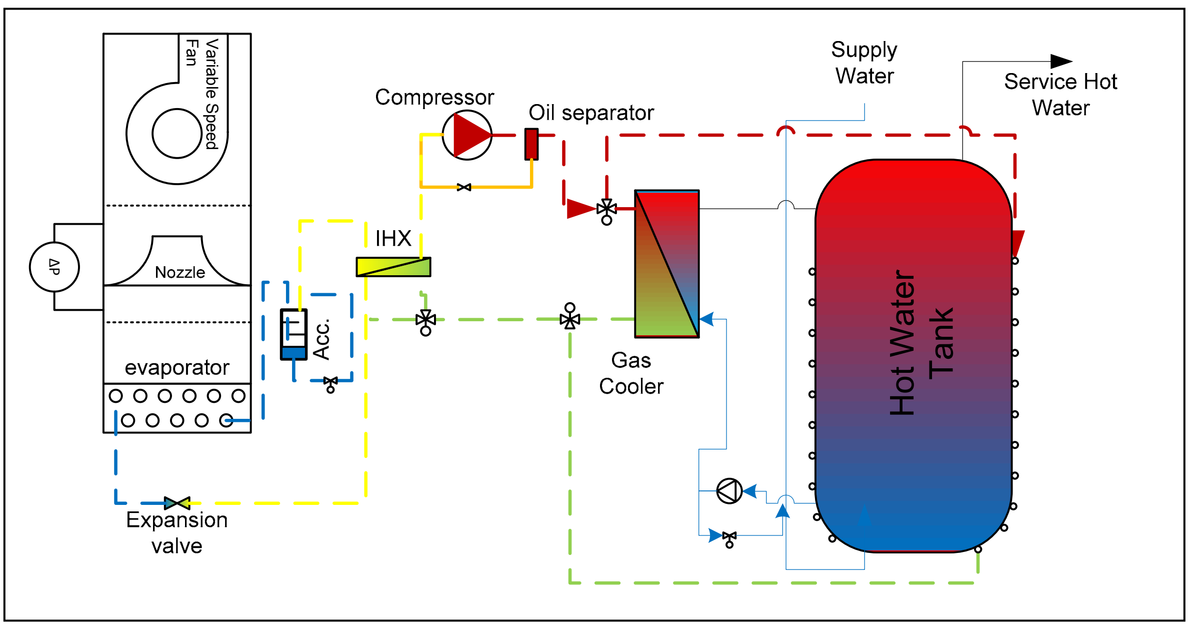
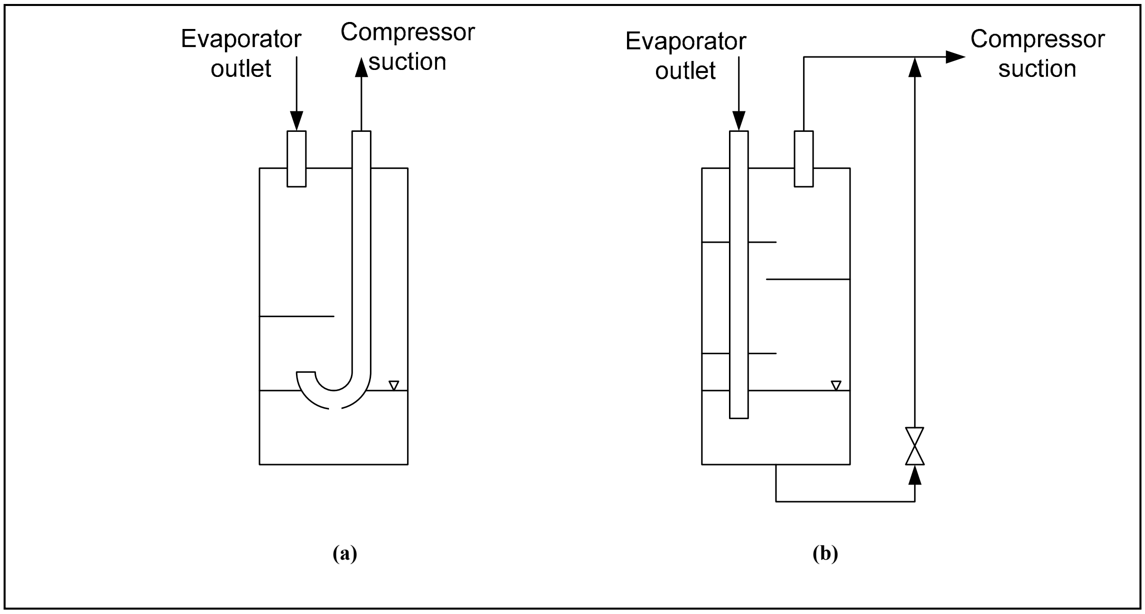


Figure 3. Fabricated CO_2 HPWH Prototype

The prototype employed a hermetic reciprocating compressor designed for transcritical CO_2 operation. This compressor has a rated heating capacity of about 2.2 kW when operating at 278 K evaporating temperature and 9 MPa discharge pressure. The operating envelope of this compressor extends to a maximum operating discharge temperature and pressure of 400 K and 12 MPa, respectively. An oil separator was installed in order to ensure proper oil return to the compressor. The oil return line of the separator was connected through a metering valve to the compressor suction line. As shown in Figure 2, two separate gas coolers were installed. The wrap-around gas cooler is made of approximately 21 m long, 6.35 mm diameter copper tubing with a rated burst pressure of 20.7 MPa wrapped around the bottom half of the storage tank in a counter flow arrangement. A brazed plate heat exchanger was designed and used as the external gas cooler. The water and refrigerant flows were in a counter-flow arrangement to improve the gas cooler effectiveness. A fixed speed pump was used with a bypass valve to control the mass flow rate of the water through the gas cooler. The pump inlet was connected to the bottom of the hot water tank and hot water from the gas cooler is injected at the top of the tank.

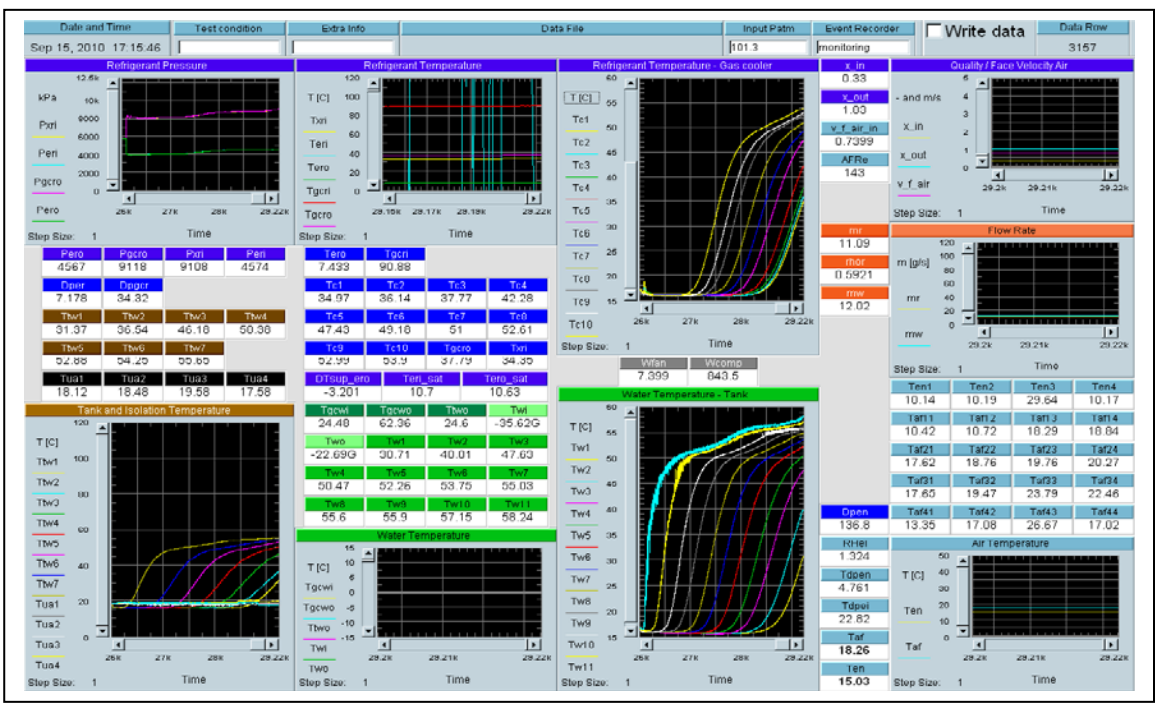
The prototype evaporator had air side geometry identical to the typical HPWH evaporators used in the US. The refrigerant flow circuiting was changed from a two circuit design to a single circuit design in order to ensure proper oil return and to maintain high tube-side heat transfer coefficient. The evaporator tubes were selected to withstand operating pressures up to 10 MPa. Since the CO_2 HPWH is an experimental prototype, a specially designed accumulator was used such that the J-tube orifice is replaced by a metering valve connecting the liquid at bottom of the accumulator to vapor at the top of the accumulator as shown in Figure 4. The metering valve at the bottom of the accumulator also provided the means to return the oil to the compressor when needed. Finally, a suction-gas heat exchanger or an Internal Heat Exchanger (IHX) was designed to provide additional subcooling after the gas cooler in order to increase the evaporator capacity. The IHX was installed with a 3-way valve to allow system operation with and without it. The IHX is a counter flow tube-in-tube heat exchanger with the high pressure fluid flowing through the inner copper tube and the lower pressure fluid flowing through the outer aluminum tube.

Figure 4. Prototype Accumulator Design: (a) Conventional, (b) Experimental



The prototype was properly instrumented to accurately evaluate system performance. A data acquisition system was used to monitor real-time data and provide pressure-enthalpy (P-h) and temperature-enthalpy (T-h) cycle analysis. An example screenshot of the data acquisition display is shown in Figure 5. Online P-h and T-h diagrams were generated in Microsoft Excel using NIST REFPROP (Lemon et al. 2010) function calls to allow monitoring of the cycle state points during testing.

Figure 5. Sample Screen Shot from the Data Acquisition System



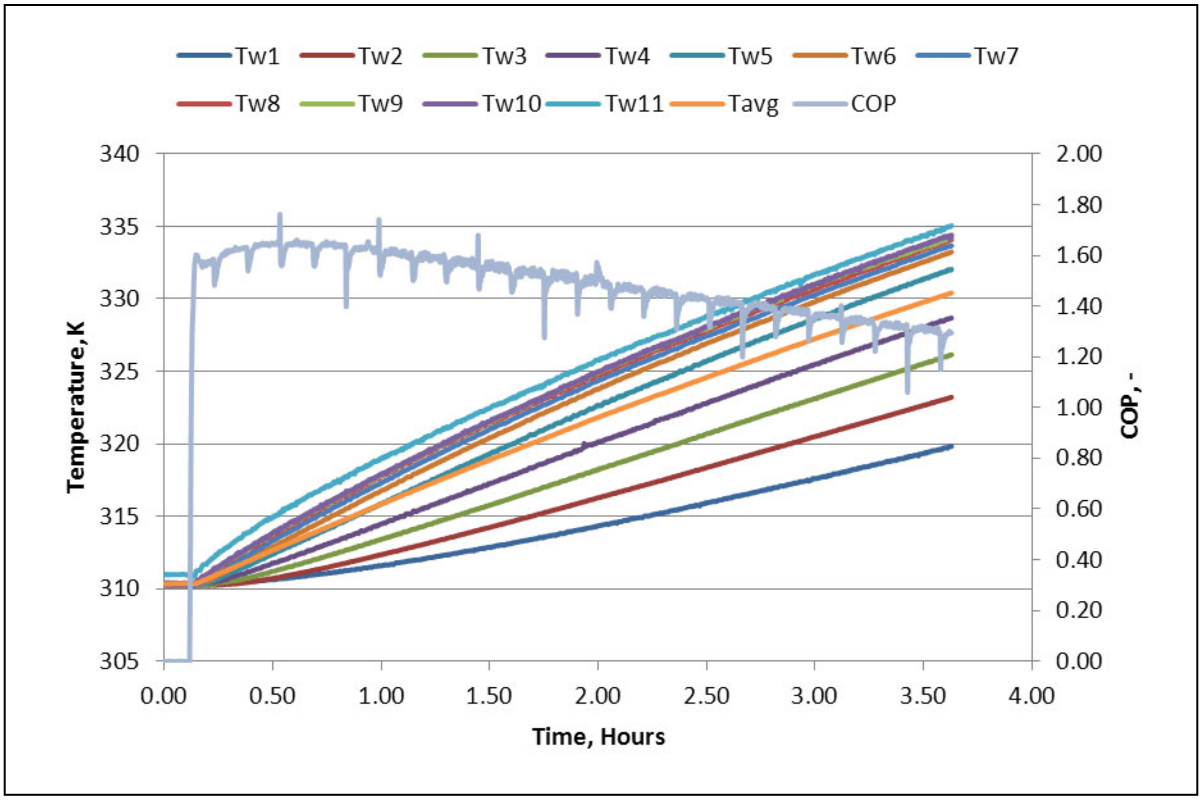
Experimental Results

An initial tank heat up test was performed using the external gas cooler with the IHX enabled. The prototype was able to heat the water to 328.15 K in 260 min with an average cycle COP of 2.68 under the following conditions:

- Ambient temperature = 292.15 K and a relative humidity = 50%.
- Evaporator airflow rate = 510 m^3/h
- Initial average tank temperature = 288.15 K
- Water mass flow rate = 11.2 g/s
- Oil separator oil return valve open for 20s every 30min

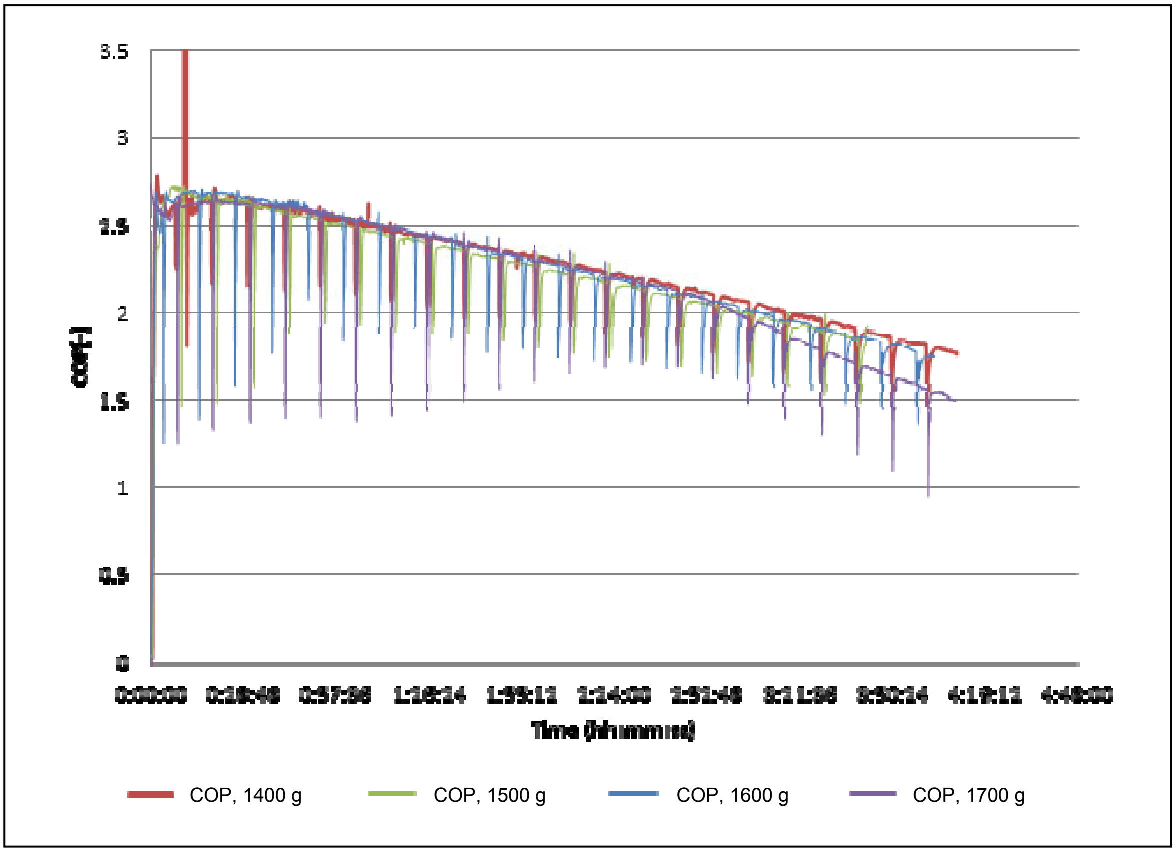
The wrap-around gas cooler configuration was tested next using the same working conditions except for an initial tank temperature of 311 K. Figure 6 depicts the results over 3.5 hours of heating from 311 K until the average tank temperature reached 330.4 K. In Figure 6, T w refers to water temperatures at 11 points equally spaced top to bottom inside the tank. With an average COP of 1.49, the system with the wrap-around gas cooler exhibited a lower COP than the system with the external gas cooler. The results also showed poor temperature stratification in the water tank. Maintaining an appropriate temperature stratification is crucial for ensuring higher COP and capacity. The frequent drop in COP shown in Figure 6 is associated with the oil return valve opening. Whenever the oil return valve is opened the system performance degrades due to the hot-gas bypass effect.

Figure 6. CO_2 HPWH Heating Up Tests (Wrap-around Gas Cooler)



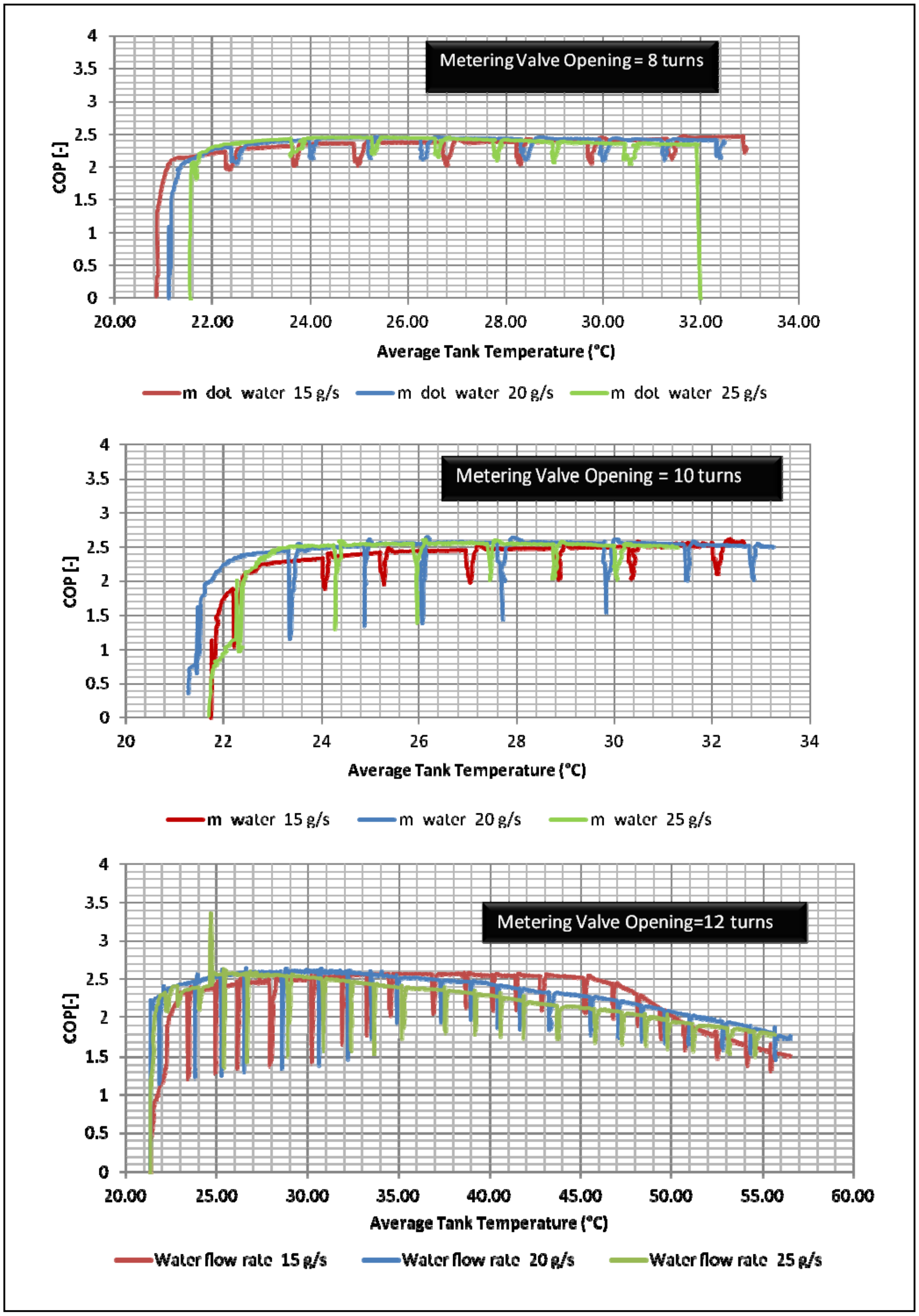
The efforts were then focused on the CO_2 HPWH with external gas cooler configuration due to its superior performance. CO_2 charge optimization tests were performed over the entire tank heating starting with an initial water temperature of 287.6 K and finishing with an average temperature of 330.4 K. The system charge was varied between 1.4 to 1.7 kg. During these tests; the oil recirculation valve was opened for 10 seconds every 5 minutes. The summary of the charge optimization test is shown in Figure 7. The results indicate that 1.4 kg of charge provides the best average COP as the tank heats up. However, slightly higher charge results in better performance during the initial tank heating. This suggests that a proper charge management system should be employed in the design of CO_2 HPWH to ensure optimal operation over the entire operating domain.

Figure 7. CO_2 Charge Optimization Test



In addition to the charge optimization, other system operating parameters were investigated such as the expansion valve opening and the water flow rate through the external gas cooler. The results of these investigations are summarized in Figure 8. The water flow rate was varied from 15 g/s to 25 g/s, and the metering valve opening was changed from 8 turns to 12 turns (18 turns is the maximum opening of the metering valve). These results suggest that for best system performance, the water flow rate should be set at 20 g/s with the metering valve opening set at 12 turns.

Figure 8. Optimum Operation Tests (Water Flow Rate and Metering Valve Opening)



Simulated Use Test Results

Further performance tests were performed based on the procedures described in the DOE standard, "Energy Conservation Program for Consumer Products: Uniform Test Method for Measuring the Energy Consumption of Water Heaters" (DOE 1998). The efficiency of the water heater when tested and normalized to the conditions prescribed in the DOE procedure for the 24 Hour Simulated Use Test is termed the Energy Factor (EF). In this test procedure, the water heater's thermostat(s) are adjusted to yield an average temperature in the water heater of 330.4 K ± 2.8 K. The ambient air temperature is maintained between 291.5 K to 294.3 K and the inlet water is regulated to be 287.6 K ± 1.1 K. The consumption of water is simulated by drawing a total of 243 L ± 3.8 L from the water heater via 6 equal draws, each at a flow rate of 11.4 L/min ± 0.95 L/min. These draws are initiated at the start of the first 6 hours of the test. No hot water draws are imposed during the remainder of the 24 hour test. The EF is computed by dividing the amount of energy removed as hot water by the normalized energy consumption.

Using the optimum operating conditions identified above, the 24-hour simulated use test was carried out to obtain the EF. The compressor engaged almost immediately after the first 37.8 liter draw began. The heat pump was engaged for the entire six-draw profile in the whole test sequence. After all six 37.8 liter draws were completed, the compressor continued operating for two more hours to raise the tank temperature back to the water heater set point of 330.4 K (135°F). The EF of this prototype was calculated to be 2.0; complying with the Energy Star rating criteria. However; the system performance was greatly affected by the compressor and gas cooler performance. The compressor did not perform well over the range of operating conditions experienced during the simulated use test. The experimental results obtained in the laboratory testing did not compare well with the manufacturer's published performance data. A rotary compressor is envisioned to provide better performance. As for the brazed plate gas cooler, the effectiveness was less than the design value. This is mainly due to the low water-side velocities. An optimized gas cooler design can increase the system capacity and COP greatly. With an improved compressor and gas cooler, the authors expect the system to operate at an EF on the order of 2.5.

Discussion

HPWHs are a relatively mature technology, although they are not commonly seen in the residential sector. They have gained popularity over the past five years within the water heating industry because of their ability to deliver significantly more heat for the same amount of electricity compared to traditional electric storage water heaters (SWH). As efficiency increases through continued research and development, the opportunity continues to grow for HPWHs to replace traditional SWHs as they approach the end of their lifetimes or as new housing installations are needed.

The magnitude of savings achievable from HPWHs ranges by geographic location. In general, HPWHs are best suited for regions with a large presence of electric resistance water heaters, high household water usage, high electricity rates, low availability of low cost fuel alternatives (e.g., natural gas, propane, oil), mild or warm ambient temperatures that allow for much heat to be extracted from surrounding air, and housing layouts with sufficient vertical clearance and surrounding space for installation. CO_2 as a refrigerant instead of an HFC allows HPWHs to maintain acceptable efficiency and capacity ratings at much lower ambient temperatures. Therefore, HPWHs are in a much better position to compete with more traditional models used in colder climates. The goal of introducing a CO_2 HPWH into the U.S. market is to help expand the overall appeal of the HPWH segment relative to less efficient SWH models and to introduce it into new regions that currently do not realize the benefits of HPWHs due to low ambient temperatures.

Currently, HPWH unit shipments account for only a small fraction of the overall water heater market with nearly 15,000 HPWH units sold in 2009 (D&R International, 2010). Annual sales nearly quadrupled between 2009 and 2010, according to preliminary data, to over 59,000 HPWH units sold as new models were introduced into the market. HPWH sales are very small in comparison to nearly 8 million gas and electric storage water heaters sold in 2010 alone. In both 2009 and 2010, HPWHs accounted for less than 1% of annual water heater market sales.

Technological hurdles for CO_2 HPWHs may include their 1) energy factor, which is currently lower than most HFC-based HPWHs; 2) ability to meet industry standards, because it is uncertain whether CO_2 HPWHs will meet building construction and safety standards since the CO_2 vapor compression cycle operates at much higher pressure than conventional refrigerants; and 3) potentially inadequate reliability that is often associated with HPWHs. Market hurdles facing CO_2 HPWHs include the payback period necessary to overcome the initial price premium; and lack of incentive for many buyers (e.g., landlords) to purchase the best technology. Addressing these technology and market hurdles is the key to the CO_2 HPWH success.

Conclusions and Recommendations

This paper presented recent efforts regarding the design and fabrication of a CO_2 HPWH for the U.S. market. The prototype's flexible design offered testing of several configurations (two gas cooler configurations, with and without IHX). Furthermore, the system design allowed for various operating parameters to be investigated. The design relied on components that would not result in significant cost addition over current HFC HPWHs available in the U.S. market. Hence a single speed compressor, single speed water pump, a double walled brazed plate heat exchanger, and simple expansion device were used. This is significantly different from designs encountered in the Japanese CO_2 HPWHs "EcoCute" which are driven by a more complex controller offering variable speed compressor, pump, and an electronic expansion valve.

The operating parameter investigation showed that the system was operated best with a system charge of 1.4 kg of CO_2 and a water flow rate of 20 g/s. The system was run under the standard DOE ambient and water supply rating conditions with 300 CFM of air flowing through the evaporator. The evaporator air flow setting is similar to current HFC based HPWHs used in the U.S. market. With this set of operating conditions, the system was shown to provide an EF of 2.0 which is the minimum required to achieve an Energy Star rating. The authors suggest the use of an improved compressor and gas cooler design along with an electronic expansion valve to improve the Energy Factor rating and low ambient temperature performance. These component improvements would result in a price premium over currently available HFC based HPWH due to current production scale as well as content value.

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