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Feasibility Analysis of a Commercial HPWH with CO_2 Refrigerant

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OAK RIDGE NATIONAL LABORATORY
MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

ORNL/TM-2017/127

Energy and Transportation Sciences Division

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Appendix A. Test Condition Matrix

ACRONYMS

HPDM	Heat Pump Design Model
HPWH	Heat Pump Water Heater
CFD	Computational Fluid Dynamics
GWP	Global Warming Potential
BTO	Building Technology Office
ECM	Electronically Commutated Motor
EF	Energy Factor
FHR	First Hour Rating
UEF	Unified Energy Factor
COP	Coefficient of Performance
FEMP	Federal Energy Management Program
HFO	Hydroflouroolefin

ABSTRACT

A scoping-level analysis has conducted to establish the feasibility of using CO_2 as refrigerant for a commercial heat pump water heater (HPWH) for U.S. applications. The DOE/ORNL Heat Pump Design Model (HPDM) modeling tool was used for the assessment with data from a Japanese heat pump water heater (Sanden) using CO_2 as refrigerant for calibration. A CFD modeling tool was used to further refine the HPDM tank model. After calibration, the model was used to simulate the performance of commercial HPWHs using CO_2 and R-134a (baseline). The parametric analysis concluded that compressor discharge pressure and water temperature stratification are critical parameters for the system. For comparable performance the compressor size and water-heater size can be significantly different for R-134 and CO_2 HPWHs. The proposed design deploying a gas-cooler configuration not only exceeds the Energy Star Energy Factor criteria i.e. 2.20, but is also comparable to some of the most efficient products in the market using conventional refrigerants.

EXECUTIVE SUMMARY

This report describes results of a scoping-level analysis to establish the feasibility of heat pump water heaters (HPWH) with CO_2 as refrigerant. The specific goal of the work was to evaluate the performance of a CO_2 HPWH and compare it with a system using conventional refrigerant (R-134a). This effort was undertaken in response to a clearly expressed target in the FY10 DOE Building Technology Office (BTO) Statement of Needs for the water heating (WH) program element. Specifically, there has been an emphasis on designing HPWHs which are highly efficient compared to electric resistance or gas-fired water heaters. With the phase-out of conventional refrigerants including R-134a, it is critical to evaluate the performance of alternative refrigerants which can potentially be successful drop-in-replacements.

A preliminary analysis was conducted to analyze the performance of an existing system which uses CO_2 as refrigerant and for which enough experimental data is available. A Sanden HPWH was selected for this purpose and the measured performance was used to calibrate the HPDM model. CFD modeling provided ancillary information to confirm processes such water temperature stratification and “backflow effect” causing mixing during water draw events. The calibrated model was then used to run a parametric analysis where factors such as water supply temperature, water circulation rate, tank stratification, and condenser configurations were considered. The performance of a commercial CO_2 system was compared with a similar system based on R-134a as the refrigerant. It was concluded that CO_2 HPWH performance can be comparable to that of HPWHs using R-134a, more so with a separated gas cooler configuration. This configuration showed better performance than the wrapped tank configuration for both CO_2 and R134a systems, but requires the use of additional infrastructure (pump, tube-in-tube heat exchanger etc.). For comparable performance, the CO2 system has a much smaller and higher pressure compressor and requires a gas cooler about twice as large as the condenser for the baseline R134a system. Furthermore, it is important to maintain an appropriately controlled water circulation rate in order to achieve higher water temperature stratification in the tank - an essential requirement for higher performance for CO_2 HPWH systems. CO_2 HPWH systems can perform with less drop-off in performance in lower ambient temperature environments and thus are well-suited for split systems; however, the associated additional equipment can hinder the application at large scale.

1. BACKGROUND

Heat pump water heaters (HPWH) are today widely used in both Japan and Europe mainly because the energy costs are high and government provides special incentives to increase their usage. However, their acceptance in the US has been relatively slow without such adoption drivers. Major obstacles to HPWH acceptance include performance, reliability, and initial and operating costs. As a matter of fact, conventional electric resistance and gas storage water heaters are the most commonly used systems in US. Even though there has been a continuous effort to improve the performance of these systems, the energy efficiencies for such systems are much less than prescribed efficiency for equipment to qualify for Energy Star labeling. In contrast, the thermal efficiency (COP) of electrically driven heat pumps using conventional refrigerants (R134a, R410a) are in the 3 to 5 range, compared to 0.8 to 0.95 observed for electric and gas water heaters which is a significant improvement.

CO_2 (also known as R744) has emerged as a viable refrigerant for heat pump technology. The natural refrigerant has multiple advantages including zero-toxicity and flammability and lowest GWP among all well-known refrigerants. The technology became popular in early 1990's in Japan and now many Japanese manufacturers have fully commercialized residential CO_2 HPWH's (collectively known as "Eco-Cute") with reported COP's ranging from 4.1 to 4.8. These reliable systems have been widely accepted as a highly promising technology and have considerably reduced Japan's dependence on electric and gas water heating. Most of the Japanese units are termed split systems where the evaporator and gas cooler are installed outdoors, unlike integrated HPWH's used in Europe and North America. (The Japanese split HPWHs are more like U.S. packaged air-to-air heat pumps in that both refrigerant heat exchangers are in the same outdoor box with the heat transferred to the indoors by water instead of by air.) CO_2 HPWHs for European markets also have separate compartments containing larger evaporators and gas coolers but are located indoors near the indoor water tanks, while drawing some or all of their air source from outdoors. For Japanese split HPWHs and European integrated HPWHs, often-time the ambient temperature is relatively lower compared to U.S. integrated systems and CO_2, due to its inherent characteristics, is well-suited for such operation. One important factor which has contributed to high acceptance of HPWHs in Japan is the use of CO_2 as the refrigerant which provides relatively high efficiency at high supply temperatures even in relatively colder climates. The present study is focused on performance evaluation of CO_2 as a refrigerant for typical integrated HWPH's used in US and Europe.

2. ANALYSIS OF AN EXISTING SYSTEM -- SANDEN GEU-45HPA HPWH

The Northwest Energy Efficiency Alliance (NEEA) contracted with Ecotope, Inc. and Cascade Engineering Services, Inc. to conduct a laboratory assessment of the Sanden (model GEU-45HPA with GES-15QTA tank) heat pump water heater (HPWH) of European integrated design for northern climate installations (Larson and Logsdon, 2013). Cascade Engineering Services of Redmond, WA evaluated the GEU-GES integrated combination using a testing plan developed by Ecotope to assess heat pump water heater performance. The plan consisted of a series of tests to assess equipment performance under a wide range of operating conditions with a specific focus on lower ambient air temperatures down to 30°F.

The parameters recorded included measurements of basic characteristics and overall performance, including first hour rating and Department of Energy (DOE) Energy Factor (EF); determining heat pump efficiency at lower ambient temperatures (30° F, 50° F, and 67° F); conducting a number-of-showers test at 50° F ambient; and measuring airflow across the evaporator coil under different ducting regimes. Appendix A presents a table describing all tests performed for subject study.

The Sanden HP used in the study is a split system (GEU-45HPA heat pump with GES-15 QTA tank), but with both units located indoors, and has an inverter driven CO_2 compressor and a tube-in-tube gas cooler.

Figure 1 shows the schematic system configuration. In the figure, the heat pump mechanical system is in the module at the left and the water tank is at the right. Table 1 shows some important characteristics of the system.

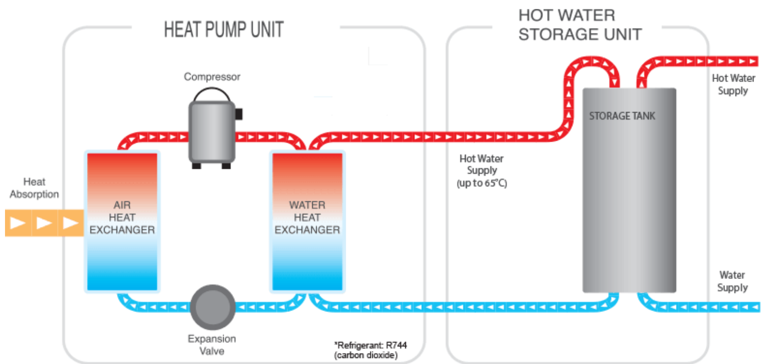

Figure 1. Schematic of Sanden HPWH (a European integrated system) (from Sanden product literature)

Table 1. Characteristics of Sanden GEU-45HPA heat pump (Larsen et al, 2013)

Component	Measurement / Description
Resistance Elements	None
Heat Pump* (W)	1,000 - 2,200
Fan** (W)	35
Standby (W)	< 1
Tank Volume (Gallons)	39.7
Refrigerant	R-744 (CO_2)
Airflow Path	Inlet and exhaust on top through separate eight-inch-diameter ports
Dimensions (per module)	31" Wide x 27" Deep x 38" High

Notes: *Includes compressor, circulation pump and fan. Range depends on water and ambient temperature.
**Measured at max flow with no ducting attached

This water heater is designed specifically for the European market, which results in different design parameters than those for the United States market. For example, the tank has a storage volume of ~40 gallons, does not have electric resistance elements, and has a fixed temperature set-point at 149° F. Ecotope et al evaluated the unit as-is; however, any equipment destined for the United States needs to follow design parameters prescribed by local authorities. As an example, DOE recommends the supply water set-point at 125°F which is significantly lower than the set-point used in current study. Also, most of the commercial HPWHs in US have tank volumes greater than 80 gallons. Nevertheless, the study provided enough information to calibrate the HPDM model and conduct some preliminary analysis. The main reasons for selecting this unit were its good energy efficiency, availability of product information, and extensive test data for calibration of the various modules of the HPDM simulation tool. The EF for the tested model of the product is about 3.39 which greatly exceeds the minimum EF criteria required for Energy Star (minimum 2.0 EF for the pre-2015 test procedure which is approximately equal to a UEF of 2.2 by the current procedure) and thus is a good representation of the most efficient products available in market. Figure 2 shows the test performance for the 24-

hour Energy Factor test consisting of six 10.7-gallon draws (64.2 gallons in total) equally spaced over six hours, followed by eighteen hours of standby, as reported by Ecotope et al.

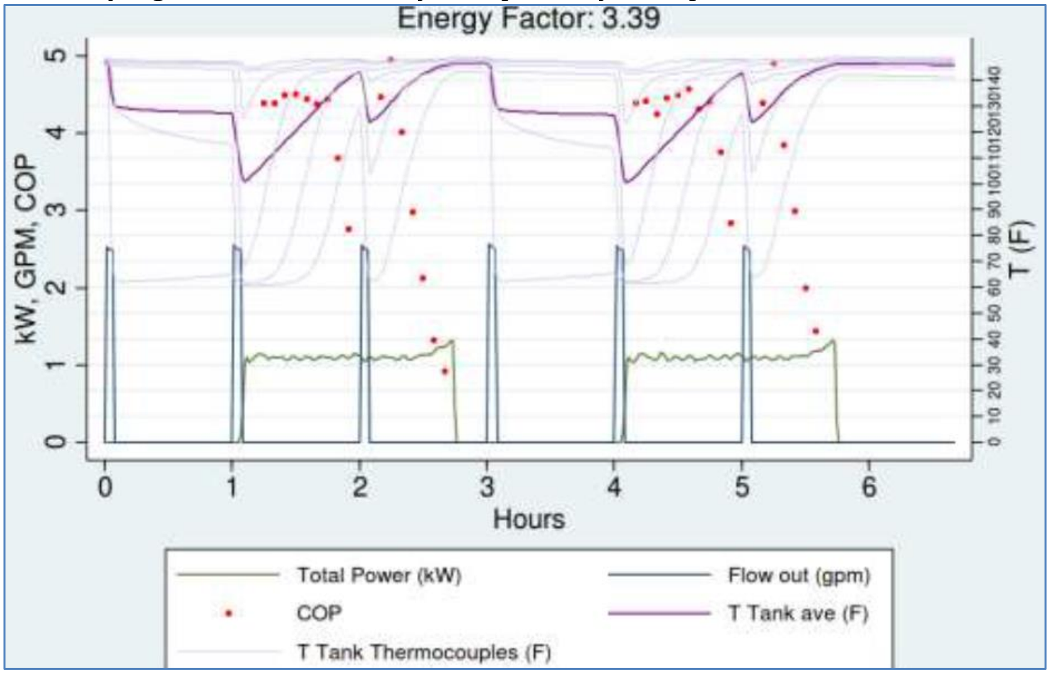

Figure 2. Performance during first six hours of 24-hour EF test (Larsen et al, 2013)

An important observation from the figure is the high water temperature stratification indicated by tank thermocouples. Other important performance parameters are presented in Table 2.

Table 2. Performance characteristics for Sanden GES HB50

Metric	Measured Value
First Hour Rating (gal)	58
Energy Factor (std. conditions)	3.39
Energy Factor @ 50° F Ambient	2.8
Northern Climate Energy Factor	2.98
Tank Heat Loss Rate (Btu/hr° F)	2.2

It is important to note that the above described performance analyses and key parameters, such as FHR and EF metrics, were based on the pre-2015 DOE test method for water heaters (US Code of Federal Regulations, 2010). In the following sections, the HPDM simulations used the same method to be consistent except towards the end where UEF and FHRs based on the newer DOE test method were utilized (DOE 2013) to compare the performance of CO_2 and R-134a systems. More specifics are provided in the following sections.

3. PERFORMANCE EVALUATION CRITERIA

U.S. Department of Energy (DOE) initiated a rulemaking to consider amendments to its old test procedures for covered residential and commercial water heaters as per recommendation of The American Energy Manufacturing and Technical Corrections Act (AEMTCA). Based on extensive testing, a new performance evaluation procedure was defined. According to the new procedure the set point for water temperature is 125^{o}F and the test condition for inlet water and ambient air temperatures are prescribed as 58^{o}F and 67.5^{o}F (35-45% relative humidity) respectively. The key performance metrics used to evaluate and compare the performance of water heaters (WH), including HPWHs, are listed below.

3.1 FIRST HOUR RATING

First Hour Rating (FHR) is a measure of the available hot water capacity of the WH (in gallons). According to the new DOE test method, hot water (125±15^{o}F) is drawn from the tank as long as the leaving water temperature is 67±2^{o}F higher than the entering water temperature. Once the leaving temperature drops below the prescribed limit, the supply is stopped until the set point of 125±15^{o}F is met again, followed by another draw. Following this procedure, the total water drawn from the tank during one hour indicates the total capacity of the heat pump and electric resistance heaters.

3.2 UNIFIED ENERGY FACTOR

Unified Energy Factor (UEF) is a measure of the efficiency of the system [per the most recent standard test methods and procedures (DOE 2013)]. It accounts the ratio of net amount of heat gained by the system (by heating the water) at the end of the test period to the total power required to operate the system. The previous EF test procedure used a single water draw pattern – six equal water draws of ~10.7 gallons each spaced equally during the first five hours of the EF test – applied to all WHs (including HPWHs) with a storage tank. In contrast, the new method uses the measured FHR value to define the draw pattern used for the UEF test. Table 3 provides the details of the draw pattern for a storage water heater based on FHR.

Table 3. Water draw pattern based on FHR

FHR greater or equal to (gals)	***FHR less than (gals)***	***Draw pattern for 24-hr UEF***
0	20	Point of use
20	55	Low usage
55	80	Medium usage
80	Max	High usage/Commercial systems

When our FHR analysis was conducted, it was concluded that under all parametric conditions the appropriate draw pattern would be for medium usage (FHR varied between 60-65 gallons). Table 4 presents the hot water draw pattern for a medium usage storage tank. The specified water draw pattern was used to determine the UEF.

Table 4. Medium usage hot water draw procedure

Draw Number	***Time During Test (hh:mm)***	***Volume gals(L)***	***Flow Rate GPM (LPM)***
1	00:00	27.0 (102)	3 (11.4)
2	00:30	2.0 (7.6)	1 (3.8)
3	00:40	1 (3.8)	1 (3.8)
4	01:40	9.0 (34.1)	1.7 (6.5)
5	10:30	15.0 (56.8)	3 (11.4)
6	11:30	5.0 (18.9)	1.7 (6.5)
7	12:00	1.0 (3.8)	1 (3.8)
8	12:45	1.0 (3.8)	1 (3.8)
9	12:50	1.0 (3.8)	1 (3.8)
10	16:00	2.0 (7.6)	1 (3.8)

11	16:15	2.0 (7.6)	1 (3.8)
12	16:30	2.0 (7.6)	1.7 (6.5)
13	16:45	2.0 (7.6)	1.7 (6.5)
14	17:00	14.0 (53.0)	3 (11.4)
Total Volume Drawn Per Day: 84 gallons (318 L)			

3.3 COEFFICIENT OF PERFORMANCE

Coefficient of performance (COP) is another related criterion which measures the performance of the heat pump. It is important to differentiate between UEF and COP. The heat pump's COP is always higher than the UEF because, in determining the COP, the energy lost through the tank wall and insulation (also known as skin effect) is also included in the heat supplied by the heat pump and hence results in a larger number. UEF is the performance of the system based on how much energy is used to heat up the water delivered from the tank, the ultimate goal of the system, while COP represents the performance of heat pump only.

4. PERFORMANCE MODELING

The DOE/ORNL Heat Pump Design Model was used to model the heat pump water heater (HPWH). HPDM is a well-recognized, public-domain HVAC equipment modeling and design tool (Shen and Rice, 2014; http://hpdmflex.ornl.gov/hpdm/wizard/welcome.php). This work draws upon some component modeling aspects of a previous ORNL HPWH analysis for forced-flow designs (Baxter et al, 2011).

4.1 FEATURES OF HPDM MODELING TOOL

HPDM has been used for multiple research and development projects across research organizations and industry. Over the past few years the simulation tool has been updated and calibrated with experimental data for numerous studies and is capable of simulating rather complex system configurations accurately, which was not possible until more recently. Some of the key features important to the current study are described as following.

4.1.1 Compressor

AHRI 10-coefficient compressor maps (ANSI/AHRI 540-99, 2010) have been used to calculate mass flow rate and power consumption when available. This information along with a compressor shell heat loss ratio relative to the power input allows calculation of the refrigerant-side vs. air-side energy balance from inlet to outlet (Dabiri and Rice, 1981). For the baseline study, the original compressor map, developed for R-134a was used. For the CO_2 system, fixed values for isentropic and volumetric efficiency were assumed, based on the manufacturer's rated values. For both compressors, shell heat loss was assumed to be 10% of the input power.

4.1.2 Evaporator

HPDM uses a segment-to-segment modeling approach, which divides a single tube into numerous mini segments. Each tube segment has individual air-side and refrigerant-side entering states, and considers possible phase transition; the ε-NTU approach has been used for heat transfer calculations within each segment. Air-side fins are simplified as an equivalent annular fin. Both refrigerant and air-side heat transfer and pressure drop were considered in the study; the coil model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant-side entering and exit states, maldistribution, and accept two-dimensional local air-side temperature, humidity and velocity inputs; the tube circuitry and 2-D boundary conditions were provided by an input file. In addition to the functionalities of the segment-to-segment fin-tube condenser, the evaporator model was capable of simulating the dehumidification process. The method of Braun, et al. (1989) was used to simulate cases of water condensing on an evaporating coil, where the driving potential for heat and mass transfer is the difference between enthalpies of the inlet air and saturated air at the refrigerant temperature. The heat transfer correlation published by Thome (2002) was used to calculate the evaporator two-phase heat transfer coefficient. Air side heat transfer correlations were obtained from Wang (2001), specific to different fin types, e.g. louvered fin, wavy fin, slit fin, etc.

4.1.3 Wrapped Tank Condenser

A wrapped-tank condenser model was developed specifically for this investigation, using a segment-to-segment modeling approach. The flow-pattern-dependent heat transfer correlation published by Thome (2003) was used to calculate the condenser two-phase heat transfer coefficient. As for the evaporator, the pressure drop correlation published by Kedzierski (1999) was used to model the two-phase pressure drop. The heat transfer between the refrigerant and water is calculated by considering the forced convection at the refrigerant-side, tube and water tank wall conductance, and water-side natural convection. The coil model simulates temperature and pressure variations in mini-segments along the refrigerant flow direction, and interacts with the node temperatures of a transient, stratified water tank model.

4.1.4 Gas Cooler (Water Heater)

Gas cooler can be tube-in-tube heat exchanger or brazed plate heat exchanger. The coolant-to-refrigerant heat exchangers are modelled using a segment-to-segment approach, i.e. dividing refrigerant and coolant channels into numerous segments. In each segment, the model considers heat transfer and pressure drop, as well as the energy balance between the refrigerant and coolant sides. For brazed plate heat exchangers, the refrigerant and coolant side heat transfer correlations were obtained from a manufacturer's product performance data; for tube-in-tube heat exchangers, the heat transfer correlations were obtained from Rousseau (2003). The gas cooler model can simulate any flow patterns, i.e. counter flow and parallel flow.

4.1.5 Stratified Water Tank

The tank model included the parameters such as thermal conductivity of thermal paste (for the wrapped-tank design) and the insulation covering the tank. Thus the heat loss from the tank was captured for the full time of the operation. The transient tank model accounted for one-dimensional water temperature stratification caused due to the natural convention. Along with the heat transfer from wrapped condenser tube, a simultaneous operation of supplemental electric water heaters was also simulated. These electric heaters are placed to provide continuous hot water supply in the case of heat pump failure or when the heat pump itself cannot provide enough heat to meet the demand. The water tank model divides water mass into 10 nodes (control volumes) in vertical direction. Each water node has uniform temperature and exchanges heat with condenser tubes in the section. The tank model captures the mechanisms of 1) piston flow, i.e. make-up water enters at the bottom and pushes the flow to the supply port at the top; 2) heat conduction between neighboring nodes; and 3) upward flow and mixing caused by natural convection. The buoyancy driven upward flow and heat transfer is simulated by Churchill and Chu (1975). In addition, we innovatively applied an empirical tuning parameter to correlate a backflow mixing effect, i.e. whirls caused by water draw, as shown in Figure 8. The tuning factor enables calibration of the water tank model to match measured water stratifications as indicated in Figure 8.

4.1.6 Expansion Devices

The compressor suction superheat degree and condenser subcooling degree were explicitly specified. As such, a specific expansion device control was not modeled and a simple assumption of constant enthalpy expansion was used.

4.1.7 Fans and Blowers

The air flow rate and power consumption were direct inputs from the laboratory measurements.

4.1.8 Refrigerant and Water Lines

Heat transfer in refrigerant connecting lines was ignored and the pressure drop was calculated using a turbulent flow model, as a function of the refrigerant mass flux. Heat losses from connecting water lines were also assumed to be zero.

4.1.9 Refrigerant Properties

In order to establish the thermophysical properties of different refrigerants, instead of using the property call function programmed to call REFPROP 9.1 directly, property look-up tables were generated. The program used 1-D and 2-D cubic spline interpolation algorithms to calculate refrigerant properties via the look-up tables; this greatly boosted the calculation speed without significantly compromising the accuracy.

4.2 ASSUMPTIONS TO SIMULATE THE CO_2 SYSTEM

The study conducted by Ecotope et al on the Sanden HPWH system provided useful data about the system performance. However, there are multiple unknowns such as compressor characteristics and gas-cooler specifications which required an iterative process to come-up with a system with comparable performance. Following are the assumptions used for the ORNL CO_2 HPWH simulations:

1. A single-speed CO_2 rotary compressor was used with fixed values of 56% isentropic efficiency and 80% volumetric efficiency. The discharge pressure varied from 1150 to 1650 psia and the discharge temperature was less than 230°F. These numbers were adopted from manufacturer data. The compressor was sized to match the HP run time to the Sanden HPWH for the same sized 39.7-gallon tank.
2. The evaporator heat exchanger and fan power were similar to Sanden system (Table 1)
3. A Grundfos UPM ECM pump (variable-speed) was used to circulate the water through gas-cooler and the water tank; the water flow rate was controlled to maximize the tank stratification by adjusting the water flow rate though the gas cooler. Equation (1) was used to predict the pumping power for given water temperature (*T*) and flow rate (*V*). As an example, pump power at 1 gpm water flow, 70°F is 11.38 W.
$$P_{pump} = 20.285 - 0.065T + 0.001T^2 - 9.396V + 3.341V^2 - 0.047VT \qquad (1)$$
4. The tube-in-tube gas-cooler was sized to obtain good agreement in design point COP with the Sanden unit which resulted in a 0.5°F temperature difference (approach temperature) between the CO_2 exit and water inlet.
5. A compressor discharge pressure control was developed based on the method described in section 4.3.
6. The calibrated CO_2 heat pump model was then coupled with an 85-gallon water tank for performance evaluation with the medium draw pattern for a commercial application.

4.3 DISCHARGE PRESSURE CONTROL STRATEGY FOR OPTIMUM PERFORMANCE

Due to the unique characteristics of carbon dioxide near the critical point and beyond, for a limited pressure range, the slope of the isotherms is small; however, at other values above and below this range, the isotherms are quite steep. As the maximum pressure of the cycle increases, the state point of gas cooler outlet changes correspondingly. As shown in Figure 3, the quantity Δh_3 is large compared to Δh_2 and this causes an increase in the COP of the cycle (equation 2- *h* is the enthalpy of the fluid).

$$COP = \frac{(h_2 - h_3) + \Delta h_2 + \Delta h_3}{(h_2 - h_1) + \Delta h_2} \qquad (2)$$

As the pressure line becomes steeper, the state point of the gas cooler outlet can reach to 3b, and then the increment of Δh_3 gets much smaller compared to initial situation (Bin et al, 2014). Hence there is an optimum discharge pressure where COP is maximum and any deviation from the optimal pressure results in lower COP. Essentially the optimum pressure range is the area where the isotherm is fairly flat (red curve). Furthermore, the pressure range where the maximum COP increase occurs depends significantly on gas cooler outlet temperature (Kauf, 1999). This in turn depends heavily on the inlet water temperature for a CO_2 system (Kauf, 1999; Chen and Gu, 2005)

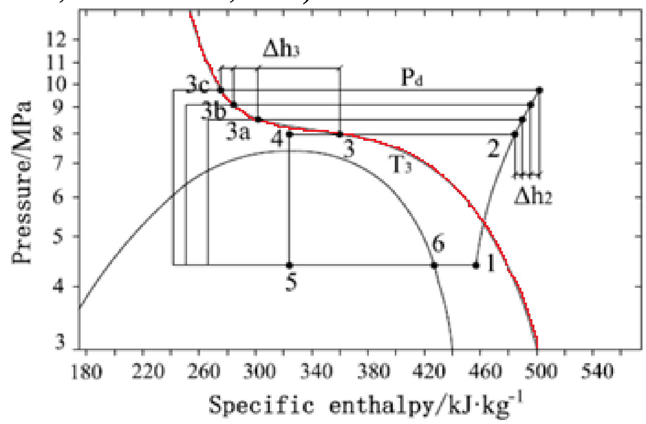

Figure 3. P-h diagram of supercritical CO_2 cycle for various high pressures (Bin et al, 2014)

This unique characteristic associated with super-critical CO_2 system dictates that there are optimal discharge pressures at which the system performs best and this was used to develop the control strategy for the water flow rate through the gas-cooler. Figure 4 shows the variation of the COP with compressor discharge pressure for various water temperatures at the inlet of the gas-cooler. Based on this information, optimal discharge pressures can be determined according to the water inlet temperature. All simulations used a standard 67.5 F ambient temperature.

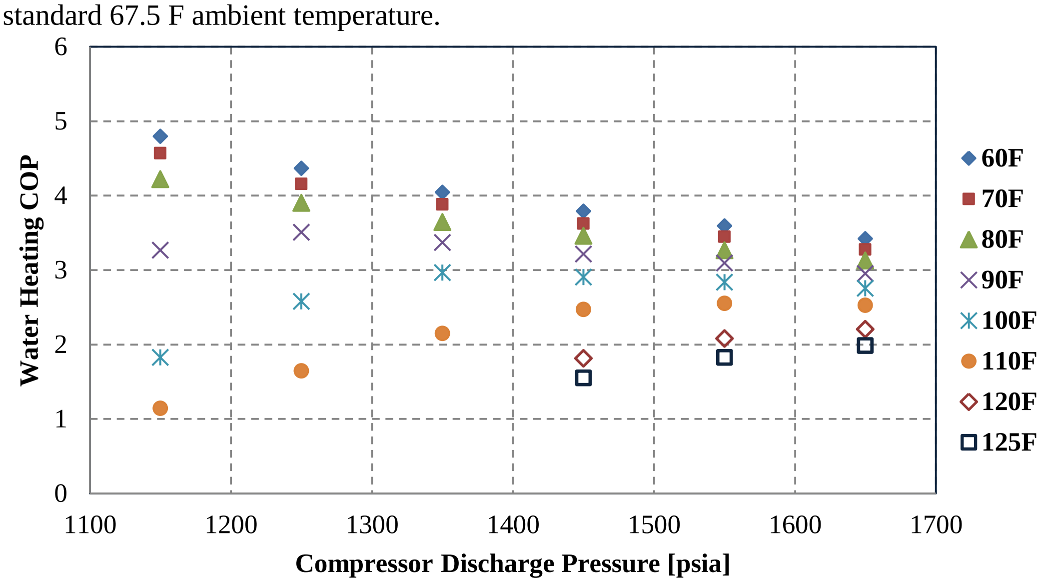

Figure 4. Variation of COP with discharge pressure for water temperature at the gas-cooler inlet

From the analysis described above, a correlation between water inlet temperature and the discharge pressure can be developed for each specific ambient temperature. Figure 5 shows optimum discharge pressure for a range of inlet water temperatures and standard ambient conditions. The resulting curve-fitting gives the equation describing the control method for compressor discharge pressure that was used in the analysis.

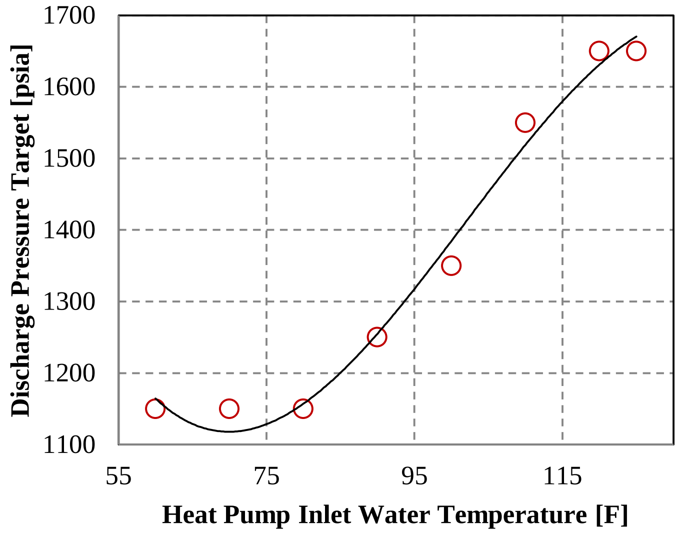

Figure 5. Discharge pressure for optimum COP at various water temperatures at the gas-cooler inlet and standard ambient conditions.

4.4 COMPARISON OF THE ORNL HPWH TO SANDEN GEU-45HPAHPWH

Based on the optimum discharge pressure control and assumptions described in section 4.2, the pseudo-steady-state performance of the modeled ORNL HPWH was evaluated for a range of inlet water temperatures. Figure 6 compares the HP COP for both systems. The optimized ORNL system shows better performance for the inlet temperatures less than 83°F while Sanden performs better beyond that. Near the set-point of 125°F, the performance of both systems is comparable. Fixed compressor speed operation of the ORNL simulation (vs increasing speed for the Sanden unit with water temperature) and fixed isentropic and volumetric efficiencies likely account for most of these differences.

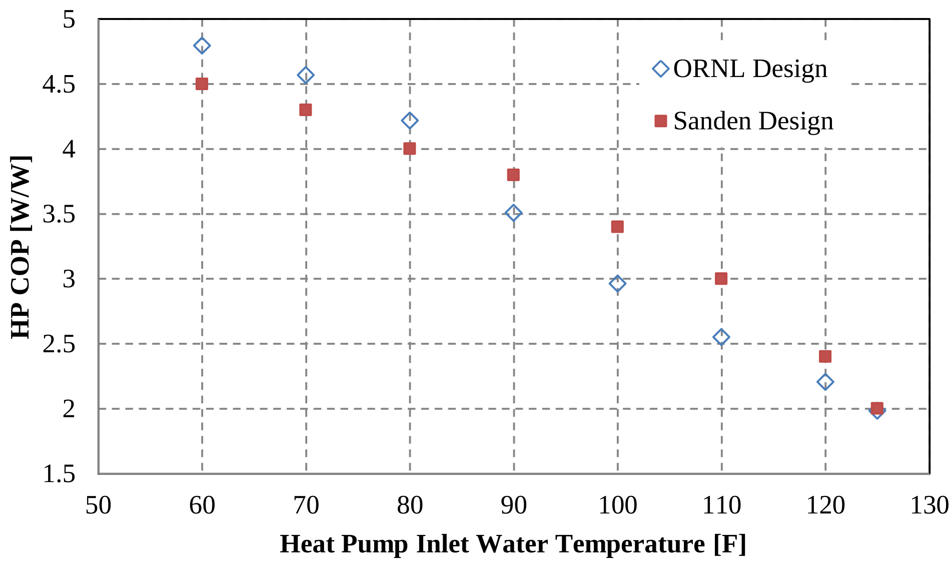

Figure 6. HP COP for ORNL versus Sanden system at standard ambient conditions.

HPWH capacity is another important parameter to consider. Figure 7 presents the comparison of the HPWH capacity for both systems. Even though the ORNL design shows a comparable performance for 60 F inlet water temperature, the capacity is lower compared to Sanden at higher temperatures.

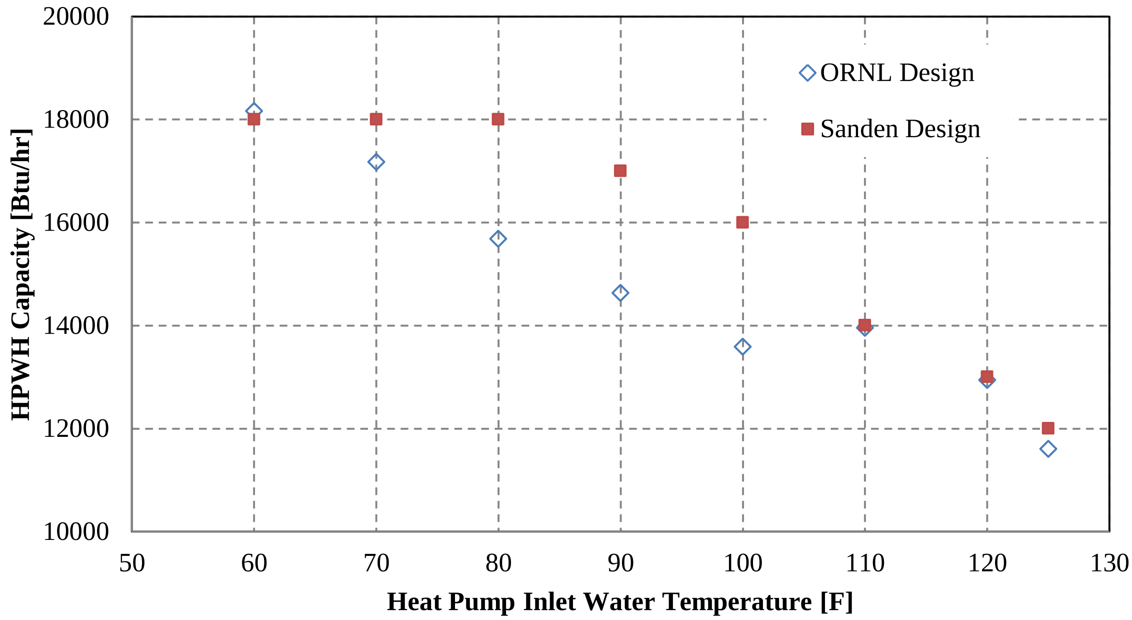

Figure 7. HP Capacity for ORNL versus Sanden system at standard ambient conditions.

4.5 CFD MODELING TO VALIDATE THE TANK MIXING ASSUMPTIONS DURING TANK DRAWS

4.5.1 Bulk Mixing During Water Draw

It is important to include and validate the bulk mixing occurring during water draw from the tank. In order to validate the assumption, a series of CFD simulations were carried out to observe the impact of mixing due to the "backflow effect". Figure 8 represents water velocity streamlines (in m/s) for an adiabatic water tank as the hot water is withdrawn from the top and replenished by cold water entering the tank through the dip tube near the bottom. The bulk mixing happening solely because of "backflow effect" is clear. It is important to note that all previous modeling approaches including EnergyPlus (Engineering Reference 2015) ignore this relatively important phenomena which dominates the natural convention currents but occurs only during water draw events.

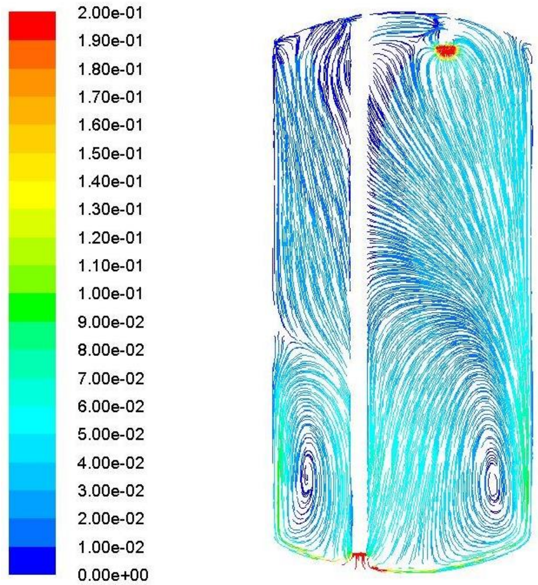

Figure 8. Streamline for water flow in the tank during draw (scale shows velocity-m/s)

4.5.2 Stratified Temperature Profile in Vertical Direction

Water stratification caused due to the heat flux is an important aspect of tank modeling as can be seen from the 2-D CFD model of a tank observed during a previous study (Gluesenkamp et al., 2016). As described above, the process was included in HPDM modeling by dividing the tank into various nodes where each node was considered at uniform temperature and neighboring nodes can interact to exchange heat. One important observation from the CFD analysis was the insignificance of radial temperature variation as can be seen in the Figure 9 and hence this was ignored in the HPDM model.

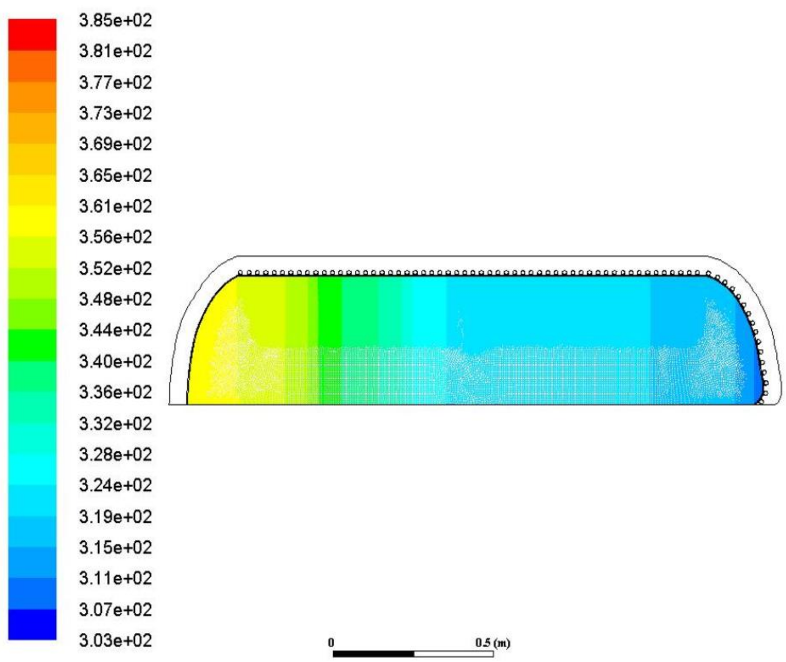

Figure 9. Water temperature profile in the tank (scale shows temperature-K)

5. PERFORMANCE COMPARISON WITH R-134a HPWH SYSTEM

The following two sections summarize the findings from the HPDM modeling simulations to compare the performance of the ORNL CO_2 HPWH system with a comparable R-134a system. The calibrated model developed based on the experimental data from the Sanden system is used for further analysis.

5.1. IMPACT OF SUPPLY WATER TEMPERATURE

Supply water temperature has a direct impact on the performance of the system. The baseline for the study, the Sanden HPWH, has a set point for supply water at 149°F and the resulting EF was approximately 3.39 as indicated in Table 2. The calibrated ORNL design with same supply water temperature showed almost similar performance. However, we need to evaluate the performance for the DOE prescribed UEF test with 125°F as the supply water temperature. It was found that the same CO_2 system with lower set-point for supply water now has a much better performance, as the UEF value was approximately 3.82. There was a noticeable decrease in the compressor run time and maximum discharge temperature as well. An important distinction to note here is the method to calculate the Energy Factor. For both ORNL set-point 149^oF and Sanden set-point 149^oF cases, the pre-2015 method was used to calculate the Energy Factor. However, for the ORNL set-point 125°F case, the post-2015 method has been used to calculate the UEF.

Table 5. Performance comparison of ORNL (calibrated) system for different supply temperatures to Sanden system

	ORNL (Set Point 149^{o}F)	**Sanden (Set Point 149^{o}F)**	**ORNL (Set Point 125^{o}F)**
Energy Factor [J/J]	3.37	3.39	3.82*
HP Run Time [min]	192	Near 200	167
Max Discharge T [F]	202	N/A	190
Average Tank Supply T [F]	142	N/A	119
Tank top T setting [F]	149	149	125
Draw in 24-hr [gallon]	64.2	64.2	84 (Commercial draw pattern)
First Hour Rating	59.9	58	51.7

*** UEF rating**

5.2. R-134a HPWH VS. CO_2 HPWH

While establishing the standalone performance of CO_2 HPWH, it is necessary to determine how the system performs compared to a conventional system. Most of existing HPWHs use R-134a as refrigerant. In order to compare the performance of the ORNL CO_2 system with a HPWH using R-134a the following assumptions were made:

1. A single-speed rotary compressor performance map obtained from the manufacturer was used in simulation. The compressor was sized to have comparable HPWH run time as the CO_2 HPWH.
2. The outdoor heat exchanger and fan power were the same as the Sanden system (Table 1)
3. The same Grundfos UPM ECM pump (variable-speed) was used to circulate the water through the condenser and the water tank and the water flow rate was controlled to maximize the tank stratification by adjusting the water flow rate.
4. The tube-in-tube water heater was sized to achieve 5% liquid phase before exiting the heat exchanger at design conditions with a fixed refrigerant charge.

Before comparing the performance of R-134a to CO_2 system, the impact of the set-point for the supply temperature is shown in Table 6, where the performances at the Sanden set-point of 149^{o}F and the DOE set-point of 125°F have been compared for R-134a as refrigerant for HPWH application. The system was optimized to have a comparable performance to the ORNL CO_2 system at the 125°F set-point as the goal was to have a comparable performance for both systems. The compressor was sized to have similar heating capacity and the heat exchanger (condenser) was designed to have comparable heat transfer rate. Note that the heat pump run time and maximum discharge temperature are lower for the 125°F set point. All the other important operating parameters are listed in Table 6.

Table 6. Performance of R-134a HPWH at two set-point temperatures

Performance Parameters	***Set Point 149°F***	***Set Point 125°F***
Unified Energy Factor [J/J]	3.33*	3.87
HP Run Time [min]	192	176
Max Discharge T [F]	178	161
Average Tank Supply T [F]	141	119
Tank top T setting [F]	145	125
Draw in 24-hr [gallon]	64.2	85
Average compressor isentropic efficiency [%]	62%	61%
Average compressor volumetric efficiency [%]	90%	92%
First hour rating [gals]	52.4	51.7

*The UEF is defined based on the procedure described in “performance evaluation criteria” which uses a set point of 125°F. However, 149°F (same as the rating procedure for the Sanden test) was used as set point to illustrate how the set point can impact the UEF value.

Table 7 compares the compressor displacement volume and required length of tube-in-tube water heater for CO_2 and R-134a as refrigerant for the HPWH. The components have been sized so that both systems have the same performance. There are some obvious differences as the required compressor size is small for CO_2 system compared to R-134a system for comparable performance due to higher suction pressure and density for CO_2. However, the required tube-in-tube gas cooler length is more than two times larger compared to the water heater size for R-134a. This difference can be attributed to the single vs. two-phase heat transfer (Figure 10). For CO_2 the working fluid does not change phase while for R-134a system the system enters as superheated vapor and condenses to liquid phase. A significant portion of the heat exchanger (HX) has two-phase refrigerant which has approximately 4-5 times higher heat transfer coefficient for the same area than supercritical CO_2 fluid.

Table 7. Component sizing for comparable performance for CO_2 and R-134a

	CO_2 HPWH	***R134a HPWH***
Compressor Displacement volume [in^3]	0.26	1.60
Length of tube-in-tube water heater [ft.]	39.96	17.13

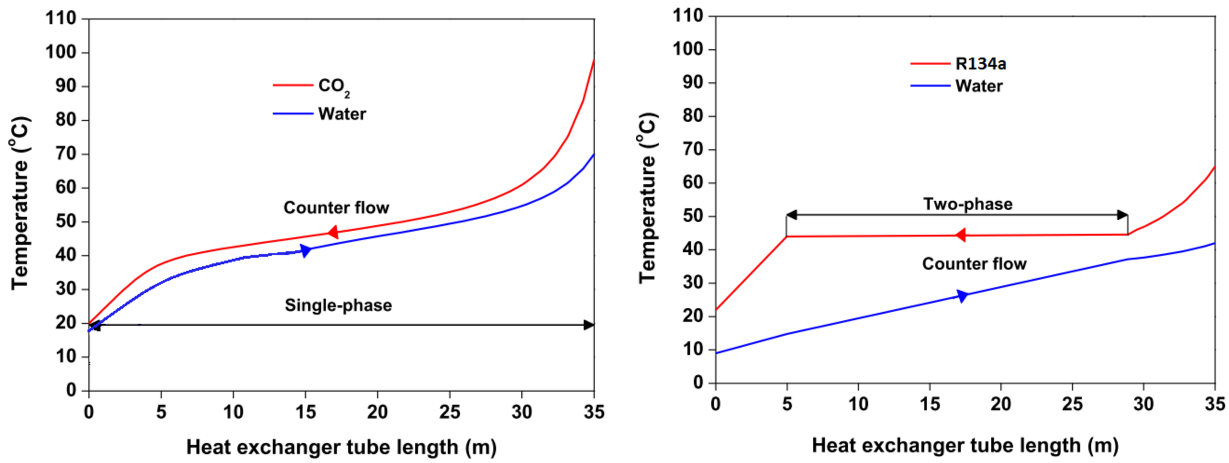

Figure 10. HX process for CO_2 (left) and R-134a (right) systems (red line-refrigerant, blue line- water)

For the CO_2 system with an adequately sized gas-cooler, the water inlet temperature dictates the CO_2 exit temperature, which strongly affects cycle COP, as discussed earlier in section 4.3. On the other hand, for the R-134a system, the water exit temperature establishes the R-134 condensing temperature which strongly affects R-134a cycle COP. Due to limited tube length with liquid phase, the exiting liquid refrigerant temperature is less affected by the water inlet temperature. It can be deduced from these observations that water stratification in the tank is more important for a CO_2 HPWH and that this system is highly sensitive to the water temperature stratification in the tank. It was observed from the report for the Sanden system as well that higher stratification helps to achieve better performance. On the other hand, the conventional refrigerants (R-134a) are less sensitive to the stratification even though more subcooling helps to improve performance; however the impact is not as dominant as it is for CO_2 system.

5.3. IMPACT OF WATER CIRCULATION RATE ON TANK WATER TEMPERATURE STRATIFICATION

For the gas-cooler configuration in a CO_2 HPWH, the water circulation rate between storage tank and the heat exchanger is an important parameter impacting performance. Water temperature stratification has been considered an important aspect especially for a CO_2-based HPWH as highlighted in Section 5.2 and the water flow rate through the pump dictates the achievable stratification. In order to analyze this effect, two different circulation rates have been considered and the resulting stratification is shown in Figure 11 (circulation rate=1 gpm) and Figure 12 (circulation rate=0.5 gpm). It can be observed that even though the top of the tank was less sensitive to the circulation rate (node 0), for the middle (node 5) and bottom (node 9) of the tank, the temperature profile is quite different for two flow rates. As mentioned earlier that because higher stratification favors better Unified Energy Factor and COP, a relatively lower circulation rate is recommended. However, if the flow rate is too low, it can adversely impact the performance as the water will be over-heated at the exit of gas-cooler which deteriorates the cycle performance. A variable speed pump can be extremely useful in this situation as the water flow rate is varied by the control strategy depending upon the stratification condition in the tank to achieve optimized performance.

Figure 13 provides the details about performance of the CO_2 and R-134a systems with fixed and variable flow rates. For comparison, the findings for both set-points (149°F and 125°F) have been presented. Lowering the flow rate to a fixed value of 0.5 gpm from 1 gpm significantly improves the CO_2 system performance; however, it was observed that water at the exit of the gas-cooler was approximately 155°F for a set-point of 149°F which decreases the performance of the system. On the other hand, when the flow rate is variable and controlled to maintain the water temperature difference in the tank (CtrlDT) to achieve stratification, the resulting UEF is noticeably higher for both systems (more so for the CO_2 system) than a fixed flow rate for a 149°F set-point. For both set-points and systems, the variable-flow rate control for stratification matching results in optimal or near optimal performance with the resulting performance nearly the same between systems. It can also be seen from Figure 13 that the R134a system performance optimizes at higher flow rates than for the CO_2 system, due to the further benefit of lowering condensing pressure.

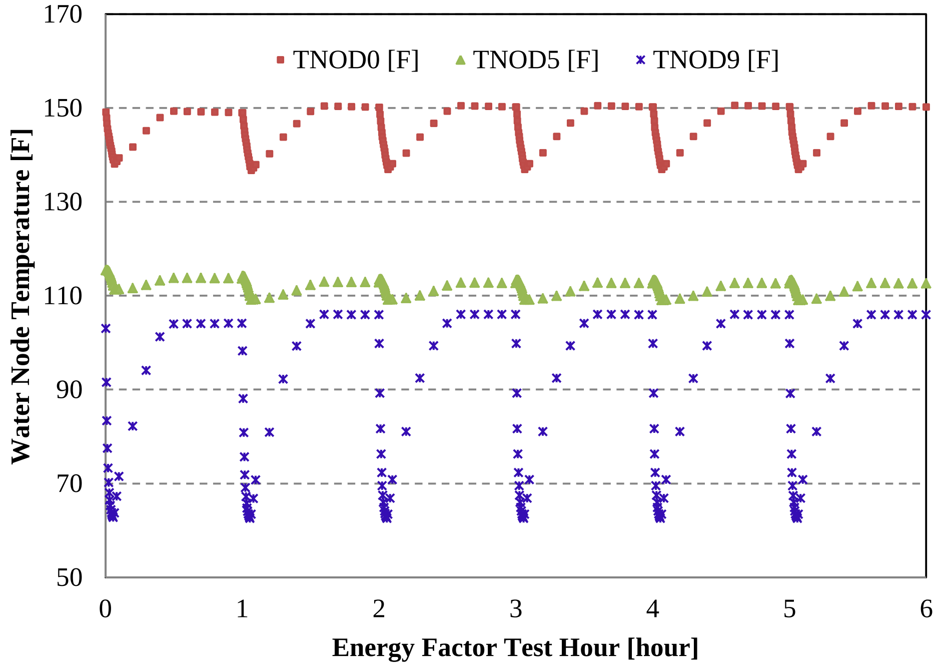

Figure 11: Water temperature stratification for 0.5 GPM water circulation rate for CO_2 HPWH

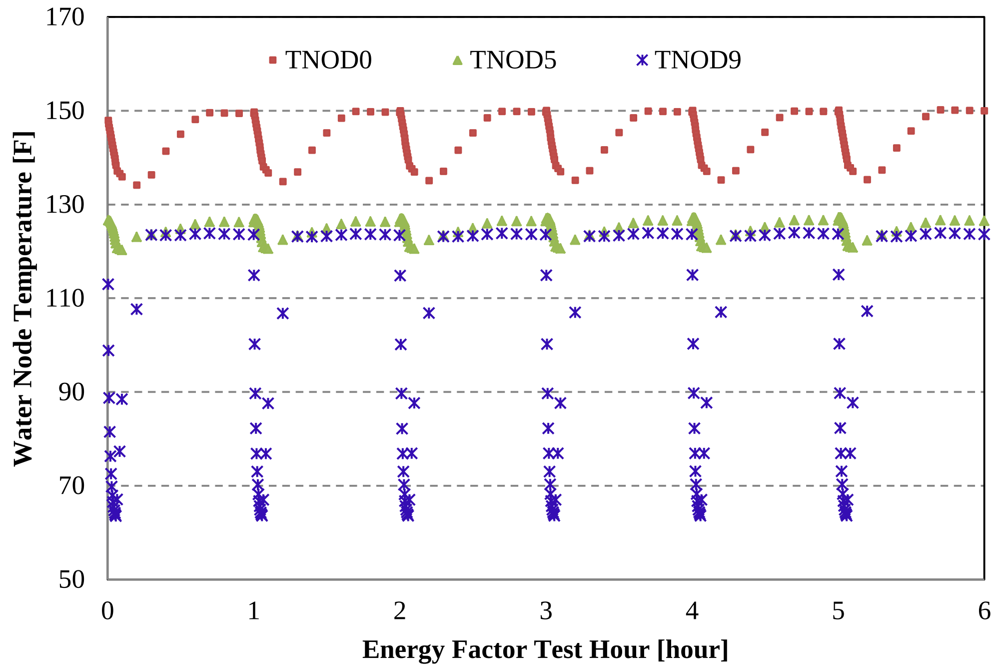

Figure 12: Water temperature stratification for 1.0 GPM water circulation rate for CO_2 HPWH

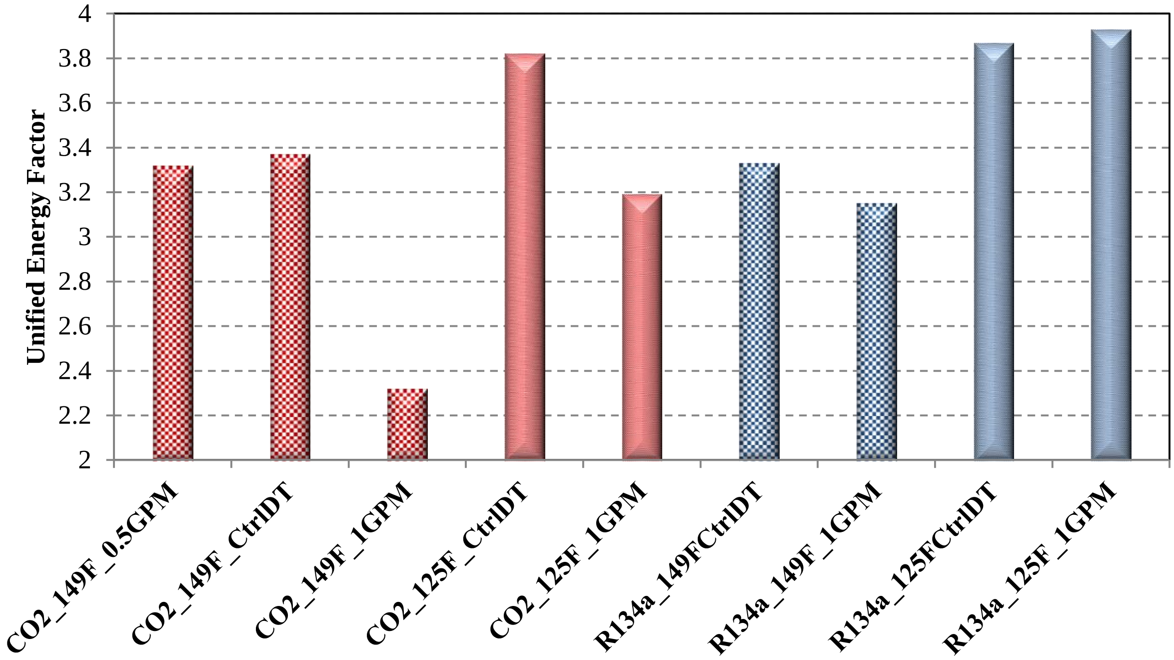

Figure 13: Water temperature stratification for 1.0 GPM water circulation rate for both refrigerant designs

5.4. GAS-COOLER VERSUS WRAPPED CONDENSER COIL CONFIGURATION

The water in the storage tank can be heated by wrapping the condenser coil on the tank surface and insulating the exterior to make sure that most of the heat is transferred through the tank wall to the water as shown in Figure 14. The other potential approach, as assumed so far in the analyses of this report, is to use a gas-cooler (a brazed plate or tube-in tube heat exchanger) which transfers the heat between super-critical CO_2 and the water circulating through the loop (Figure 14). An additional pump to circulate the water is required in this case with appropriate piping for connection to the water heater. Both configurations have advantages and disadvantages. Additional components in a separate modular water heater approach (condenser for R-134a case, gas-cooler for CO_2 system, and water pump) make it more complicated and costly. Furthermore, the fouling on water-side of the gas-cooler requires frequent cleaning in most designs (due to the smaller flow passageways). On the other hand, for the wrapped configurations a better tank thermal insulation is mandatory. Regardless of these various aspects, it is important to analyze the impact of configuration on the performance of the system.

Figure 15 compares the Unified Energy Factor for both configurations with CO_2 and R-134a as working fluids where the water temperature set-point is 125^{o}F and both systems have variable speed pumps to control water flow rate for optimized performance. It is clear that a tube-in-tube gas-cooler (for CO_2 system) or condenser (for R-134a) results in a significantly higher UEF for the same operating conditions. However, it is important to emphasize that this increase in performance has associated costs in terms of extra components, a larger footprint, and potentially additional maintenance requirements. One reason for the performance improvement is that variable water flow control can better match the tank stratification under changing conditions than does a fixed tank wrap configuration.

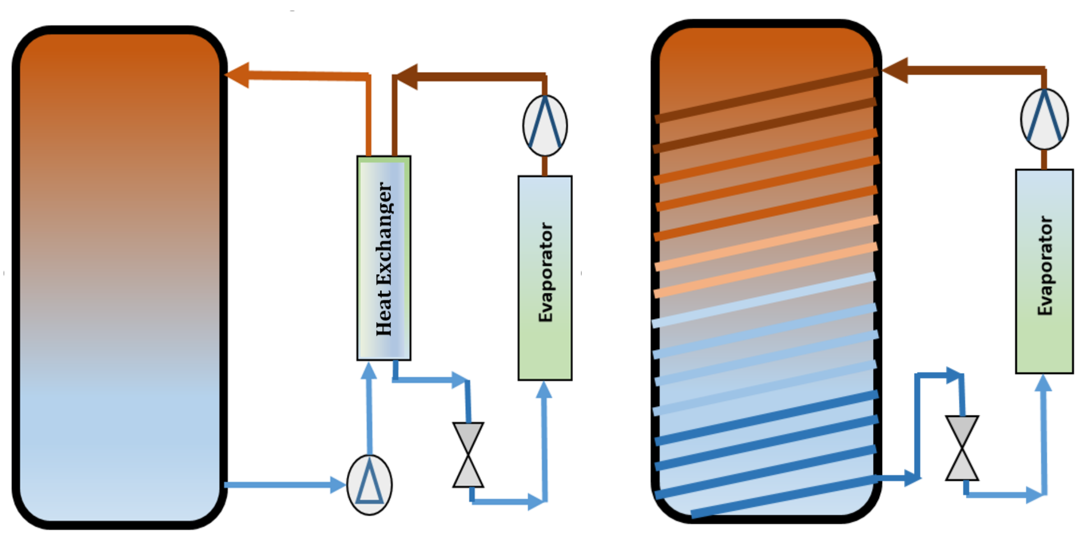

Figure 14: Two configurations for HPWH (left: separate hot water HX, right: wrapped coil HX)

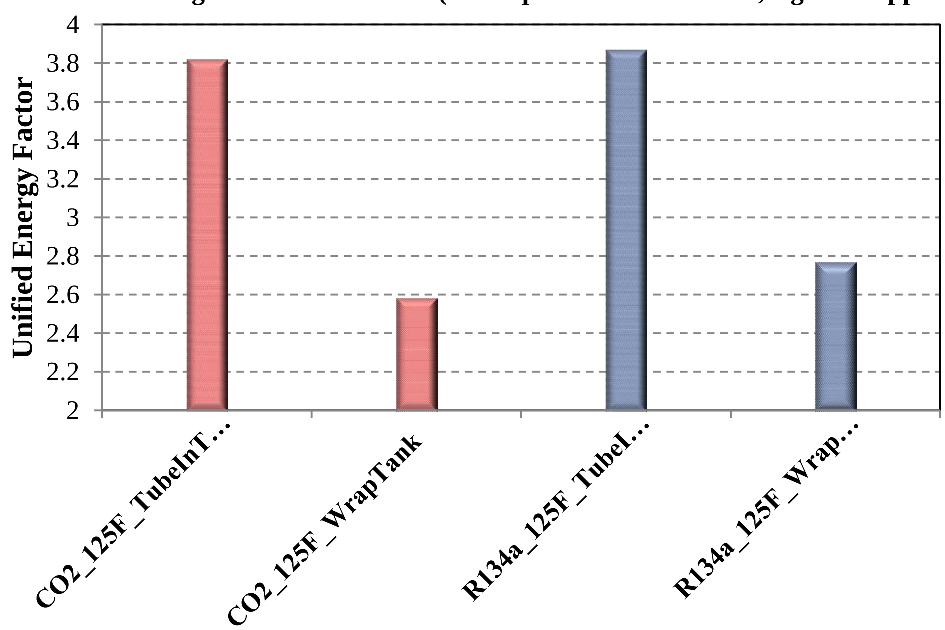

Figure 15: HP performance for tube-in-tube vs. wrapped coil configurations

5.5. IMPACT OF AMBIENT TEMPERATURE

Ambient conditions (dry bulb temperature and relative humidity) directly impact the performance of heat pump water heaters. As presented in Figure 16, the tested Sanden system showed a rather strong sensitivity of COP to ambient temperature. Table 8 shows some predicted results for performance of a HPWH with CO_2 and R-134a as working fluids when the ambient temperature is 50°F (about 20°F lower than the method prescribed by DOE for evaluation of UEF) and the set point for water temperature at 149°F (about 15°F higher than standard set-point). It can be noted that, compared to the earlier results at rated ambient conditions, lower ambient temperatures decrease the Energy Factor for both systems. Another important observation is the heat pump run time. It is clear from the values in Table 8 that CO_2 system has higher capacity since the average run time for comparable performance is relatively lower for the CO_2 system. This result favors the use of CO_2 in split systems (like Sanden unit) where the heat pump is placed outdoors in a relatively cold environment or uses most or all outdoor air source if located indoors as in the European design. Most of the heat pumps produced in Japan are split systems and CO_2 as refrigerant provides a potential advantage by heating more effectively at lower ambient temperatures. The variable-speed capability of the Sanden unit (for both Japanese split and European integrated designs using outdoor air) also shortens the run times at lower ambients by increasing speed to provide a higher, more constant heating output, as can also be seen in Figure 16.

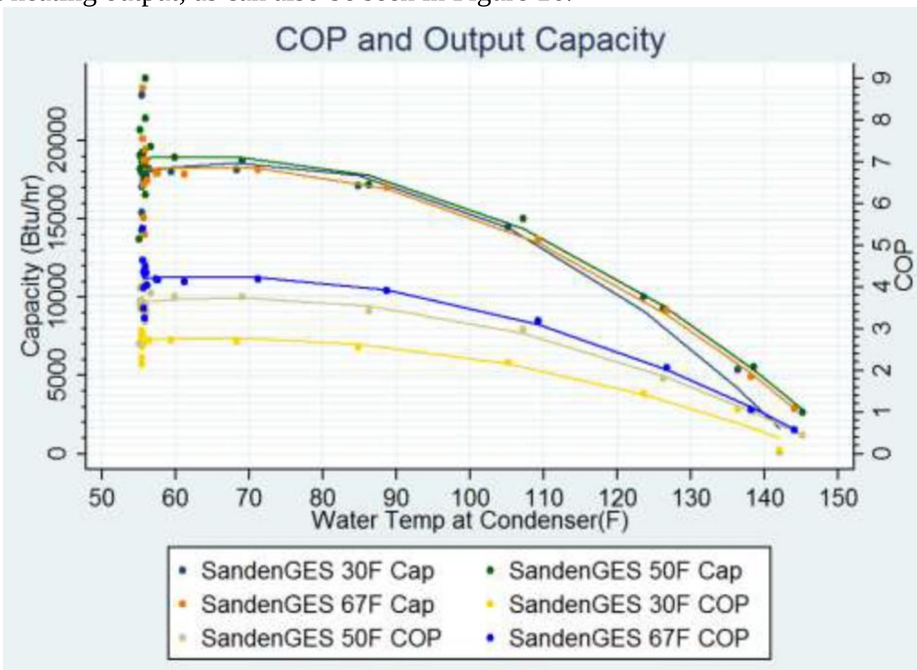

Figure 16. Performance of Sanden system at different ambient temperatures (Larsen et al, 2013)

Table 8. Performance of CO_2 vs. R-134a systems at 50°F ambient temperature

	CO_2 Water temperature set-point 149 °F, Ambient temperature 50 °F	**R134a Water temperature set-point 149 °F, Ambient temperature 50 °F**
Unified Energy Factor [J/J]	2.68	2.73
HP Run Time [min]	235	276
Max Discharge T [F]	229	189
Average Tank Supply T [F]	141	142
Tank top T setting [F]	149	149
Draw in 24 hrs [gallon]	64.2	64.2
Average compressor isentropic efficiency [%]	56%	59%
Average compressor volumetric efficiency [%]	80%	88%

6. COMPARISON OF PERFORMANCE TO CURRENT ENERGY STAR HPWH SYSTEMS

Heat pump water heaters are the most efficient type of electric water heater available and are used as the best available option to electric-resistance WHs. Since the emergence of the technology, multiple high-efficiency HPWH products have been introduced by manufacturers. While examining the performance of alternative refrigerants, it is important to review the performance of existing products. The following table gives a relative comparison provided by FEMP (Energy Star, 2016).

For the analysis presented in Table 9, annual energy cost is calculated based on an assumed electricity price of $0.09/kWh, which is the average electricity price at federal facilities and the lifetime energy cost includes the future electricity price trends and a 3% discount rate. Lifetime cost savings are the difference between the lifetime energy cost of the less efficient model and the lifetime energy cost of the Energy Star model or best available model. The performance for the Best Available Model is based on the April 2016 Energy Star-Qualified products list.

Table 9. Typical performance for various HPWHs in market

Performance	*Best Available*	*Energy Star*	*Less Efficient*
Unified Energy Factor	3.39	2.20	0.95
Annual energy use	958 kwh	1476 kwh	3437 kwh
Annual energy cost	$89	$137	$319
Lifetime energy cost	$1000	$1541	$3587
Lifetime cost saving	$2587	$2046	-

The following table presents a more comprehensive and recent listing of the Energy Factors for energy efficient products identified by AHRI. Most of such products generally use electric heat as the auxiliary heating source.

Table 10. Current best HPWH models in market and associated UEFs (AHRI certified products)

AHRI Number	*OEM Name*	*Model Number*	*Unified Energy Factor*
9060121	A.O. SMITH WATER PRODUCTS CO.	FPTU-50 120	3.24
9060328	A.O. SMITH WATER PRODUCTS CO.	HPTU-80N 120	3.07
8214670	AMERICAN WATER HEATER COMPANY	HPHE10280H045DV 120	3.07
7599406	BRADFORD WHITE CORP.	RE2H80R10B-1NCWT	3.26
7551748	GE APPLIANCES, A HAIER COMPANY	BEH80DCEJSB*	3.26
8797248	GE APPLIANCES, A HAIER COMPANY	GEH80DHEKSC*	3.26
8215264	GSW WATER HEATERS	G1080TDE-HPHE-45 120	3.07
9060345	GSW WATER HEATERS	G1080TDE-HPHE-45N 120	3.07
8215272	JOHN WOOD	JW1080TDE-HPHE-45 120	3.07
8215210	LOCHINVAR, LLC	HPA081KD 120	3.07
9060329	LOCHINVAR, LLC	HPA082KD 120	3.07
8215188	RELIANCE WATER HEATER COMPANY	10-80-DHPHT 120	3.07
9952303	RHEEM SALES COMPANY, INC.	10E80-HP4D	3.50
9952302	RHEEM SALES COMPANY, INC.	XE80T10HD50U0	3.50
8215221	SEARS BRANDS MANAGEMENT CORP.	153.592800	3.07
9060323	STATE WATER HEATERS	HP6-80-DHPT 120	3.07
9060324	U.S. CRAFTMASTER WATER HEATERS	HPHE2F80HD045V 120	3.07
9060326	U.S. CRAFTMASTER WATER HEATERS	HPHE2F80HD045VU 120	3.07

While the simulated UEFs of our analysis of best cases are somewhat higher than the tested units on the market, it should be noted that we have not accounted for heat losses from the high-side heat exchangers and from connecting lines. However, if one assumes that such losses will be similar in size between the CO2 and R134a or alternative HFO systems, the qualitative performance findings of near equal performance levels should still apply.

7. CONCLUSIONS

Based on the findings described above, it can be concluded that CO_2 can be an effective substitute for refrigerant (R-134) for HPWH applications. However, substantial modifications of the system configurations are required to achieve comparable performance such as the additional infrastructure of an appropriate compressor, pump, water-flow control, and gas-cooler. The system is highly sensitive to water circulation rate which directly impacts the stratification and system efficiency and this is an additional requirement along with relatively higher maintenance cost due to potential fouling of the gas-cooler. Regardless of these apparent complexities, the system can perform well in lower ambient temperatures and is most suited for split (i.e. modular) HPWH systems using some or all outdoor air. Another important factor to consider is the system arrangement. The Sanden system was an indoor configuration with relatively smaller heat loss. For an outdoor modular system, these losses can be higher which can further degrade the performance.

8. ORNL RECOMMENDATIONS

Based on the findings described above, ORNL recommends that the project pass scoping analysis and proceed to the next stage, which is finalizing the better option between HFO and CO_2 as an alternative refrigerant based on cost effectiveness and opportunity for innovation, followed by development and lab testing of the appropriate system. For the CO2 system, a more complete CO2 compressor map should be included. Representative compressor shell and storage tank heat loss factors and a cost-benefit analysis for liquid-line/suction-line heat exchanger are additional aspects to be considered.

9. ACKNOWLEDGEMENTS

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APPENDIX A. TEST CONDITION MATRIX

Table A-1. Test matrix for Sanden HPWH performance evaluation (Larsen et al, 2013)

DOE Standard Rating Point Tests												
Test Name	Ambient Air Conditions					Inlet Water		Outlet Water		Airflow	Operating Mode	Notes
	Dry-Bulb		Wet-Bulb							inch. static pressure		
	F	C	F	C	RH	F	C	F	C			
DOE-1-hour	67.5	20	57	14	50%	58	14	149	57	0.0"	"Comfort"	Follow test sequence in Federal Register 10 CFR Part 430 Section **5.1.4**
DOE-24-hour	67.5	20	57	14	50%	58	14	149	57	0.0"	"Comfort"	Follow test sequence in Federal Register 10 CFR Part 430 Section **5.1.5**