Hydrofluoroolefins as Refrigerants for heat pump water heating applications

Kashif Nawaz
Oak Ridge National Laboratory, United States of America, nawazk@ornl.gov

Bo Shen
shenb@ornl.gov

Ahmed Elatar
ORNL, United States of America, elataraf@ornl.gov

Van Baxter
ORNL, United States of America, baxtervd@ornl.gov

HYDROFLUOROOLEFINS AS REFRIGERANTS FOR HEAT PUMP WATER HEATING APPLICATIONS

Kashif NAWAZ[1]*, Bo SHEN[2], Ahmed ELATAR[3], Van D. BAXTER[4]

[1] Oak Ridge National Laboratory,
Oak Ridge, Tennessee, USA
(865)241-0972, nawazk@ornl.gov

[2] Oak Ridge National Laboratory,
Oak Ridge, Tennessee, USA
(865)241-0972, shenb@ornl.gov

[3] Oak Ridge National Laboratory,
Oak Ridge, Tennessee, USA
(865)241-0972, elatafa@ornl.gov

[4] Oak Ridge National Laboratory,
Oak Ridge, Tennessee, USA
(865)241-0972, baxter@ornl.gov

* Corresponding Author

ABSTRACT

Recently there has been a growing interest in utilization of alternative refrigerants for residential and commercial HVAC&R systems due to the potential environmental impacts of conventional fluids (CFCs and HFCs). Hydrofluoroolefins such as R1234yf and R1234ze(E) has shown promising performance and are being extensively considered for range of heat pump applications. This study illustrates the performance evaluation of R1234yf and R1234ze(E) as drop-in-replacement for R134a for heat pump water heating (HPWH). A component-based model is used to predict the performance. Key performance parameters such as unified energy factor, first hour rating and thermal stratification in the water tank are investigated. Along with system modeling appropriate experiments to demonstrate the performance improvements. Both modeling and experimental efforts suggest that HFOs when deployed as alternative refrigerants can provide comparable system performance to that of the baseline system containing R134a. Other concerning parameters such as total system refrigerant charge, the condenser discharge temperature and thermal stratification in water storage tank are established to investigate the potential retrofit of the system to further improve the performance of the system.

1. INTRODUCTION

Residential and commercial water heating uses a significant amount of energy and accounts for approximately 10% of all residential and commercial site energy usage in the United States, making it the fourth largest energy end use by buildings (DOE Building Technologies Office, 2016). On the global scale water heating consumed about 15-20% of total residential energy in 2015 for OEDC and non-OEDC countries, Figure 1 (International Energy Agency, 201 2016).

Despite recent advancements in energy efficiency, most residential water heaters are either conventional natural gas-fired or electricity-fired storage types. Although such systems are quite simple, their system efficiency is very low. Conversely, under appropriate conditions, electrically driven, vapor compression (VC) heat pumps represent a system opportunity with much higher thermal efficiency resulting in significant energy savings (Anderson *et al.*, 1985). Heat

Pump Water Heaters (HPWH) are a relatively newer technology where a VC heat pump (HP) system is used to heat the water by transferring heat from relatively low temperature ambient air to the water in a hot water tank. The traditional heat pump system is highly complex as selection of components (evaporator, compressor etc.) plays a critical role in the overall efficiency of the system (Baxter *et al.*, 2011). When the objective is to heat water, the design becomes even more complicated as then there are some additional components (condenser, water storage tank, tank insulation etc.) which can directly impact the performance of the system (Huang and Chyng, 2001; Shah and Hrnjak, 2014; Franco, 2011; Baxter, 2016).

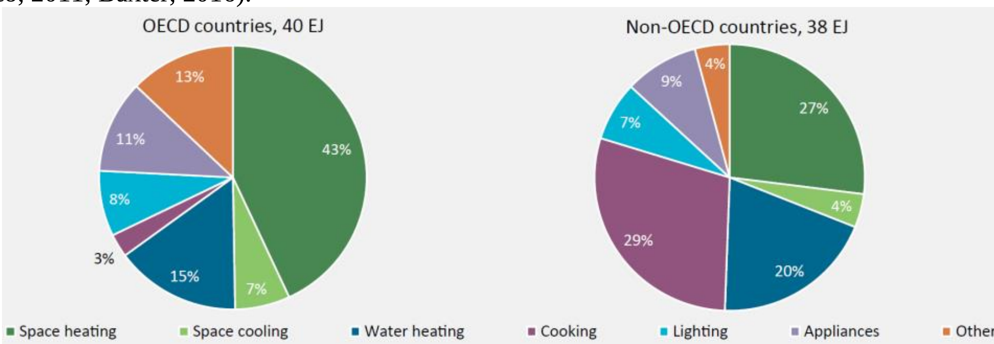

Figure 1: Energy consumed by different utilities for OECD and non-OECD countries

Numerous studies have been conducted to evaluate the thermal performance of HPWH systems with varying levels of complex details including the working fluid, thermodynamic cycle, tank sizes and water draw rate and a range of different analysis methods have been deployed including energy analysis, entropy analysis and exergy analysis for individual components and the whole system (Liapradit *et al.*, 2008; Fernandez *et al.*, 2010). It has been concluded that the coefficient of performance (COP) is affected by many factors, such as, environmental condition, working fluid, refrigerant charge level, expansion device characteristics, water tank and frequency of compressor, and so on as reported by Hepbasli and Kalinci (2009).

A key parameter of interest is the working fluid or refrigerant used in the VC cycle. Early HPWH systems used R-22 which is now in the process of being phased out to its non-zero ozone depletion potential (ODP). R-134a has emerged as the refrigerant of choice for most current HPWH products with reasonably high unified energy factor (UEF) ratings. However, the concerns about its relatively high GWP have led to world-wide efforts to phase down its use in HVAC&R and HPWH applications in the future. Since, HPWH system are essentially heat pumps relaying on inverse refrigeration cycle, the technology has been equally affected (Harby, 2017).

HFO-based refrigerants have been identified as very promising low GWP alternatives for the current higher GWP refrigerants for high-temperature heat pumps (Abdelaziz *et al.*, 2015, 2016; AHRI, 2015). McLinden *et al.*, 2017 reported that only few pure fluids possess the combination of chemical, environmental, thermodynamic, and safety properties necessary for a refrigerant and he maximum efficiency occurs at a relatively high volumetric refrigeration capacity, but there are few fluids in this range. Their comprehensive study proposed HFOs as potential replacement for conventional refrigerants Brown *et al.*, (2010) used the group contribution methods to predict the critical temperatures, critical pressures, critical densities, acentric factors, and ideal gas specific heats at constant pressure for eight fluorinated olefins or HFOs, incluing: R-1225ye(E), R-1225ye(Z), R-1225zc, R-1234ye(E), R-1234yf, R-1234ze(E), R-1234ze(Z), and R-1243zf and compared their thermophysical properties to those of R134a to assess their potential to replace R134a.

There are several studies focusing solely on HFOs or blends as potential substitutes for R134a and R410a in HVAC&R applications. Koyama *et al.*, (2010) conducted a study to compare the performance of R-410a and HFOs in an air-source heat pump (ASHP) system with cooling capacity of 1.8-2.4 kW and heating capacity of 1.6-2.4kW. They recommended use of smaller diameter tube of the heat exchanger and an increased number of heat exchanger circuits to enable the HFOs to more closely match the performance of the baseline system. In another similar study Hara *et al.*, (2010) tested the performance of a residential air conditioner with 4.5 kW cooling capacity using both R-410A and R-1234yf mixture. Most of the prior comparative studies evaluating R1234yf as a substitute for R134a are based on automobile air-conditioning applications. Lee and Jung (2012) compared the performance of R134a and R1234yf and concluded that there was marginal difference in the capacity of the system and the compressor discharge temperature was about 6-7 F lower for R1234yf. Zilio *et al.*, (2011) conducted an experimental study using R1234yf as a drop-in-replacement for R134a based automotive air conditioning system with some modifications. Bryson *et al.*, (2011) evaluated the performance of a car air conditioning system using R152a and R1234yf to replace R134a for automotive applications. For this drop-in-replacement study both refrigerants had COPs and cooling capacities comparable to the baseline system containing R134a. Reasor *et al.*, (2010) evaluated the possibility of R1234yf to be a drop-in replacement for an existing system with R134a or R410A, comparing thermophysical properties and simulating operational conditions. Leck (2010) experimentally studied R1234yf, and refrigerants blends developed by DuPont, as replacement for various high-GWP refrigerants and showed that such alternative refrigerants can meet the performance requirements without significant modifications to the system. Barve and Cremaschi (2012) performed drop-in testing of R-1234yf in a 5 RT (17.6 kW) split system ASHP. Recently, Abdelaziz *et al.,* (2012, 2015) conducted an extensive study evaluating the performance of a wide range of low GWP refrigerants blends containing R1234yf and R1234ze(E) for residential split and commercial rooftop air-conditioning systems. It was concluded that HFO blends as refrigerants are promising replacements for R134a and R410a since there was marginal difference in the performance. HPWH systems using CO_2 as a natural refrigerant has been a relatively recent development in Japan (Yamaguchi *et al.,* 2011; Bowers *et al.,* 2012). Even though the system has shown promising results and has been widely accepted commercially in Japan and other areas, the performance of a CO_2 heat pump is severely affected by the air temperature as well as the inlet and outlet water temperatures (Fernandez *et al.,* 2010; Yamaguchi *et al.,* 2011; Lin *et al.,* 2013). Although most of the above described studies are experimental, several studies have been focused on the performance modeling where researchers have specifically study the components which are relevant to the current analysis i.e., heat exchanger design, thermal stratification etc. Nash *et al.* (2017) modelled a water storage tank having an immersed helical coil. They used a quasi-steady-state approach to model the immersed coil by introducing four tuning parameters to adjust to match the experimental data and concluded that the model's accuracy can be significantly improved by adequately discretizing the tank. Baeton *et al.,* (2016) developed a one-dimensional water tank model accounting for the buoyancy and mixing mechanisms by deriving modeling parameters from CFD simulations and experimental data. Fan *et al.,* (2015) conducted a study for solar water tank with a built-in heat exchanger spiral. They used a CFD analysis to investigate the impact of natural convection on the water temperature stratification and factor leading to the distortion of the stratified pattern. They derived a generalized correlation to calculate the heat loss changing with temperature gradient in the tank, considering the influences of tank volume, height to diameter ratio, tank insulation, thickness and material property of the tank and initial thermal conditions of the tank. Most of such studies have concluded that the performance of a water heating system using a CO_2 heat pump and a storage tank is affected by many factors including the ambient conditions such as air and feed-water temperatures, the hot water demand, and the operating conditions such as startup and shutdown. Such systems require complex configuration involving a variable speed pump and a gas-cooler which makes the cost of manufacturing and maintenance much higher than HPWH systems which can operate at lower pressures.

Though there have been multiple studies on the performance of HFOs as low GWP refrigerants for HVAC&R application, there is rare literature available to explore the potential of such refrigerants for HPWH applications (Murphy *et al.,* 2011). The present study is focused on investigation of the performance of such refrigerants as substitutes for R134a which is currently used by most HPWH manufacturers. The objective is to determine the impact of HFOs refrigerants as drop-in-replacement for R134a. In order to accomplish this an extensive modeling approach has been adopted to determine the feasibility of R1234yf and R1234ze(E) to replace R134a where the performance of baseline system operating with R134a has been accurately predicted and then the same platform is used to predict the key performance parameters for replacement refrigerants. Additionally, a parametric analysis has been conducted to predict the impact of key design parameters on the COP and UEF of the HPWH. Since the thermophyscial properties of proposed substitute are comparable to the R134a, the hypothesis of no extensive modification to existing system is evaluated to establish the drop-in-replacement potential for HFOs.

2. ANALYSIS STRATEGY

Modeling a HPWH system has been a challenge due to the varying complexities of the subsystems and the integration of the heat pump and water tank. The DOE/ORNL Heat Pump Design Model (HPDM) has been a reliable, public-domain platform for designing, optimizing, and analyzing heat pumps of varying complexity for both residential and commercial applications. The platform was used to develop a HPWH system model that included a wrapped tank condenser and accounted for features such as thermal stratification, piston effect, and mixing. The model was calibrated against measured test data and was used to evaluate HPWH performance with a range of refrigerants.

The current study draws upon some component modeling aspects of a previous ORNL HPWH analysis for forced-flow designs (Baxter *et al.,* 2011). Some of the key HPDM features important to the current study are described here. AHRI 10-coefficient compressor maps have been used to calculate mass flow rate and power consumption as shown by Eqs. (1) and (2), respectively, as function of evaporation and condensation temperatures (ANSI/AHRI 2010).

$$\dot{m} = \alpha_1 + \alpha_2 T_{evap} + \alpha_3 T_{cond} + \alpha_4 T_{evap}^2 + \alpha_5 T_{evap} T_{cond} + \alpha_6 T_{cond}^2 + \alpha_7 T_{evap}^3 + \alpha_8 T_{evap}^2 T_{cond} + \alpha_9 T_{cond}^2 T_{evap} + \alpha_{10} T_{cond}^3 \quad (1)$$

$$\dot{W} = \beta_1 + \beta_2 T_{evap} + \beta_3 T_{cond} + \beta_4 T_{evap}^2 + \beta_5 T_{evap} T_{cond} + \beta_6 T_{cond}^2 + \beta_7 T_{evap}^3 + \beta_8 T_{evap}^2 T_{cond} + \beta_9 T_{cond}^2 T_{evap} + \beta_{10} T_{cond}^3 \quad (2)$$

where T_{evap} and T_{cond} are the compressor suction and discharge saturation temperatures, and α and β are the compressor map coefficients for mass flow rate and power respectively. $\alpha_{1,2,\ldots,10}$ and $\beta_{1,2,\ldots,10}$ are the mass flow rate and power coefficients respectively for compressor. The 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. In addition to the functionalities of the segment-to-segment fin-tube condenser model, the evaporator model can simulate the dehumidification process. 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's two-phase heat transfer coefficient. The tank model includes the thermal conductivity of the thermal paste (used to ensure good contact between the wrapped tank condenser tubes and the tank wall) and the insulation covering the tank. Thus, heat loss from the tank was captured for the full time of the operation. The transient tank model accounts for one-dimensional water temperature stratification due to natural convention. Fig. 2 shows an example case (i.e., mixing caused by advection during water draw) for the CFD model used for the analysis. Additional details can be found in Elatar *et al.* (2017), Nawaz *et al.*, (2017, 2018) and Shen *et al.*, (2018).

A residential HPWH was considered as the baseline water-heating unit for performance evaluation and model calibration as shown in Fig 3. The performance of this unit has been evaluated previously at ORNL (Murphy *et al.*, 2011).

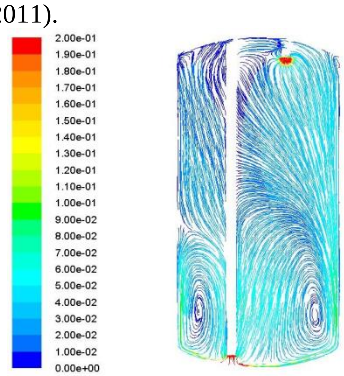

Fig. 2. Streamline for water flow in the tank during draw (scale shows velocity in m/s)

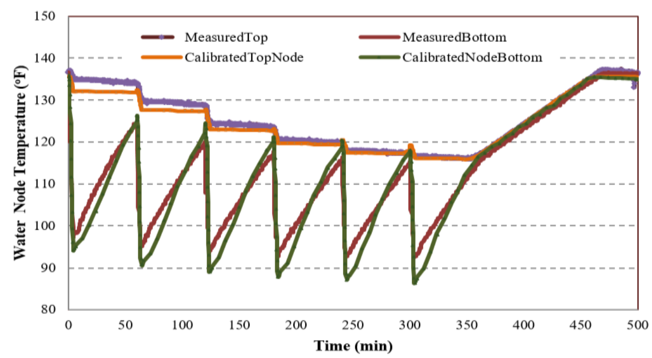

Fig. 3. Temperature stratification variation (measured vs. predicted behavior)

3. PERFORMANCE EVALUATION CRITERIA

The US Department of Energy (DOE) initiated a rulemaking to consider amendments to its old test procedures for covered residential and commercial water heaters, as recommended in the American Energy Manufacturing and Technical Corrections Act. Based on the extensive testing, a new performance evaluation procedure was defined. In the new procedure, the tank water temperature set point is 125°F, and the test conditions for inlet water and ambient air temperature are prescribed as 58°F and 67.5°F (35–45% relative humidity), respectively. Key performance metrics used to evaluate and compare the performance of water heaters (WHs), including HPWHs, are listed below.

First Hour Rating (FHR) is a measure of the available hot water capacity of the WH (in gallons). Per the new DOE test method, hot water (125±15°F) is drawn from the tank as long as the temperature is more than 67±2°F higher than the entering water temperature. Once the temperature drops below the prescribed limit, the supply is stopped until the set point of 125±1.5°F is met again. Following the procedure, the total water drawn from the tank during 1 hour indicates the total capacity of the heat pump and electric resistance heaters.

Unified Energy Factor (UEF) is a measure of system efficiency. It is defined as the ratio of the total heat delivered from the system (by heating the water) to the total power required to operate the system. Equation (3) describes the UEF.

$$UEF = \sum_{k=1}^{n} \frac{M_k c_p (T_s - T_i)}{W_i} \quad (3)$$

In Eq. (3) k represents the individual hot water draw, considering that multiple draws are required by the test procedure and that n (total number of draws) can vary according to the method proposed by DOE (small, medium, and large usage pattern), and M is the total mass drawn for each respective draw. T_s, T_i, and W_i represent the supply water temperature, inlet water temperature, and total energy consumed by the unit (power times unit run time), respectively. 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 5 hours of the EF test—and applied it to all WHs (including HPWHs) with a storage tank. In contrast, the new method uses the measured FHR value to define the hot water draw pattern. Table 2 provides the details of the draw pattern for a storage water heater based on the FHR.

Table 2. Water draw pattern based on FHR

FHR greater or equal to (gal)	FHR less than (gal)	Draw pattern for 24-h UEF
0	20	Point of use
20	55	Low usage
55	80	Medium usage
80	Max	High usage

The FHR analysis concluded that under all parametric conditions, the appropriate draw pattern for the HPWH unit being analyzed was for medium usage (FHR varied between 57 and 64 gallons). Table 3 presents the water draw pattern for a medium usage storage tank. The specified water draw pattern was used to determine the UEF.

Table 3. Medium water draw procedure

Draw Number	Time During Test (hh:mm)	Volume (gal (L))	Flow Rate (GPM (LPM))
1	00:00	15.0 (56.8)	1.7 (6.5)
2	00:30	2.0 (7.6)	1 (3.8)
3	01:40	9.0 (34.1)	1.7 (6.5)
4	10:30	9.0 (34.1)	1.7 (6.5)
5	11:30	5.0 (18.9)	1.7 (6.5)
6	12:00	1.0 (3.8)	1 (3.8)
7	12:45	1.0 (3.8)	1 (3.8)
8	12:50	1.0 (3.8)	1 (3.8)
9	16:00	1.0 (3.8)	1 (3.8)
10	16:15	2.0 (7.6)	1 (3.8)
11	16:45	2.0 (7.6)	1.7 (6.5)
12	17:00	7.0 (26.5)	1.7 (6.5)
Total volume drawn per day: 55 gal (208 L)			

4. PARAMETERS FOR PERFORMANCE EVALUATION

To optimize system performance, a parametric study was conducted to investigate the impacts of the condenser wrap pattern and the heat loss from the water storage tank. Two representative insulation effectiveness values were considered for the heat loss from the tank, 90% and 95%. The effectiveness is 90% when the tank loses 10% of the energy input to the water. The effectiveness is 95% when half that amount of energy is lost through the tank wall and insulation material to the environment. This essentially accounts for two values for the resistance to heat loss. Similarly, to explore the impact of condenser configuration, two different wrap patterns were considered for analysis as shown in Fig. 4. The counterflow pattern represents the flow of the refrigerant entering from the top section of the tank and moving downwards. In the parallel-counterflow configuration, the refrigerant enters close the middle of the tank, moves upwards, and then comes back to the middle section to continue downwards (identical to the pattern used by the prototype systems evaluated in the prior experimental study). In both cases, the refrigerant enters the system as superheated vapor, goes through a phase-change process, and exits as a subcooled liquid. Table 4 summarizes the cases considered in current study.

Table 4. Simulation cases for parametric analysis

Case number	Wrap pattern	Tank insulation effectiveness (%)
1	Parallel-counterflow	90
2	Parallel-counterflow	95
3	Counterflow	90
4	Counterflow	95

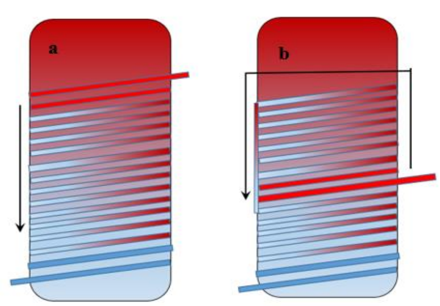

Fig. 4. Condenser wrap configurations: (a) counterflow, (b) parallel-counterflow.

5. DISCUSSION OF RESULTS

The first critical step for the performance analysis was to establish the first hour rating (FHR) as this information determines the water draw pattern and both COP and UEF factors depend on that. Figure 6 presents the FHR for different refrigerants at different design parameters. It's obvious that for all different cases the FHR stayed between 59 and 64 gallons. This made the situation easier as according to Table 2 all cases suggested that a medium usage draw should be used for further analysis.

As mentioned above, FHR is a direct indication of the system capacity. A larger FHR indicates a higher system water heating capacity. It can be observed from the analysis that R1234yf consistently shows a comparable FHR when compared with baseline R134a (values between 61 and 63 gallons), whereas R1234ze(E) has a somewhat lower average FHR closer to 60 gallons. Overall evaporator size and the condenser tube size are dominant factors but the response of different refrigerants to these parameters varies. Regardless of the trivial difference, it can be observed that both HFOs showed comparable performance to the baseline system containing R134a. Relatively smaller values for R1234ze(E) can be attributed to the lower volumetric capacity of the refrigerant (Table 1).

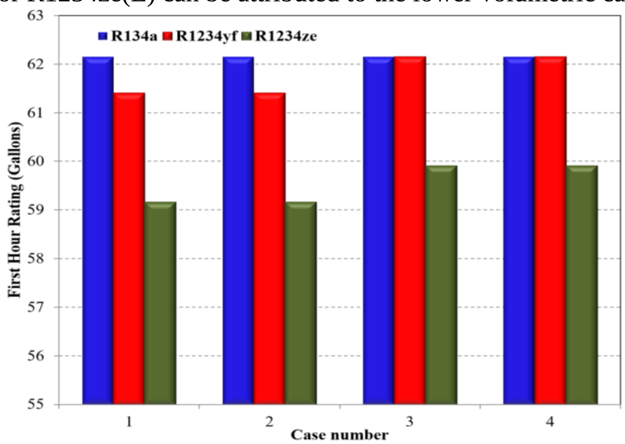

Figure 6. First Hour Rating for different refrigerants with varying design options

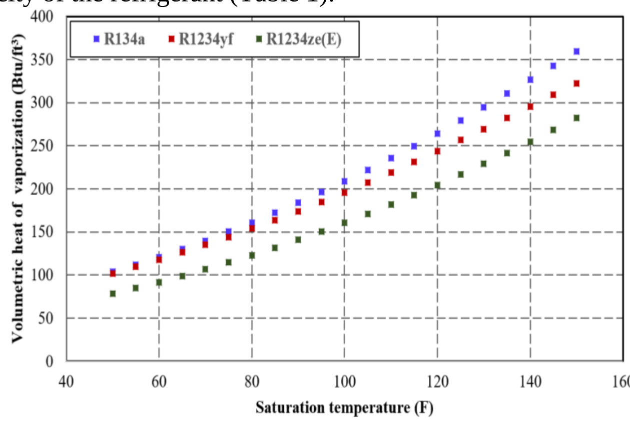

Figure 7. Volumetric heat of vaporization at different saturation temperatures

The relative comparison of the volumetric heat of vaporization (volumetric capacity) vs. saturation temperature is presented in Figure 7. As mentioned above among three refrigerants, R1234ze(E) has the lowest heat of vaporization. On average the values are about 25% and 37% lower over the operation range. Due to the significantly smaller capacity an R1234ze(E) system is expected to have the lowest FHR as presented in Figure 5.

Unified Energy Factor (UEF) is a direct measure of the system efficiency. A larger UEF denotes a better performance. Figure 8 presents the 24 hour UEF for the three refrigerants as the design parameters vary. It's important to distinguish between UEF and EF. While EF is the Energy Factor based on pre-2015 performance evaluation criteria (6 equal draws separated by one hour), the UEF is the Unified Energy Factor calculated per the new test procedure according to the draw pattern indicated in Table 3. It is obvious that design parameters play a critical role and UEF can change based on the selection of wrap pattern, evaporator size and condenser tube size. There is a wide range of calculated UEFs varying from 2.97 for R1234ze(E) in Case 2 to 3.64 for R134a in Case 7. A 0.5-inch diameter parallel-counter wrap condenser with smaller evaporator and 90% tank insulation effectiveness results in the least UEF whereas a 0.31-inch diameter counter wrap condenser with larger evaporator and 95% tank insulation effectiveness provides the best UEF.

Another important observation is that even though the relative values for UEF for different refrigerants don't vary much, R1234ze(E) shows the consistently lowest UEF in all parametric cases considered in the study.

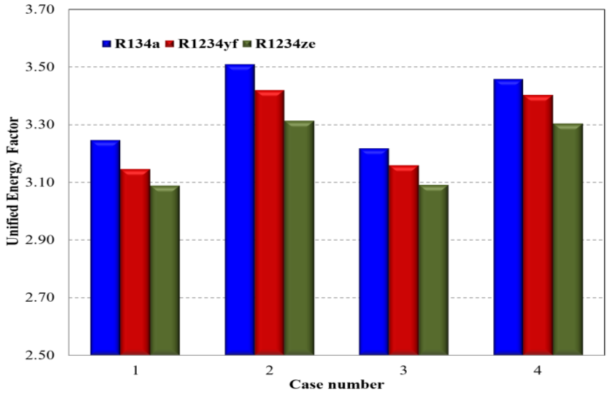

Figure 8. Unified Energy Factor (UEF) for different refrigerants with varying design options

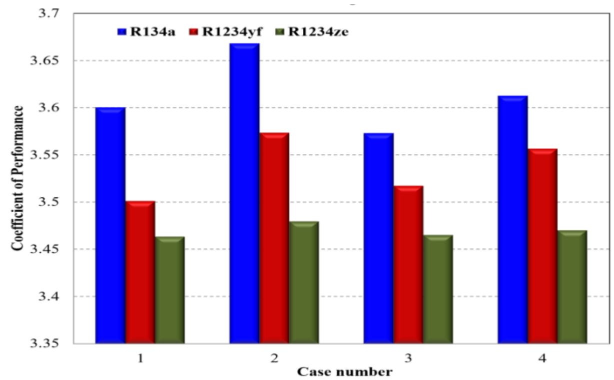

Figure 9. Coefficient of Performance (COP) for different refrigerants with varying design options

Average COP is typical parameter used to describe the efficiency of heat pump. It is important to recognize that COP represents the efficiency of heat pump only and can't be used to indicate the performance of whole system. However, this is still an important parameter to consider as it shows the stand-alone performance of heat pump. Figure 9 compares the COP for different design options. The performance somewhat follows the same trend as for UEF. The best COP is obtained for parallel-counter wrap pattern with larger evaporator and 0.31-inch condenser tube diameter with 95% tank insulation effectiveness. When the individual performance of different refrigerants is compared for a specific design option, it is obvious that the COP is comparable to the baseline except one case when counter wrap pattern is deployed with larger evaporator and 0.5-inch condenser tube diameter.

It's also important to note the relative difference between COPs for same design options (evaporator size, wrap pattern and condenser tube diameter) and different insulation effectiveness. A relatively lower COP for 90% insulation effectiveness is associated with relatively hotter supply water. A higher supply water temperature for 90% thermal insulation (Figure 10) is predicted because the higher tank loss leads to a longer heat pump run time. The heat pump run time has a direct relationship with the supply water temperature where relatively longer run time results in increase in the supply water temperature.

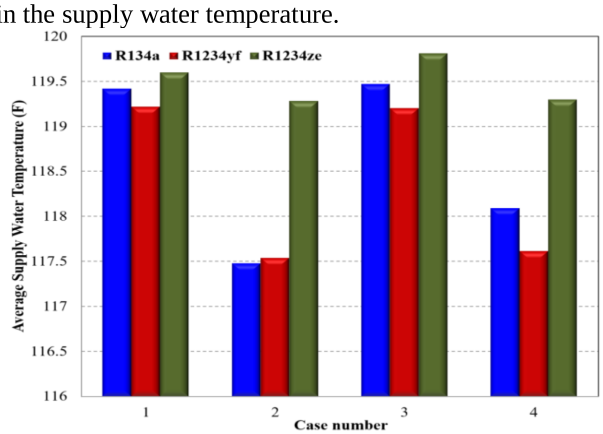

Figure 10. Average supply water temperature for different refrigerants with varying design options

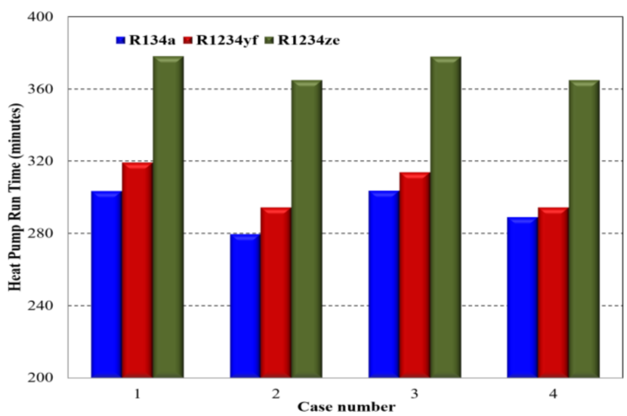

Figure 11. Heat pump run time for different refrigerants with varying design options

Figure 11 presents the total heat pump run time during a 24-hour UEF test. For any design selection R1234ze(E) has the largest run time. Again, this can be associated to the characteristics of the refrigerant (volumetric capacity) which causes on average 25-30% longer run time when compared with R134a and R1234yf.

For drop-in-replacement it is desirable that the proposed replacement refrigerants have a comparable and preferably lower compressor discharge temperature than that of the baseline R134a as this ensure that a similar compressor with existing lubrication can be safely used. Figure 12 compares the discharge temperature of the three refrigerants and indicates that the maximum compressor discharge temperature is about 25-40F lower for R1234yf and about 15-30 lower for R1234ze(E) compared to the baseline line refrigerant (R134a). This suggests that an easy substitution is possible for R1234yf. However, since volumetric capacity of R1234ze(E) is lower, a relatively larger compressor is required for comparable performance i.e. COP and UEF.

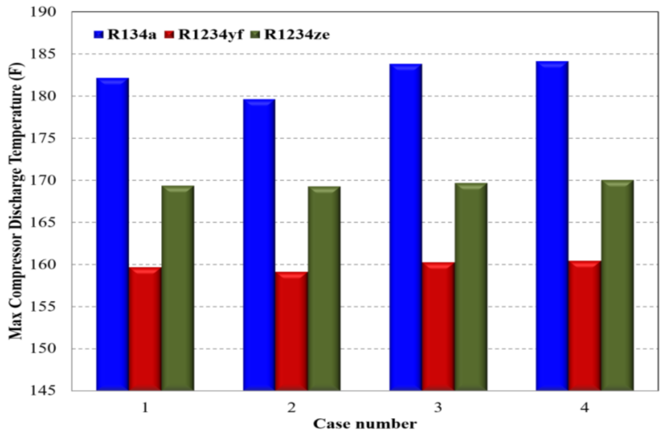

Figure 12. Max compressor discharge temperature for different refrigerants with varying design options

6. CONCLUSIONS

A simulation study was conducted for an HPWH application to evaluate the feasibility of using R1234yf or R1234ze(E) as a substitute for R-134a. A model was developed to account for the impact of a range of heat pump and water tank design parameters. It was found that

1. R1234yf and R1234ze(E) had comparable FHR and UEF values to those of R134a. A slightly lower performance was attributed to the relatively lower volumetric capacity of R1234ze(E). This can be resolved by increasing the size of the compressor.
2. The compressor discharge temperature was lower for R1234yf and R1234ze(E) than for R134a. Therefore, a compressor of similar design can be used for replacement refrigerants, and using the existing compressor lubricants is feasible.
3. Thermal stratification was found to be desirable for improved performance; however, it often occurred as a result of a longer heat pump run time, which degraded the performance of the system.
4. Considering the various performance parameters, both R1234yf and R1234ze(E) can be used for HPWH applications. No changes or minimal changes to the baseline R134a system would be required for R1234yf. A somewhat larger compressor displacement would be needed for R1234ze(E) to achieve a similar water heating capacity to R134a.

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