DROP-IN EVALUATION OF REFRIGERANT 1234YF IN A RESIDENTIAL INTEGRAL HEAT PUMP WATER HEATER

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Abstract: This paper describes a laboratory evaluation of the relative performance of a residential integral electric heat pump water heater operating with two working fluids: (1) Refrigerant 134a (R-134a), the standard design working fluid, and (2) Refrigerant 1234yf (R-1234yf), a potential replacement option with similar thermodynamic and transport properties, but with a substantially lower estimated global warming potential. The US 24-Hour Simulated Use and First-Hour Rating Tests were performed on the unit employing the nameplate charge of R-134a. After replacement of the R-134a by R-1234yf (with no changes to the hardware or software), heat-up tests were conducted to determine the appropriate R-1234yf charge. Finally, the 24-hour Simulated Use and First-Hour Rating Tests were repeated with R-1234yf. The first-hour rating calculated from the data for the R-1234yf case was found to be the same as that for the R-134a case, but the energy factor for the R-1234yf case was about 6% lower than that for the R-134a on this ”drop-in” basis. It is suggested that modification of the thermostatic expansion valve would allow a closer approach to the R-134a system energy factor.

Key Words: heat pump water heater, low GWP refrigerants, drop-in tests, energy factor, first-hour rating

1 INTRODUCTION

The hydrofluoroolefin (HFO) compound R-1234yf has been suggested (Calm 2008) as a near drop-in replacement for the hydrofluorocarbon (HFC) compound R-134a because the two compounds have very similar thermodynamic and transport properties, based on the REFPROP database (Lemmon et al. 2007)-- especially for mobile air conditioning situations. Although R-1234yf is mildly flammable, it has an estimated global warming potential (GWP) of 4 (over a 100-year time horizon on a scale normalized to GWP=1 for carbon dioxide), substantially lower than the corresponding GWP of 1410 attributed to R-134a. Therefore, it has been suggested that substitution of R-1234yf for R-134a in existing equipment designs might be a viable method of reducing global warming concerns. If refrigerant is

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(unintentionally) released from such equipment to the atmosphere, the resultant direct global warming impact would be reduced by the substitution of R-1234yf for R-134a. However, if such substitution resulted in any decrease in equipment energy efficiency or capacity performance under normal operating conditions, the attractiveness of R-1234yf as a potential working fluid replacement could be reduced. Lower energy efficiency for electrically powered equipment, including heat pump water heaters or other heating, ventilating, air-conditioning, or refrigeration (HVAC/R) systems, would lead to a larger indirect global warming contribution. Such an effect could be much larger over the expected life of the equipment than that from an unintentional release of the entire system refrigerant charge for most stationary HVAC/R systems. In addition, a reduction in capacity could reduce the acceptability of the energy-efficient equipment in the marketplace.

Several residential electric heat pump water heaters rated as "Energy Star" (http://www.energystar.gov) under U.S. Department of Energy regulations [first-hour rating (FHR) ≥ 50 gal (189 l) and energy factor (EF) ≥ 2.0, among other requirements] employ R-134a as their working fluid. The purpose of this study was to determine the efficiency and capacity performance of an unmodified production model of one of these units operating with R-1234yf relative to that with the baseline R-134a.

If performance is not significantly reduced, if safety and reliability impacts are negligible, and if the combined cost of the replacement refrigerant and any associated hardware or software modifications is not significantly greater than that of the current technology, then consideration of the alternative refrigerant based on global warming concerns may be justified. If, on the other hand, performance penalties are substantial, safety or reliability considerations are problematic, significant hardware or software alterations are required, or refrigerant costs are considerably higher, such an alternative may not be attractive.

2 BASELINE R-134A TESTING

The tested model employed a vapor compression system with a top-mounted compressor and evaporator/fan assembly, a thermostatic expansion valve, and a tank-wrapped condenser coil as the primary water-heating mechanism. It also incorporated a tank-mounted electrical resistance water-heating capability as a backup mechanism when called for by conditions. After verifying the nameplate charge of R-134a in the unit, First-Hour Rating Tests and 24-Hour Simulated Use Tests (United States Government 2010) were conducted to determine its baseline FHR and EF. The results very closely matched the corresponding Energy Star-listed values for this model.

3 REPLACEMENT R-1234YF CHARGE-DETERMINATION TESTING

3.1 Preparations

After leak-checking procedures and removal of the standard charge of R-134a (leaving the associated oil in the unit because of its reported compatibility with R-1234yf), the refrigerant system was evacuated to high vacuum with a vacuum pump. No hardware or software modifications were made to the unit to accommodate the alternate refrigerant. However, because the properties of R-1234yf were (similar, but) not identical to those of R-134a, a matrix of preliminary heat-up tests at selected charge levels was conducted over a range of ambient air temperatures (10.0 to 43.3°C) to establish the correct R-1234yf charge within the limits of the standard hardware and software. These tests employed the same supply water temperature and water flow rate specified for the 24-Hour Simulated Use Test (United States Government 2010), but were of shorter duration to accelerate screening efforts over the various charge levels and ambient air temperatures. Specifically, the heat-up test procedure

consisted of setting the water thermostat at its maximum set point, allowing the unit to heat the tank water to this set point temperature (thereby terminating the heating activity), conducting a water draw sufficient to activate the unit's upper electric resistance (backup) heating element, terminating the draw, and (following resistance element deactivation) allowing the unit's heat pump system to return the tank water to the set point temperature.

For the tests, time-scanned instrumentation was installed on the unit to provide not only temperature, humidity, power, flow, and pressure data consistent with the methods prescribed for the 24-Simulated Use Test (United States Government 2010), but also relevant refrigerant temperature and pressure data from the vapor compression system. From this data several parameters were deduced and employed as figures-of-merit to assess charge adequacy. The desired refrigerant evaporator exit superheat was that which would maximize effective heat transfer in the evaporator (maximizing evaporator saturation pressure and temperature) while minimizing the likelihood of refrigerant liquid entering the compressor under any condition over the entire system design operating range. The desired refrigerant condenser exit subcooling was that which would maximize effective heat transfer in the condenser (minimizing condenser saturation pressure and temperature) while minimizing the likelihood of refrigerant vapor entering the thermostatic expansion valve under any condition over the entire system design operating range. The desired recovery energy consumption and recovery time were the minimum values required to raise the average tank water temperature from a fixed lower intermediate value to a fixed higher intermediate value. Such recovery characteristics were intended to reflect energy efficiency and capacity attributes that would ultimately be realized in the EF and FHR determinations from the more protracted rating tests.

The liquid volume of the standard R-134a mass charge was calculated, using the REFPROP database (Lemmon et al. 2007), at condenser operating conditions corresponding to the maximum water temperature setting. This value served as the beginning volume estimate for the appropriate R-1234yf charge. The corresponding mass charge estimate was determined from the liquid density of (the less-dense) R-1234yf at the same condensing temperature, again based on the REFPROP database (Lemmon et al. 2007). To avoid starting with an overcharged condition and excess recovery procedures, the initial charge of R-1234yf was chosen to be about 87% of the estimated appropriate mass charge

3.2 Tests

After the initial charge was weighed into the unit, heat-up tests were conducted at four different ambient air temperatures. As shown in Figure 1, the minimum evaporator exit superheat (in non-dimensionalized form relative to the R-134a baseline value at 10°C ambient air temperature) was encountered at the lowest ambient air temperature, but frost formation on the evaporator prevented proper operation at this condition. The minimum condenser exit subcooling (in non-dimensionalized form relative to the R-134a baseline value at 10°C ambient air temperature) at 10°C is shown in Figure 2 to be about 0.89. Both the frost formation and the relatively low subcooling level were interpreted to be indications of a refrigerant undercharge situation. Because of the frozen evaporator at the lowest ambient temperature with the starting charge, emphasis was placed on monitoring evaporator performance as refrigerant charge was gradually increased.

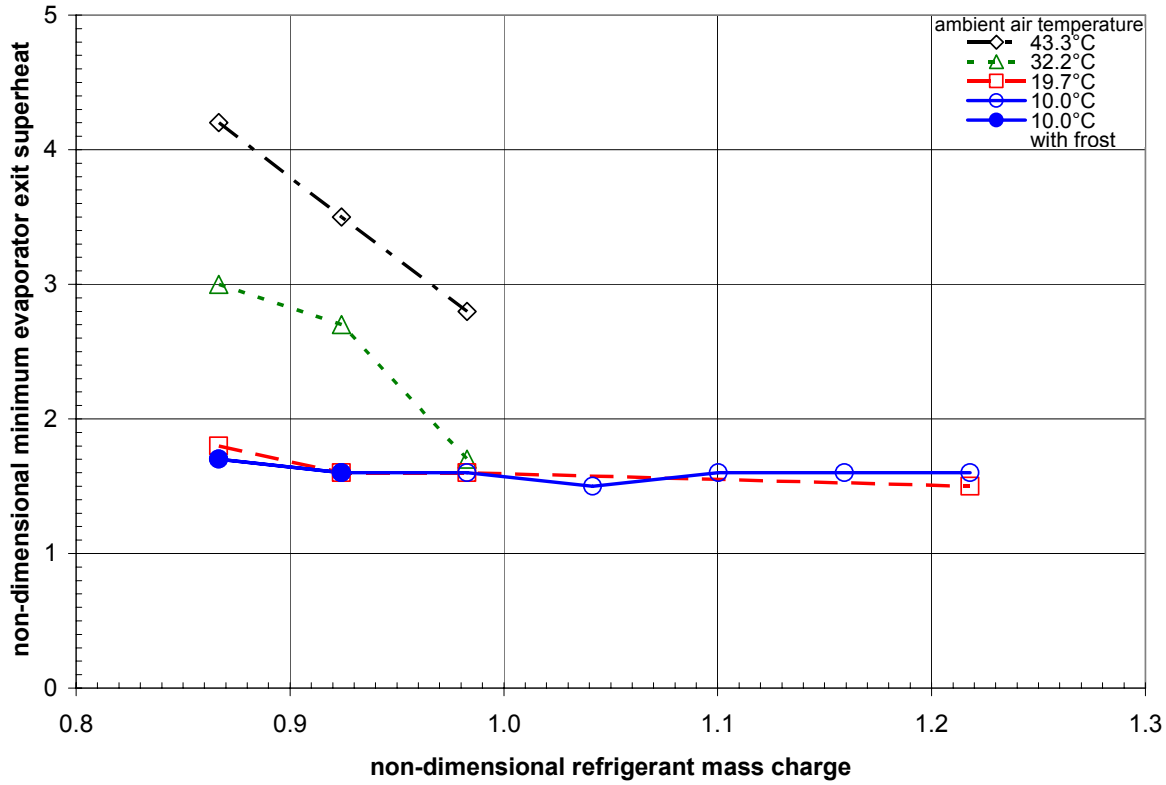


Figure 1: Evaporator superheat versus charge for various ambient air temperatures

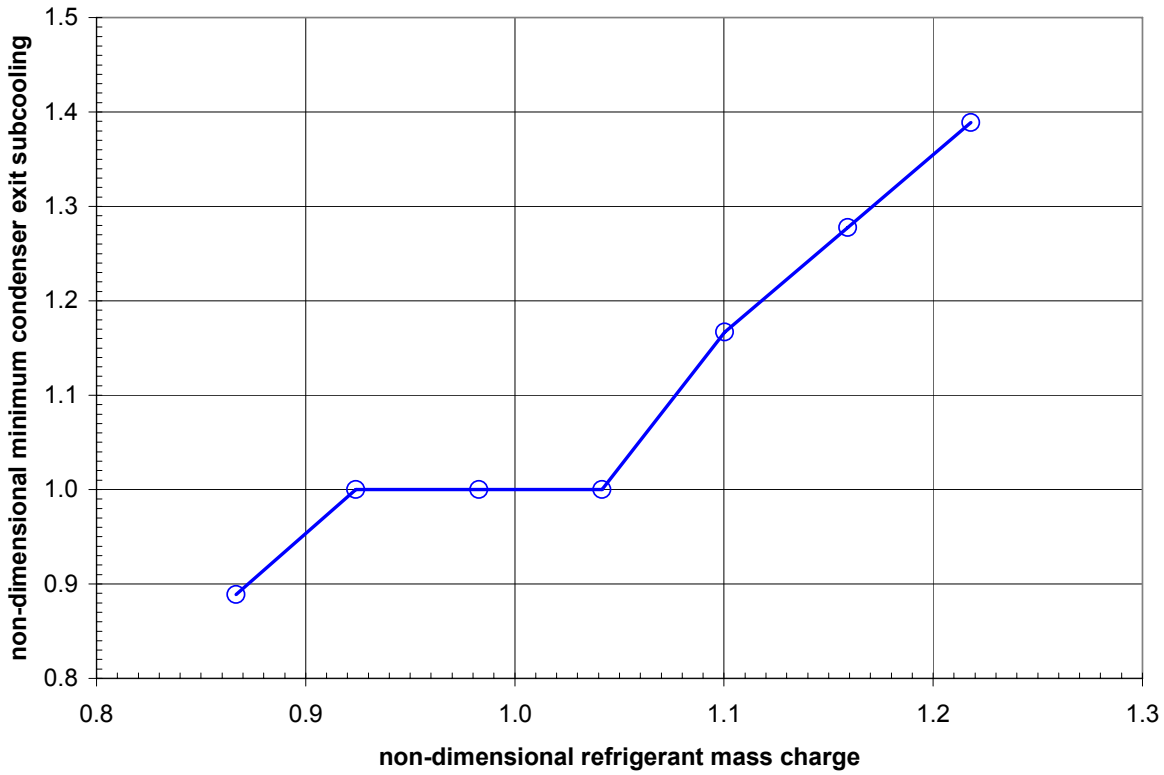


Figure 2: Condenser subcooling versus charge for 10°C ambient air temperature

The charge (represented on the abscissas of Figures 1 and 2 in non-dimensionalized form relative to the estimated appropriate mass charge of R-1234yf) was then increased to about 0.92, and the heat-up tests were repeated. Although superheat decreased and subcooling increased after the refrigerant addition as expected, frost was again observed on the evaporator surface at the lowest ambient air temperature condition. The next refrigerant increment brought the total relative charge to about 0.98, within 2% of the original estimate for the appropriate charge. The associated heat-up tests showed no frost formation on the evaporator, identifying this as the minimum acceptable charge.

As with the initial charge, the minimum evaporator exit superheat was achieved at the lowest ambient air temperature (10°C). However, under these conditions, although the subcooling value reached 100% of the corresponding value for the unit operating with the standard R-134a charge, the observed minimum evaporator exit superheat was about 60% above the corresponding R-134a system value. In addition, the normalized (relative to the R-134a baseline value at 10°C ambient air temperature) recovery energy consumption and recovery time, given in Figure 3, were approximately 14% and 15%, respectively, greater than the corresponding R-134a system values.

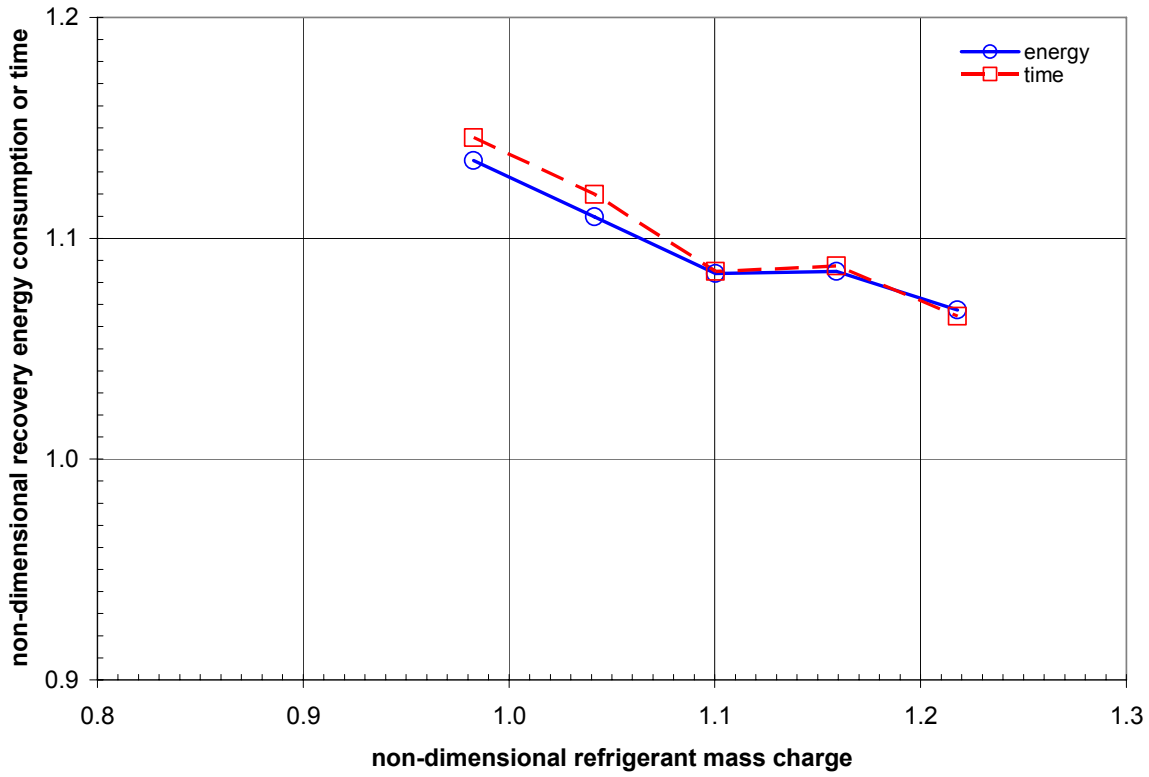


Figure 3: Recovery energy consumption/time versus charge for 10°C ambient air temperature

In order to possibly (a) reduce the evaporator superheat (and, consequently, improve its heat transfer effectiveness), (b) reduce the recovery energy consumption, and (c) reduce the recovery time, more refrigerant was added. Stepwise addition of R-1234yf (up to a total charge about 22% in excess of the original estimate for the "appropriate" charge) produced no significant change in the minimum evaporator exit superheat (see Figure 1), but did produce a significant increase (for non-dimensional charges above 1.04) in the minimum condenser exit subcooling (see Figure 2) to about 39% above the corresponding R-134a system value. As shown in Figure 3, the heat pump recovery energy consumption and time decreased somewhat as the non-dimensional charge increased from 0.98 to 1.10, but their

rate of decrease was much lower in the charge range from 1.10 to 1.22--ending with about 107% and 106% of the R-134a system values, respectively, at the highest charge. The close proximity of the two lines in Figure 3 reflects the fact that the average power draw for the unit during the recovery period was almost identical for the R-1234yf-charged system as it was for the R-134-charged system.

3.3 Analysis

From these results, it was judged that further refrigerant charge increases were not likely to produce significant reductions in minimum evaporator exit superheat, recovery energy consumption, or recovery time. Because the unit employed a non-adjustable superheat type thermostatic valve as its expansion device, it was expected that this component was limiting the minimum evaporator exit superheat. Because this valve was intended for operation with the original working fluid, its bulb was charged with R-134a. We calculated the pressure differential across the valve diaphragm when the net force from this pressure differential just counteracted that from the spring/diaphragm assembly for the superheat and evaporating temperature determined in the baseline R-134a unit tests (where both the system and the thermostatic expansion valve bulb were charged with R-134a). This value represented the differential pressure required to provide a given controlled flow area within the valve orifice.

Then we employed this pressure differential to estimate the superheat that would provide the same differential (and, therefore, the same controlled flow area) in the same valve (bulb charged with R-134a) operating with an R-1234yf-charged refrigeration system at the same evaporating temperature. The calculations showed that the superheat estimated on this basis was about 32% greater than the value determined for the R-134a/R-134a baseline. Although this calculation addresses the variations in saturation pressures between the two working fluids, other properties also impact thermostatic expansion valve performance. In particular, the liquid density of R-1234yf is approximately 11% smaller than that of R-134a at the representative operating condensing temperature—indicating that, with all other things equal, less mass flow would be allowed through the valve (at the same controlled flow area) for the R-1234yf case than for the R-134a case, likely providing less complete wetting of the evaporator and greater superheat. Although complete and simultaneous accounting for all such effects was beyond the scope of this study, it is suggested that the remainder of the approximately 60% excess superheat with the R-1234yf-charged system over that of the R-134a-charged system observed in the present tests is likely associated with other property differences.

Similar calculations for the situation with R-1234yf both in the valve bulb and in the refrigeration system gave a corresponding superheat (with the same representative evaporating temperature) within 0.1% of the R-134a/R-134a base value. This result suggests that the same thermostatic expansion valve hardware might be successfully used with a R-1234yf-charged refrigeration system if the bulb working fluid were replaced with R-1234yf—although, to account for the somewhat different liquid density or other properties of R-1234yf relative to those of R-134a, the orifice size might also need to be changed.

4 PERFORMANCE TESTING

4.1 Results

To complete performance characterization of the unit with the standard hardware and software components, both a First-Hour Rating Test and a 24-Hour Simulated Use Test were conducted with the 122% R-1234yf charge. The resultant FHR was found to match the value previously determined for the same unit operating with the standard charge of R-134a. This result was expected since the primary determinants of FHR for this unit were the

characteristics associated with the water tank (volume, mixing characteristics, etc.) and the upper resistance heating element (location, power, activation period, etc.)—none of which were altered by the refrigerant replacement. The resultant EF was determined to be approximately 6% lower than the value determined previously for the same unit operating with the standard charge of R-134a.

Some operational parameters that differed during the 24-hour Simulated Use Test employing R-1234yf from those observed during the same test using R-134a were: evaporator exit superheat (higher), condenser exit subcooling (higher), evaporator temperatures (lower), compressor energy consumption (higher), and refrigeration system heating capacity (lower).

4.2 Interpretation

It is postulated that the mismatch between the R-134a-charged thermostatic expansion valve and the R-1234yf-charged refrigeration system caused underutilization of the evaporator (increased dry regions, increased exit superheats, depressed evaporator temperatures). It may also have led to a slightly overcharged condition in the condenser (added liquid leg backup and increased condenser exit subcooling) in the attempt to reduce the evaporator exit superheat. The combination of various effects likely led to increased compressor energy consumption and reduced refrigeration system heating capacity.

5 CONCLUSIONS

Simply replacing the standard working fluid R-134a with alternative working fluid R-1234yf in this unit reduced the associated EF by about 6%, but did not significantly affect the associated FHR. If, in addition, the thermostatic expansion valve bulb Refrigerant 134a working fluid were replaced by R-1234yf working fluid, the correct refrigeration system charge could probably be established so as to produce approximately the same evaporator exit superheat. However, it is possible that the size of orifice in the thermostatic expansion valve would also have to be changed to accommodate adequate flows of the lower liquid density R-1234yf working fluid. It appears that such measures may then allow the energy factor performance of the unit employing R-1234yf to more closely approach that of the standard unit charged with R-134a.

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