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Performance evaluation and optimization of lower GWP refrigerants in a residential heat pump

Jethro Medina[a*], Joshua Hughes[a]

[a]The Chemours Company, 201 Discovery Boulevard, Newark, DE 19713, United States

Abstract

R-410A is the prevailing fluid in stationary air conditioning and heat pump applications. Its global warming impact, however, has garnered attention for lower GWP options. Existing alternatives include R-32 and R-454B, which provide notable GWP reductions from R-410A (AR4 GWPs of 2088 vs. 675 and 465 respectively). Although R-454B may prove to be a long-term solution, system optimizations, such as variable speed compressor use and heat exchanger changes, enable the use of even lower GWP fluids.

This paper explores the performance of several low GWP refrigerants (<300 GWP) in stationary air conditioning and heating applications. A commercial, residential R-410A heat pump was modified, per OEM guidance, with a suitable variable speed compressor. The heat pump and corresponding evaporator were placed in a pair of environmental chambers. The evaluated refrigerants consisted of blended HFO-based compositions. Variable speed compressor use allowed each fluid to match R-410A air-side capacities.

Presented data will include the following: air-side capacities, energy efficiencies, compressor speeds, and operating parameters. These performance metrics and operating conditions will be compared to those of R-410A. Performance will consist of both air-conditioning and heating environmental conditions, following the guidelines of AHRI 210/240. These blends further extend the potential options of low GWP alternatives, utilizing optimization technologies such as a variable speed compressor.

Keywords: Residential heat pump; Performance; Refigerat; R-454A; R-454C; Low GWP;R-1234yf; HFO; Variable Speed Compressor

1. Introduction

The phase out of CFC and HCFC refrigerants, due to the Montreal Protocol, has been completed. Consequently, the transition to non-ozone depleting HFC refrigerants resulted in the continued use of high Global Warming Potential (GWP) fluids. With recent regulations, such as the Kigali Agreement to the Montreal Protocol, requiring the phase down of HFC refrigerants, came the advent of hydrofluo-olefin (HFO) refrigerants, such as R-1234yf. With a GWP of <1, R-1234yf and R-1234yf containing refrigerant blends offer significant GWP reductions relative to incumbent HFC refrigerants.

The residential air conditioning and heating space predominantly uses R-410A (GWP 2088) in heat pump systems. Refrigerant blends containing R-1234yf, such as R-454B (GWP 467), are approved for new equipment use and offer significant GWP reductions. In preparation for future regulations calling for further GWP reductions, additional refrigerant blends were studied in a residential heat pump system. Refrigerants R-454C (GWP 146) and R-454A (GWP 288) are binary refrigerant blends consisting of R-32 and R-1234yf. Both fluids are in commercial use in stationary freezer applications. They each have potential in residential heat pump systems with some system optimizations, such as single-speed compressor changes, variable speed compressor use, or heat exchanger adjustments.

2. Fluid Overview, System Architecture, and Thermodynamic Cycle Model Study

2.1. Fluid properties and test system

A comparison of the physical and chemical properties of several lower GWP alternatives to R-410A are shown below in Table 1. Fluids in the comparison include R-410A, R-454B, R-454A, and R-454C.

Table 1. Physical and chemical properties of R-410A and lower GWP alternatives

Fluid	R-410A	R-454B	R-454A	R-454C
Name	R-32/R-125	R-32/R-1234yf	R-32/R-1234yf	R-32/R-1234yf
Composition, wt. %	50.0/50.0	68.9/31.1	35.0/65.0	21.5/78.5
Molecular Weight	72.59	62.61	80.47	90.78
Normal Boiling Point, °C	-51.36	-49.49	-42.16	-37.75
Sat Vap Pres (kPa) @ 20 °C	1443	1323	1012	847
Sat Vap Pres (kPa) @ 60 °C	3834	3541	2793	2368
Critical Temperature, °C	71.3	78.1	81.7	85.7
GWP (AR5)	1924	467	238	146

The alternatives above offer significant GWP reduction relative to R-410A. Each are binary refrigerant blends of R-32 and R-1234yf at differing compositions. Saturated vapor pressures for R-454A and R-454C indicate lower operating pressures in a heat pump system. Normal boiling points of the two fluids indicate capacity differences, which can be mitigated with system optimizations.

Test system architecture consisted of a residential heat pump designed for R-410A, which was an 8.79 kW, 16 SEER ducted, split system heat pump. The single-speed compressor native to the system was replaced with a variable speed compressor with comparable system performance at 60 Hz. The POE 32 centistoke lubricant used for each refrigerant was identical to what was specified for R-410A use. Architectures for both the cooling and heating modes are shown below in Figure 1.

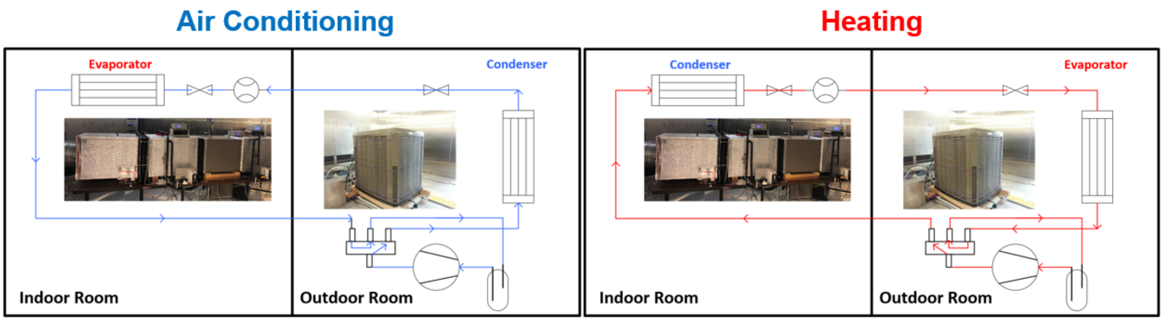

Fig. 1. Heat pump setup and schematic

The test heat pump was comprised of two heat exchangers, a variable speed scroll compressor, a mass flow meter, and a suction line accumulator. The indoor and outdoor room heat exchangers have an electronic expansion valve (EEV) and adjustable thermostatic expansion valve (TXV) respectively. A pair of environmental chambers was used to control the temperature and humidity of the indoor and outdoor units to applicable test conditions. Chamber test conditions are listed below in Table 2, which are compliant with ANSI/AHRI Standard 210/240-2023 and ISO Standard 5151-2017.

Table 2. Chamber test conditions for cooling and heating

Cooling/Heating	Condition	IR DBT ℃	IR WBT ℃	OR DBT ℃	OR WBT ℃
Air Conditioning	Cooling B*	26.7	19.4	27.8	18.3
Air Conditioning	Cooling A*	26.7	19.4	35.0	23.9
Air Conditioning	ISO T3**	29.0	19.0	46.0	24.0
Heating	H1*	21.1	15.6	8.3	6.1
Heating	H3*	21.1	15.6	-8.3	-9.4

* ANSI/AHRI 210/240-2023

** ISO Standard 5151-2017

Refrigerant charge optimizations were performed for each fluid at the Cooling A and H1 conditions for cooling and heating respectively. As the system was designed for R-410A, a singular charge optimization was performed for R-410A at the Cooling A condition. Optimized refrigerant charges were different between the modes for both R-454A and R-454C.

2.2. Thermodynamic cycle modelling

Thermodynamic cycle analyses were completed, comparing relative performance of R-454A and R-454C to R-410A at representative air conditioning and heating conditions. Cooling cycle model results are shown below in Table 3. Cycle conditions were as follows: 10 C evaporator, 46.1 C condenser, 11.1 K superheat, 8.3 K subcooling, 70% compressor efficiency, and equivalent compressor displacement. Heating cycle model results are below in Table 4, performed at the following conditions: 0 C evaporator, 55.0 C condenser, 10.0 K superheat, 3.0 K subcooling, and 70% compressor efficiency.

Table 3. Air conditioning thermodynamic cycle analyses

	R-410A	R-454B	R-454A	R-454C
Capacity vs. R-410A	-	-4%	-23%	-33%
COP vs. R-410A	-	2%	3%	5%
Evap Glide [K]	0.1	1.1	4.7	6.1
T, Dis [C]	81	87	77	73
P, Dis [kPa]	2801	2615	2131	1842

Table 4. Heating thermodynamic cycle analyses

	R-410A	**R-454B**	**R-454A**	**R-454C**
Capacity vs. R-410A	-	-1%	-24%	-36%
COP vs. R-410A	-	4%	3%	4%
Evap Glide [K]	0.1	1.0	4.0	4.7
T, Dis [C]	102	111	95	88
P, Dis [kPa]	3435	3207	2615	2261

For both cooling and heating conditions, R-454A and R-454C exhibited notably lower capacities than R-410A but similar COPs. Compressor discharge temperatures and pressures were decreased with both alterative refrigerant blends. Both R-454A and R-45C are zeotropic refrigerant blends, having higher temperature glides than R-410A.

Lower cooling and heating capacities indicate system optimization for use in a residential heat pump. For this work, a variable speed compressor, as described earlier, was installed to compensate for the lower capacities of R-454A and R-454C. By adjusting refrigerant mass flow rate, cooling and heating capacities can match those of R-410A. Increased compressor speeds will impact total system energy consumption, therefore, impacting efficiencies.

The test system was optimized with a comparable variable speed compressor. The inherent heat exchangers, designed for R-410A use, were unchanged. By optimizing the heat exchangers, the increase in heat transfer area will further account for lower fluid capacities and the increased pressure drop in the heat from lower density refrigerants. Additionally, optimized heat exchangers can better account for fluids with higher temperature glides and greater optimized refrigerant charge sizes relative to R-410A.

3. Results and Discussion

3.1 Air conditioning test results

As specified in the above section, performance evaluations were conducted in a split system residential heat pump designed for R-410A use. A variable speed compressor of similar performance at 60 Hz to the single speed compressor native to the commercial system was installed. The system was installed in a pair of environmental chambers. Testing was conducted following ASHRAE Standard 37, with chamber temperature and humidity set according to ANSI/AHRI Standard 210/240-2023 and ISO Standard 5151-2017, as shown previously in Table 2.

Refrigerant charge size optimizations were performed for both cooling (Cooling A) and heating (H1) modes. For the optimization, R-410A was run at 3600 RPM (60 Hz) to mimic the performance of a single speed compressor. The optimal refrigerant charge yielded the highest COP at the singular compressor speed with appropriate subcooling. The refrigerant charge optimizations for R-454A and R-454C involved adjusting compressor speed to match R-410A capacity at Cooling A and H1, depending on the selected mode. Table 5 displays the results for cooling mode, along with an expected refrigerant charge estimated from liquid densities of each refrigerant.

Table 5. Charge optimization – Air conditioning

Refrigerant	Optimized Charge kg	Relative to R-410A %	*Estimated Charge* *kg*	*Relative to R-410A* *%*
R-410A	5.22	0%	*5.22*	*0%*
R-454A	4.99	-4%	*5.02*	*-4%*
R-454C	5.67	9%	*5.12*	*-2%*

Optimized refrigerant charge for R-454A was similar to its estimated charge. The estimated refrigerant charge of R-454C was 2% less than R-410A. However, the optimized refrigerant charge was about 9% greater than that of R-410A. At the lower refrigerant charge, the refrigerant was not fully subcooled entering the expansion valve. The increased refrigerant charge provided the optimal COP and adequate subcooling.

R-454A and R-454C performance was evaluated twice at each condition specified in Table 2. One evaluation was with using a compressor speed of 3600 RPM and matching evaporator superheat of R-410A. The second evaluation had an adjusted compressor speed to match the capacity of R-410A at that temperature condition. Evaporator superheat was also adjusted to better match the evaporator temperature of R-410A for the matched capacity conditions. Figures 2 and 3 depict cooling capacity and COP, relative to R-410A.

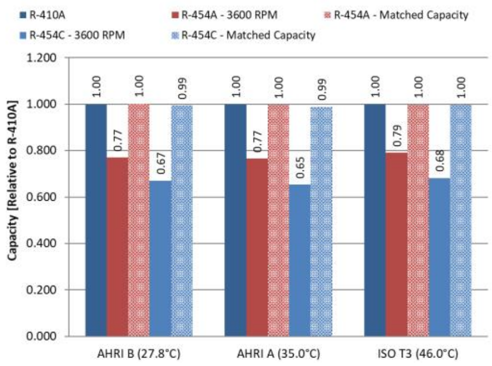

Fig 2. Cooling capacity relative to R-410A

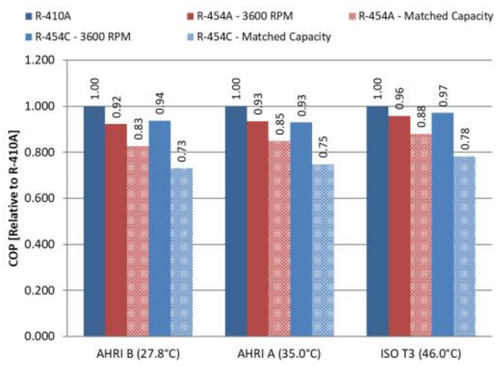

Fig 3. COP relative to R-410A

With compressor speed adjustment, the R-454A and R-454C cooling capacities were within 1% of R-410A capacities at the evaluated conditions. Cooling capacities for both fluids were similar to the thermodynamic cycle model when studied at 3600 RPM compressor speed. Relative COPs at the matched capacity conditions were lower than expected. The lower capacities of R-454A and R-454C required an increase in compressor speed, subsequently increasing energy consumption. The energy penalty is larger than expected considering the comparable COP of both fluids relative to R-410A from the thermodynamic cycle models. Discharge pressures for R-454A and R-454C in the matched capacity tests were higher than cycle models would suggest, indicating a potential penalty from high refrigerant charge. Particularly for R-454C, refrigerant charge was about 9% higher than R-410A with an expected charge being less than R-410A. Additional system optimization or using a system designed for R-454C would likely mitigate this issue. Energy and compressor speed increases are shown below in Figures 4 and 5.

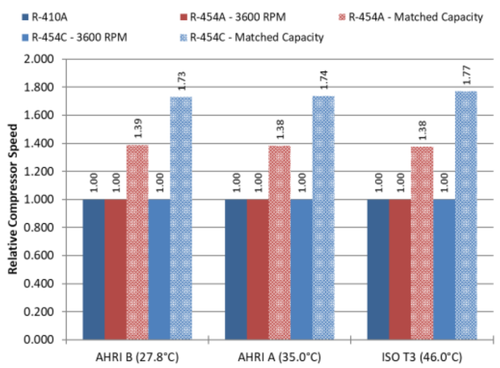

Fig. 4. Relative compressor speed

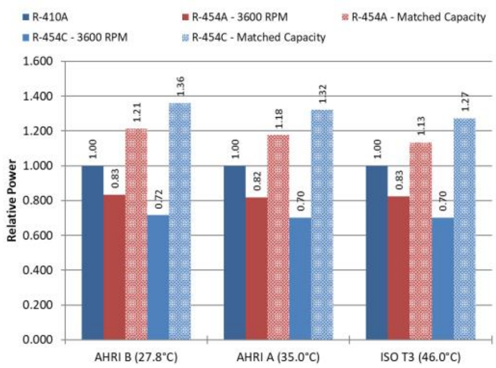

Fig. 5. Relative energy consumption

With lower volumetric capacities, compressor speed increases for R-454A and R-454C are shown above in Figure 4. From Figure 5, relative energy consumption for both fluids at 3600 RPM were about 20% and 30% lower than that of R-410A. Upon increasing speed to match R-410A capacity, increase in energy consumption was greater than the cooling capacity increase, resulting in lower relative COPs shown in Figure 3. This could also be mitigated with further system optimizations or by using a system designed for use with one of the alternative refrigerant blends. Figures 6 and 7 illustrate discharge temperature and refrigerant mass flow rate trends for the testing.

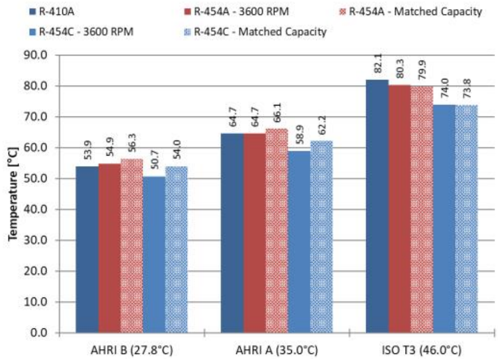

Fig. 6. Compressor Discharge Temperatures

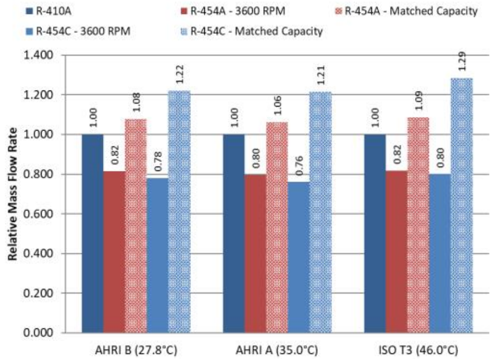

Fig. 7. Relative refrigerant mass flow rate

Compressor discharge temperatures follow the expected trend from cycle model evaluations. R-454C yielded the lowest discharge temperatures, followed by R-454A. The same trend was observed with refrigerant mass flow rate. R-454C relative mass flow rate was greatest due to its lower volumetric capacity among the fluids tested. R-454A and R-454C had flow rates about 8% and 24% higher than R-410A, which accounted for cooling capacity increases of about 23% and 33% respectively. Evaporator superheat and EEV inlet subcooling temperatures are below in Figures 8 and 9.

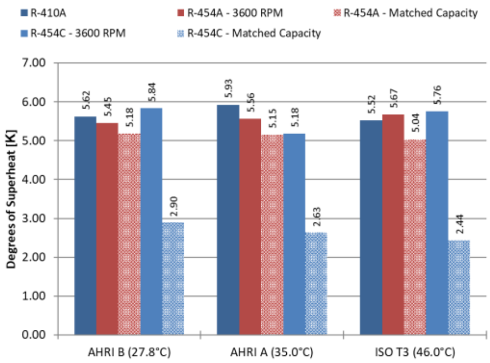

Fig. 8. Evaporator outlet superheats

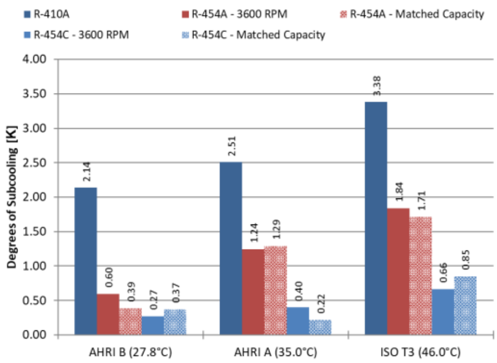

Fig. 9. EEV inlet subcooling

Evaporator superheat was maintained at about 10 F for the 3600 RPM evaluations and adjusted to match the average evaporator temperatures of R-410A at in the matched capacity studies. The higher temperature glide R-454C required more superheat adjustment to match R-410A evaporator temperature. Subcooling values for R-454A and R-454C, though low, were the measured values for the optimal refrigerant charges for both fluids. There is a sight glass on the system before the EEV to visually confirm phase of fluid entering the EEV. For each test, fully subcooled liquid was entering the EEV.

3.2 Heating test results

Performance evaluations in heating mode are in progress. Evaluations for R-454A and R-454C will be complete in 2023.

4. Conclusions

Lower GWP alternatives to R-410A in stationary applications are available in the form of HFO containing refrigerant blends, such as R-454B, and other solutions. The potential for further reducing global warming impact in such systems exists with options such as R-454A and R-454C. Such refrigerants will require optimizations to common designs, such as a variable speed compressor, to achieve comparable cooling capacities to the incumbent HFC. Evaluations in a R-410A residential heat pump showed the potential of even lower GWP alternatives. Comparable cooling capacity was achieved with R-454A and R-454C. Relative COPs can be improved with further system optimization or choosing a system better suited for use with the alternative refrigerants. The incurred energy penalties can be reduced with these changes. Refrigerant flow increase can compensate for lower capacity fluids, enabling further reductions in GWP.

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