2016

Multi-Year Evaluation of R-449A as a Replacement for R-22 in Low Temperature and Medium Temperature Refrigeration Applications

Andrew Pansulla
Chemours, United States of America, andrew.r.pansulla@chemours.com

Charles Allgood
Chemours, United States of America, charles.c.allgood-1@chemours.com

Multi-Year Evaluation of R-449A as a Replacement for R-22 in Low Temperature and Medium Temperature Refrigeration Applications

Andrew PANSULLA[1]*, Charles ALLGOOD[2]

[1] The Chemours Company, Fluorochemicals,
Wilmington, Delaware, United States of America
Tel. 302-683-8299, Fax. 302-683-8255, andrew.r.pansulla@chemours.com

[2] The Chemours Company, Fluorochemicals,
Wilmington, Delaware, United States of America
Tel. 302-683-8247, Fax. 302-683-8255, charles.c.allgood-1@chemours.com

* Corresponding Author

ABSTRACT

HCFC based refrigerants, such as R-22, are being phased out globally due to issues with stratospheric ozone depletion. In order to effectively do so, a series of HFC blend alternatives, such as R-407C and R-404A, have been introduced. While these refrigerants have no ozone depletion potential (ODP), they do occasionally have similar, or even higher, global warming potential (GWP) than the fluids they replaced. The global HVACR industry is now researching and developing a new generation of low GWP and zero ODP refrigerants that will possess similar operational characteristics and favorable energy efficiencies when compared to their HCFC predecessors.

Hydrofluoro olefins (HFOs) and HFO blends have recently emerged as a potential new class of low GWP, zero ODP refrigerant fluids. R-449A (XP40) is an HFO-based refrigerant blend that was initially developed as a replacement for high GWP HFC blends like R-404A/R-507, but now is being investigated as a potential R-22 alternative in refrigeration applications as well.

This paper will report the results of an extensive study investigating the performance of R-449A relative to R-22 in both low temperature and medium temperature refrigeration applications. Data presented includes results of constant compressor displacement thermodynamic cycle models, compressor calorimeter testing at various operating conditions, display case freezer testing in temperature and humidity controlled environmental chambers, and real world field system data from operating supermarkets in the United States. Operational and energy performance data for the various systems will be compared to baseline operation on R-22.

1. INTRODUCTION

R-22 has long been a staple of the refrigeration and air conditioning industries. However, its ozone depletion potential (ODP) has prompted the global refrigeration industry to look for non-ozone depleting alternatives. On January 1st, 2015, the United States EPA finalized the plan to completely phase out R-22 in the United States. This plan calls for the aggressive linear reduction in the amount of R-22 chemical manufacturers can produce each year. By the end of 2019, no new or imported virgin R-22 will be allowed in the United States (Powell, 2014). Owners of R-22 systems are now left with the choice of servicing their equipment with an existing and diminishing supply of R-22, installing a new system with an acceptable alternative refrigerant, or retrofitting the system to operate with a new refrigerant. While HFC blends have long been used at retrofit gases for R-22, their high GWP (in many cases higher than R-22) has led to the search for not only a non-ozone depleting refrigerant, but one that has low global warming potential.

Hydrofluoro olefins (HFOs) are a new class of refrigerants that have recently entered the refrigeration industry. Due to their chemical nature, HFOs have very low GWP and zero ODP. HFO molecules can be blended with existing HFCs, to develop a new class of low GWP refrigerants for a variety of applications. R-449A is an HFO-containing refr3igerant blend that was initially developed to replace R-404A. The composition of R-449A is 24.3 wt% R-32, 24.7 wt % R-125, 25.3 wt% R-1234yf, and 25.7 wt% R-134a. Despite being initially developed as a replacement for R-404A, results from thermodynamic modeling, calorimeter tests, and system test suggest that R-449A may work well as a replacement for R-22.

2. THERMODYNAMIC PROPERTIES AND CYCLE MODELS

A comparison of the basic physical properties of R-449A and R-22 are shown below in Table 1.

Table 1. Physical properties of R-449A and R-22

Refrigerant	**R-22**	**R-404A**	**R-449A**
Molecular Weight	86.46 g/mol	97.60 g/mol	87.2 g/mol
Boiling Point at 1 atm	-41.5 ºF (-40.8 ºC)	-49.8 ºF (-45.4 ºC)	-50.7 ºF (-46.0 ºC)
Critical Pressure	723.7 psia (4990 kPa [abs])	541.7 psia (3734.9 kpa [abs])	655.0 psia (4447 kPa [abs])
Critical Temperature	205.6 ºF (96.4 ºC)	161.82 ºF (72.1 ºC)	178.7 ºF (81.5 ºC)
Liquid Density at 70 ºF (21.1 ºC)	75.3 lb/ft^3 (1205.7 kg/m^3)	66.3 lb/ft^3 (1062.2 kg/m^3)	69.5 lb/ft^3 (1113.3 kg/m^3)
Ozone Depletion Potential (R-11 = 1.0) (UNEP, 2000)	0.055	0	0
Global Warming Potential (IPCC AR5, 2013)	1760	3922	1282

Refrigeration performance of R-449A versus R-22 was calculated using thermodynamic cycle models run at standard low and medium temperature evaporator conditions. A diagram of the model is shown in Figure 1 (NIST, 2015) below.

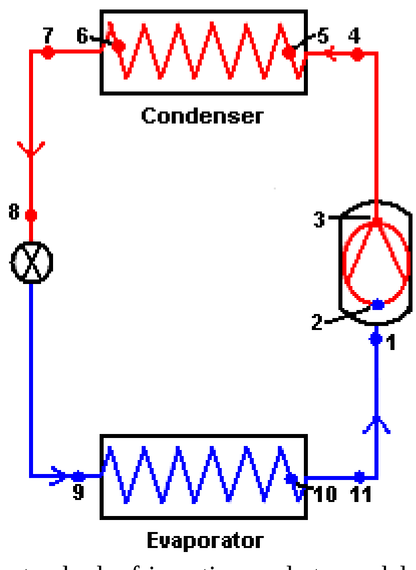

Figure 1: Schematic of a standard refrigeration cycle to model refrigerant performance

The low temperature conditions were as follows: -20 ºF (-28.9ºC) evaporator, 105 ºF (40.5 ºC) condenser, 40 ºF (4.4 ºC) return gas temperature and a 95 ºF (35 ºC) sub cooled temperature. In order to account for the temperature glide of the blend, comparisons were made at equal average condenser and evaporator temperatures for the two refrigerants. Also, the isentropic and volumetric efficiencies were assumed to be 70% and 90%, respectively. To better model a potential retrofit scenario, the same volumetric displacement was used. Tables 2 summarizes the low temperature thermodynamic model results.

Table 2. Low Temperature Thermodynamic Cycle Models (no liquid injection)

Refrigerant	**R-22**	**R-449A**
Suction Pressure	24.9 psia (171.7 kpa [abs])	26.4 psia (182.2 kpa [abs])
Discharge Pressure	225.4 psia (1554.4 kpa [abs])	258.9 psia (1785.5 kpa [abs])
Compressor Discharge Temperature	297.9 ºF (147.7 ºC)	250.6 ºF (121.4 ºC)
Relative Volumetric Capacity	1.00	0.96
Relative COP	1.00	0.90
Relative Mass Flow	1.00	1.07

Many R-22 low temperature refrigeration systems use liquid injection to lower the compressor discharge temperature. Table 3 below shows the impact of maintaining a maximum compressor discharge temperature of 275 ºF (135 ºC) using liquid injection for the R-22 system.

Table 3. Low Temperature Thermodynamic Cycle Models (with liquid injection)

Refrigerant	**R-22 (with Liquid Injection)**	**R-449A**
Suction Pressure	24.9 psia (171.7 kpa [abs])	26.4 psia (182.2 kpa [abs])
Discharge Pressure	225.4 psia (1554.4 kpa [abs])	258.9 psia (1785.5 kpa [abs])
Compressor Discharge Temperature	275 ºF (135 ºC)	250.6 ºF (121.4 ºC)
Relative Volumetric Capacity	1.00	1.00
Relative COP	1.00	0.96
Relative Mass Flow	1.00	1.14

For the medium temperature conditions, the evaporator temperature was raised to 20 ºF (-6.7 ºC) and the return gas temperature was increased to 50 ºF (10 ºC). The medium temperature condenser and sub cooled temperatures were unchanged from the low temperature model. The discharge temperatures of R-22 and R-449A are low enough that no liquid injection correction models are necessary. Table 4 below summarizes the medium temperature thermodynamic cycle results.

Table 4. Medium Temperature Thermodynamic Models of R-449A and R-22

Refrigerant	**R-22**	**R-449A**
Suction Pressure	57.8 psia (398.5 kpa [abs])	63.3 psia (436.4 kpa [abs])
Discharge Pressure	225.4 psia (1554.1 kpa [abs])	258.6 psia (1782.9 kpa [abs])
Compressor Discharge Temperature	209.2 ºF (98.4 ºC)	184.3 ºF (84.6 ºC)
Capacity	1.00	1.03
COP	1.00	0.94
Mass Flow	1.00	1.11

3. COMPRESSOR CALORIMETER TESTING

Calorimetry provides a way to accurately measure the capacity and COP of refrigerants by means of a heat balance. The suction and discharge pressures of the refrigerant can be set by the operator to test a variety of different refrigeration cycles. Electric heaters are used in the evaporator to heat up the refrigerant to a specified return gas temperature. Refrigerant charge is varied to give more or less sub cooling. The performance of R-449A was tested versus R-22 in a compressor calorimeter using a 3 ton semi hermetic compressor. The experimental set for the calorimeter were in accordance with ASHRAE standard 23.1. A picture of the calorimeter is shown in Figure 2.

Figure 2. Compressor calorimeter used to experimentally test R-449A versus R-22

All tests were run in a constant temperature compartment held at 95 ºF (35 ºC). The low evaporation temperature tests were run with an average evaporator temperature of -25 ºF (-31.6 ºC). Condenser temperatures were varied to determine the impact hotter condensers would have on the performance of the refrigerant. The refrigerant charge was varied so that there would always be 10 ºR (~5.5 ºK) of sub cooling from the average condensing temperature. Liquid injection was used to keep the maximum discharge temperature at 275 ºF (135 ºC). For the R-22 tests, 3GS mineral oil was used to lubricate the compressor. To test R-449A, POE RL32-3MAF oil was used. Figures 3 and 4 below shows the capacity and COP respectively of R-22 and R-449A as a function of the condenser temperature for two return gas temperatures (RGT).

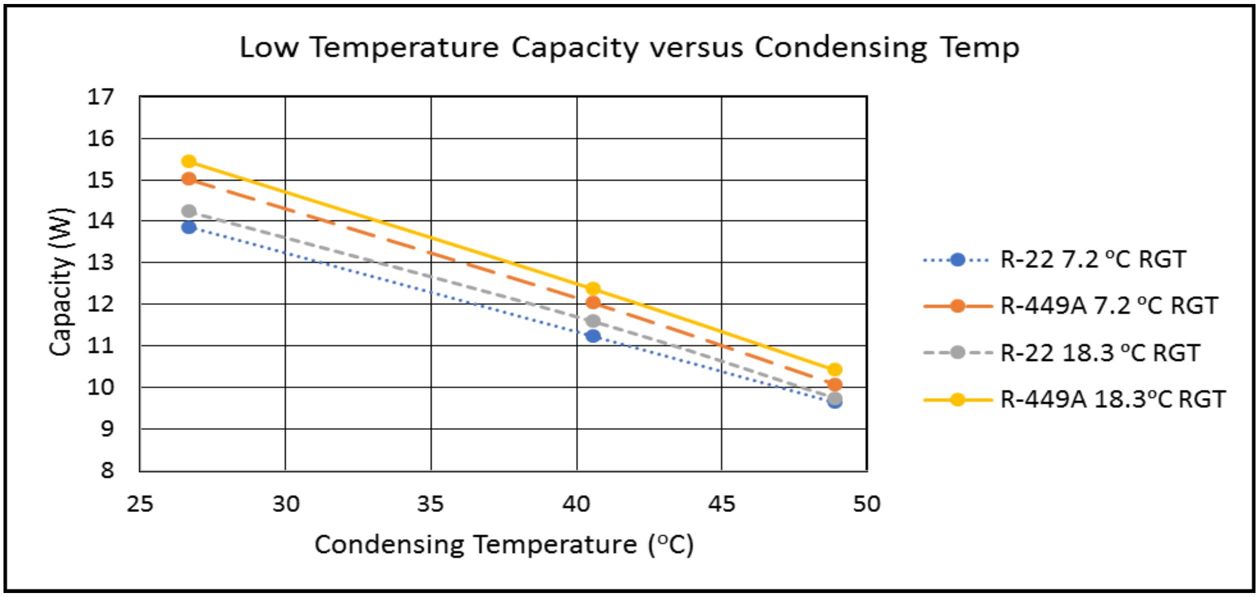

Figure 3. Low temp capacity as a function of the condensing temperature

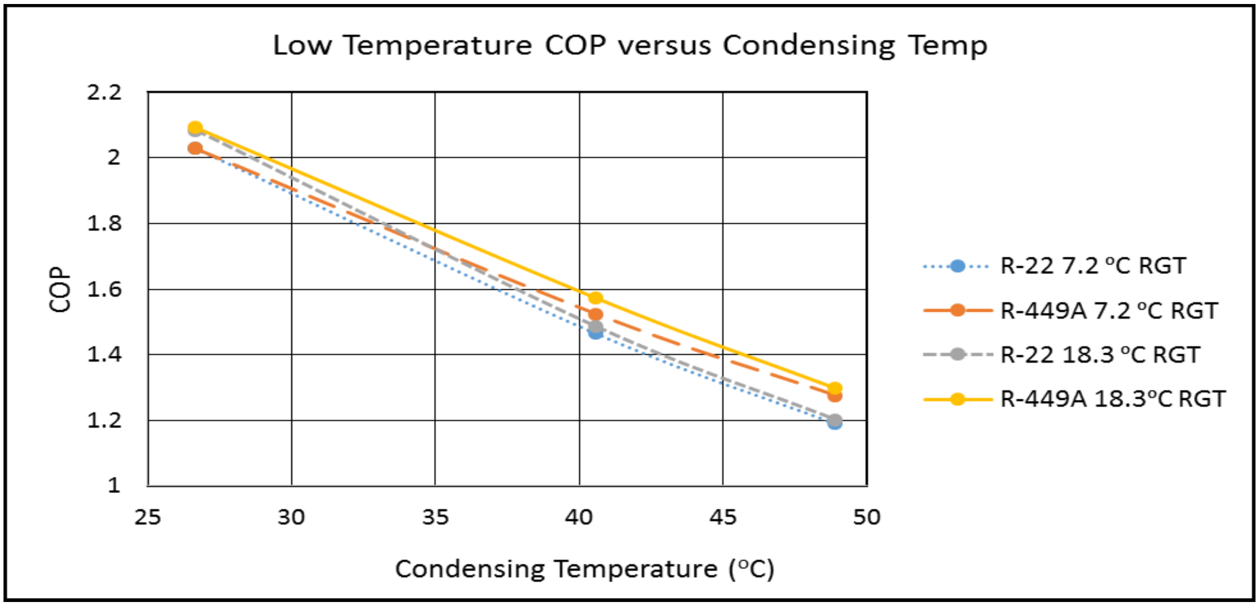

Figure 4. Low temp COP as a function of the condensing temperature

All 105 °F (40.5 °C) and 120 °F (48.9 °C) condensing temperatures for R-22 needed liquid injection as did the 120 °F (48.9 °C) condensing 65 °F (18.3 °C) return gas temperature test for R-449A.
Medium temperature tests were run at 20 °F (-6.7 °C). Like the low temperature tests, different condensing temperatures were examined to determine the impact warmer condensing temperatures have on the performance of the refrigerant. Figures 5 and 6 show the capacity and COP respectively.

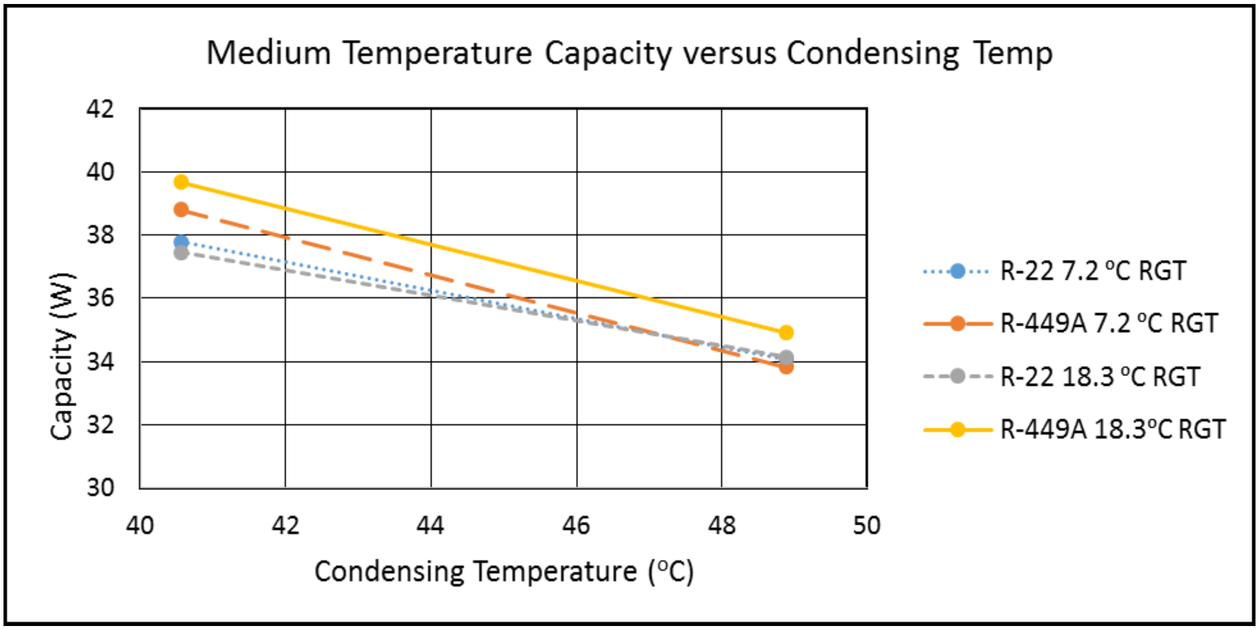

Figure 5. Medium temp capacity as a function of the condensing temperature

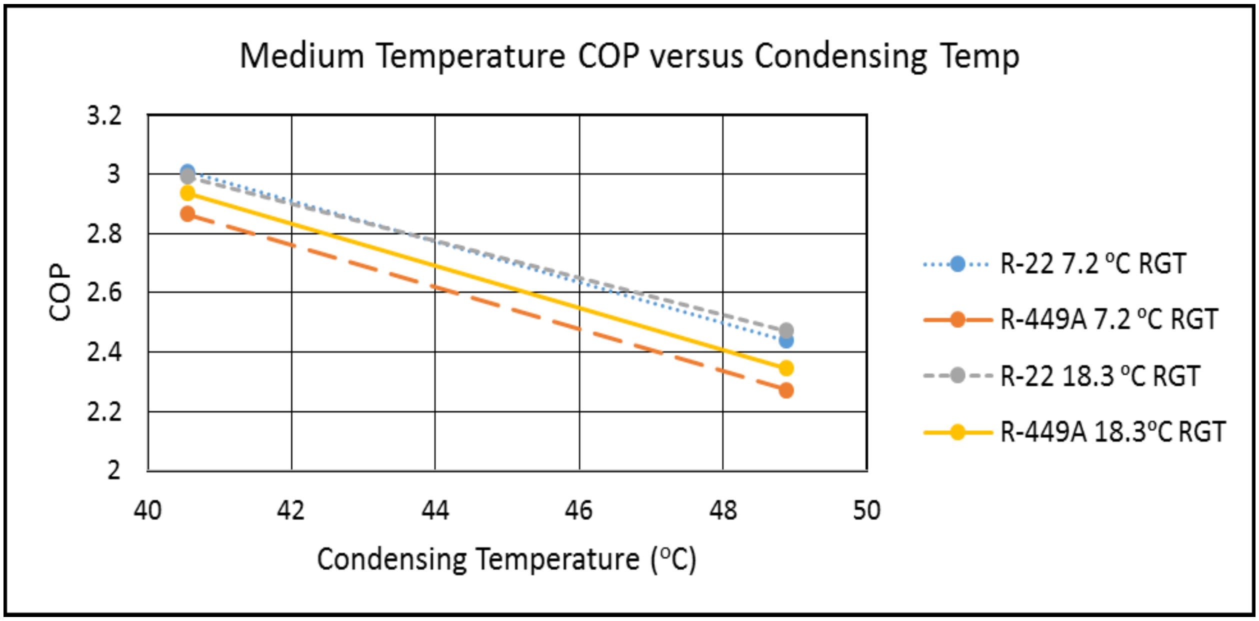

Figure 6. Medium temp COP as a function of the condensing temperature

4. CONTROLLED ENVIRONMENT SYSTEM TESTING

While compressor calorimeters provide an accurate picture for capacity and COP under highly controlled conditions, actual system testing provides a way to accurately quantify performance of a complete system that includes pressure drop, heat transfer and oil return effects. An open coffin case and a single condensing unit were placed in indoor and outdoor environmental chambers respectively. The experimental test set up of the system was in accordance with ASHRAE standard 72. Temperature and humidity of the test chambers are specified by ARI Standard 210/240. Pictures of the condensing unit and coffin case are below in Figure 7.

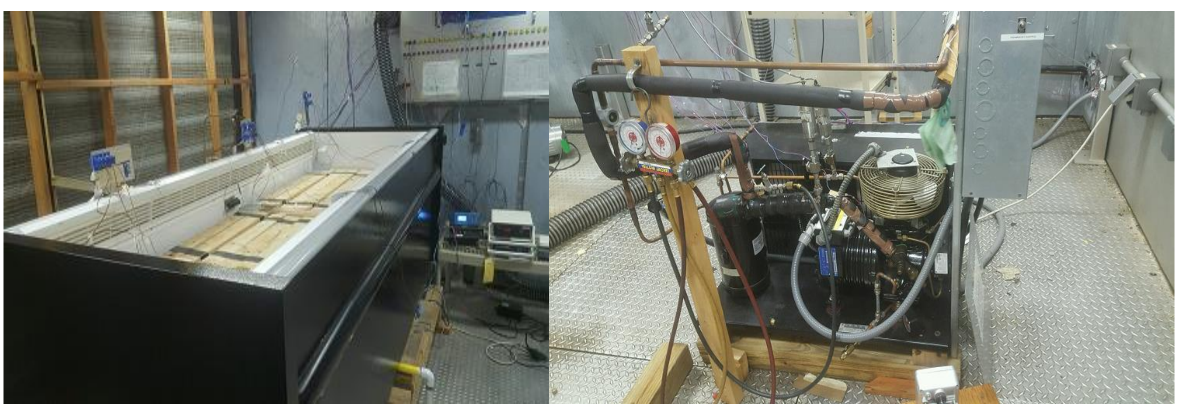

Figure 7. Experimental Setup for controlled environment system testing

The outdoor room housing the condenser and compressor were maintained at 82.0 ± 0.5 °F (27.8 ± 0.27 °C) with a dew point of 56.0 ± 0.5 °F (13.3 ± 0.27 °C). In the indoor room, the coffin case was exposed to an ambient temperature of 75 ± 0.5 °F (23.9 ± 0.27 °C) with a dew point of 58 ± 0.5 °F (14.4 ± 0.27 °C). All tests were run at the same average evaporator temperature. An EEV was used to regulate the amount of evaporator superheat.

With locations specified by ASHARE Standard 72, twelve tubs containing a 50/50 mixture (by volume) of propylene glycol and distilled water were placed in the coffin case. To simulate product mass, wood was placed

throughout the coffin case so that up to 90% of the net usable volume was occupied. Temperature and pressure readings were taken every six seconds for a 24 hour period. At each 12 hour mark, the system was defrosted. The thermostat was set so that the low temperature tub temperature during the last 3/4s of the running cycle in between defrosts would be 3.0 ± 1.0 ºF (-16.1 ± 0.55 ºC). For the medium temperature refrigeration tests, an EPR was used to set the suction pressure so that the tub temperatures would be approximately 38.0 ± 1.0 ºF (3.3 ± 0.55 ºC).
Figure 8 depicts the average temperature, the average of the coldest and warmest tubs and the single coldest and warmest points recorded for both low temperature and medium temperatures conditions.

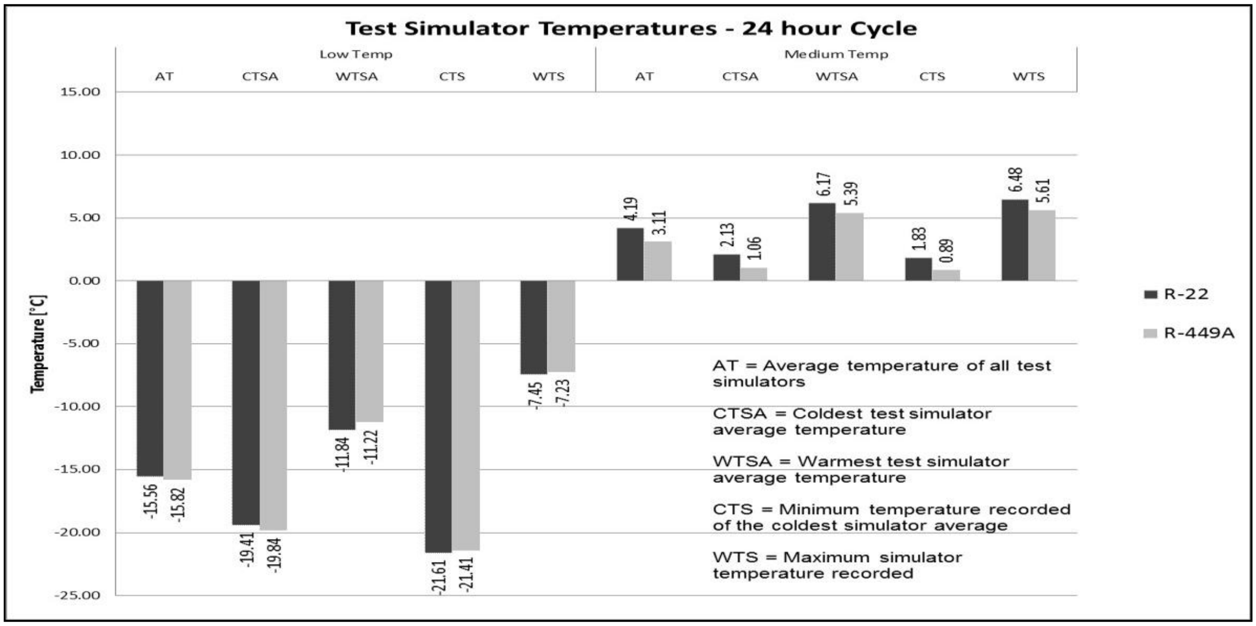

Figure 8. Test simulator temperatures

Since both tested refrigerants were exposed to the same elements, system and time, an energy comparison was made. Figure 9 shows the absolute energy usage of R-22 and R-449A.

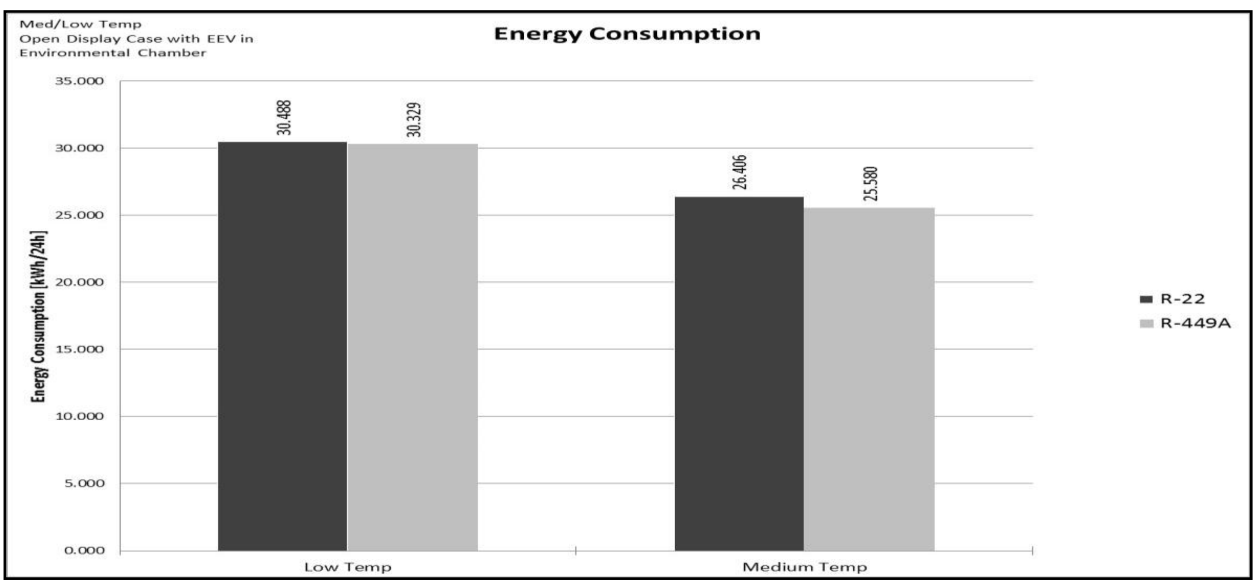

Figure 9. Absolute energy consumption of R-22 and R-449A under controlled ambient conditions.

5. DISCUSSION OF RESULTS

When converting an R-22 system to another refrigerant, there are certain measures that have to be taken regardless of what refrigerant is being used. For example, most R-22 refrigeration systems use mineral oil as a compressor lubricant. In order to run a system with an HFC or an HFO refrigerant, the lubricant has to be changed to POE. Also, R-22 and R-22 containing blends interact relatively strongly with many elastomers causing significant swelling in the seals. Any time an R-22 retrofit is performed the critical elastomeric seals should be replaced in order to prevent refrigerant leaks from the lasting effects of R-22 on the seals.

The pressure-temperature relationship of the retrofit gas is another important factor to consider when trying to determine what makes an ideal retrofit gas. If the suction pressure of the retrofit gas is too high or low relative to the existing refrigerant, then the powerhead on the TXV may not function properly. Low temperature thermodynamic modeling calculates the suction pressure of R-449A to be 1.5 psi (10.3 kpa [abs]) higher than R-22. For medium temperature applications this delta is expected to increase to 5.5 psi (37.9 kpa [abs]). Both of the differences in pressure are well within the adjustable range of properly sized TXVs. If the TXV is nonadjustable, then the superheat in the evaporator will increase. On the condensing side of the system, the discharge pressure of R-449A is 33.5 psia (230.9 kpa [abs]) larger than R-22. This elevated discharge pressure will ensure that there is enough of a pressure gradient to push the refrigerant through the TXV. Conversely, because R-449A has a higher discharge pressure but similar to only slightly higher suction pressures than R-22, the compression ratio of R-449A will be higher than that of R-22.

Thermodynamic cycle modeling has computed the capacity of R-449A to be equivalent to slightly larger than R-22, the COP (with no R-22 liquid injection) to be 4-6% lower and an 11-14% higher mass flow. Since thermodynamic cycle model results are limited to being purely theoretical, the performance claims were tested in a compressor calorimeter for comparison. For the low evaporator compressor calorimeter tests, the capacity and COP were slightly higher than the model predicted. This discrepancy may be due to an isentropic efficiency slightly lower than 70%. Lower isentropic efficiencies increase the discharge temperature and as the discharge temperature increases, more liquid injection is needed to maintain the maximum discharge temperature of the compressor at 275ºF (135ºC). Across all medium temperature calorimeter tests, R-449A had a 3% higher capacity and a 5% lower COP which is generally in agreement with the theoretical thermodynamic modeling.

System testing in controlled environmental chambers subjected R-22 and R-449A to the same ambient conditions. In both low temperature and medium evaporation temperature conditions, R-449A used slightly less energy than R-22. While the environmental chamber testing was being run, R-449A was also retrofitted into an R-22 medium temperature supermarket rack in the United States. Figure 10 plots the energy usage of the supermarket refrigeration system as a function of the ambient temperature for both R-449A and R-22.

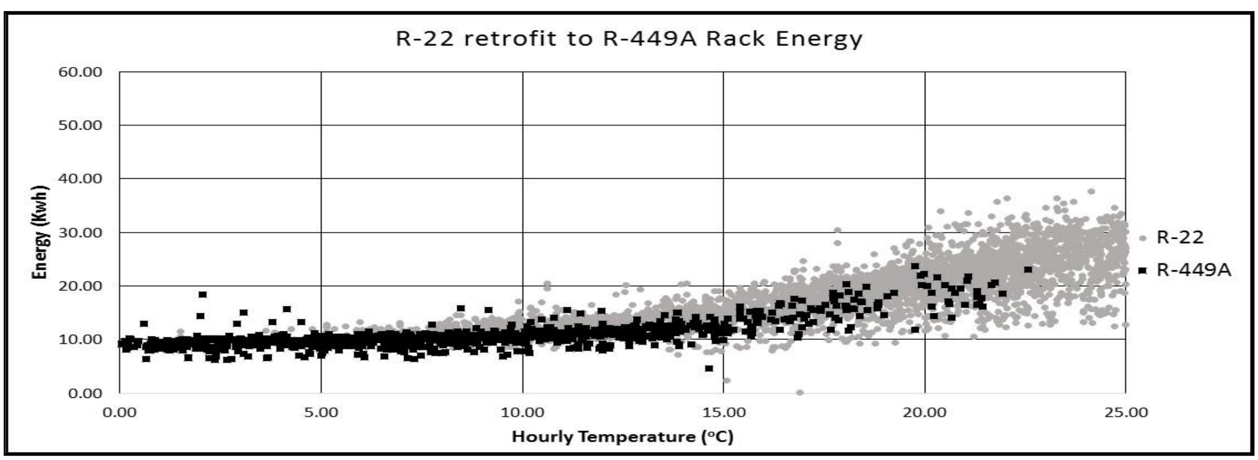

Figure 10. Medium Evaporating Temperature R-22 to R-449A Energy Data

There are many factors that impact supermarket energy data such as fluctuations in temperature in between recorded data points, varying cooling loads, and how busy the store is on a given day. Nevertheless, to date, the collected energy data has been following the same trend as the controlled environmental system testing. The retrofit occurred in the late fall of 2015, warming condensing temperature data for R-449A will be collected throughout the summer of 2016.

6. CONCLUSIONS

R-449A was initially developed as a low GWP replacement for R-404A/R-507. Thermodynamic modeling, compressor calorimeter and controlled environment system testing has concluded that R-449A can also be used as a replacement for R-22 in low and medium temperature refrigeration applications. The capacity and COP of R-449A are both expected to be near or exceed the capacity and COP of R-22. While the mass flow of R-449A will be 11-14% higher, the suction pressure delta between R-22 and R-449A is a close enough match that changes to the thermal expansion valve will not be necessary. Controlled environment system testing concluded that R-449A uses slightly less energy compared to R-22. This claim was verified in a real world system test by a supermarket that replaced their R-22 with R-449A in a medium temperature refrigeration rack.

NOMENCLATURE

HFO	Hydrofluoroolefin
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
HVACR	Heating Ventilation Air Conditioning and Refrigeration
COP	Coefficient of Performance
TXV	Thermostatic Expansion Valve
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers

REFERENCES

AHRI, *Standard 210/240-2008, Unitary A/C and Air Source Heat Pump Standard*, Arlington, VA, 2008.

ASHRAE, *Standard 23.1 Methods of Testing for Performance Rating Positive Displacement Refrigerant Compressors and Condensing Units That Operate at Subcritical Temperatures of the Refrigerant*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2010.

ASHRAE, *Standard 72 Method of Testing Open and Closed Commercial Refrigerators and Freezers*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2014.

IPCC. 2013. *Climate Change 2013: The physical Science Basis.* Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA.

J. Steven Brown, Piotr A. Domanski, Eric W. Lemmon. (2015). *Cycle D Version 5.0*. National Institute of Standards and Technology.

Powell, P. (2014). EPA Finalizes R-22 Phaseout Plan. *ACHR News.* http://www.achrnews.com/articles/127966-epa-finalizes-r-22-phaseout-plan

UNEP. 2000. *The Montreal Protocol on Substances that Deplete the Ozone Layer.* The Vienna Convention for the Protection of the Ozone Layer & The Montreal Protocol on Substances that Deplete the Ozone Layer. UNON, Nariobi, Kenya.

ACKNOWLEDGEMENT

Joshua Hughes, Barbara Minor and Sonali Shah all played an integral role in the generation of the data presented in this paper.