ORNL/TM-2015/351

Alternative Refrigerant Evaluation for High-Ambient-Temperature Environments: R-22 and R-410A Alternatives for Mini-Split Air Conditioners

Omar Abdelaziz
Jeffrey Munk
Som Shrestha
Randall Linkous
William Goetzler
Matthew Guernsey
Theo Kassuga

Working Paper:
Approved for Public Release;
Distribution is Unlimited.

August 2015

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

DOCUMENT AVAILABILITY

Reports produced after January 1, 1996, are generally available free via US Department of Energy (DOE) SciTech Connect.

Website http://www.osti.gov/scitech/

Reports produced before January 1, 1996, may be purchased by members of the public from the following source:

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
Telephone 703-605-6000 (1-800-553-6847)
TDD 703-487-4639
Fax 703-605-6900
E-mail info@ntis.gov
Website http://www.ntis.gov/help/ordermethods.aspx

Reports are available to DOE employees, DOE contractors, Energy Technology Data Exchange representatives, and International Nuclear Information System representatives from the following source:

Office of Scientific and Technical Information
PO Box 62
Oak Ridge, TN 37831
Telephone 865-576-8401
Fax 865-576-5728
E-mail reports@osti.gov
Website http://www.osti.gov/contact.html

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ORNL/TM-2015/351

Energy and Transportation Science Division

ALTERNATIVE REFRIGERANT EVALUATION FOR HIGH-AMBIENT-TEMPERATURE ENVIRONMENTS: R-22 AND R-410A ALTERNATIVES FOR MINI-SPLIT AIR CONDITIONERS

Omar Abdelaziz, Jeffrey Munk, Som Shrestha, Randall Linkous,
William Goetzler,* Matthew Guernsey*, and Theo Kassuga*

---

*Navigant Consulting, Inc., Burlington, Massachusetts 01803

Date Published: August 2015

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37831-6283
managed by
UT-BATTELLE, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ACRONYMS

AC	air conditioner
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
ASME	American Society of Mechanical Engineers
BEE	Bureau of Energy Efficiency
BIS	Bureau of Indian Standards
BTRIC	ORNL's Building Technologies Research and Integration Center
CAP	Compliance Assistance Programme at UNEP's Regional Office for West Asia
CFC	chlorofluorocarbon
CFD	computational fluid dynamics
CFM	cubic feet per minute
COP	coefficient of performance
DOE	US Department of Energy
EER	energy efficiency ratio
EGYPRA	Egyptian Program for Promoting Low-GWP Refrigerants' Alternatives
EPA	US Environmental Protection Agency
GWP	global warming potential
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
HFO	hydrofluoroolefin (unsaturated HFC)
HP	heat pump
HPDM	ORNL's Heat Pump Design Model
IIR	International Institute of Refrigeration
LCCP	life cycle climate performance
MBH	1,000 British thermal units per hour
MLF	Multilateral Fund for the Implementation of the Montreal Protocol
NSCL	National Superconducting Cyclotron Laboratory
ODS	ozone-depleting substance
ORNL	Oak Ridge National Laboratory
POE	polyolester (oil)
PRAHA	Promoting Low-GWP Alternative Refrigerants in the Air-Conditioning Industry for High Ambient Conditions
R&D	research and development
RAMA	Refrigeration & Air-Conditioning Manufacturers Association
RTOC	Refrigeration and Air Conditioning Technical Options Committee of UNEP (also referred to as the UNEP Technical Options Committee on Refrigeration, Air-Conditioning, and Heat Pumps)
TEAP	Technology and Economics Assessment Panel
TR	refrigeration tons
UNEP	United Nations Environmental Programme
UNIDO	United Nations Industrial Development Organization
VRF	variable refrigerant flow

EXECUTIVE SUMMARY

The Oak Ridge National Laboratory (ORNL) High-Ambient-Temperature Evaluation Program for Low-GWP Refrigerants aims to develop an understanding of the performance of low global warming potential (low-GWP) alternative refrigerants to hydrochlorofluorocarbon (HCFC) and hydrofluorocarbon (HFC) refrigerants in mini-split air conditioners under high-ambient-temperature conditions. This interim report describes the parties involved, the alternative refrigerants selection process, the test procedures, and the preliminary results.

ORNL designed a test matrix of 84 tests. Table 1 shows the refrigerants identified for testing in the two "soft-optimized" ductless mini-split air conditioners provided by Carrier for the testing.[1] Both units have a cooling capacity of 5.25 kW_{th}; one unit is designed to operate with R-22 [2.78 coefficient of performance (COP), equivalent to a 9.5 energy efficiency ratio (EER)], and the other with R-410A (3.37 COP, equivalent to an 11.5 EER). Table 2 shows the characteristics of the alternative refrigerants evaluated in the R-22 unit.

Table 1. ORNL test plan summary[a]

	Base	**N-20B**	**DR-3**	**ARM-20B**	**L-20A (R-444B)**	**R-290**	**ARM-71A**	**R-32**	**DR-55**	**L-41 (R-447A)**	**HPR-2A**	**Total # of Tests**
R-22 Unit	I	I	I	I	I	F						**42**
R-410A Unit	F						F	F	F	F	F	**42**

[a] I = Completed for the interim report; F=To be completed for the final report

---

[1] Soft-optimized units are production units that have undergone modifications such as refrigerant charge optimization, lubricant change, and flow control device changes to run with a different refrigerant. This is contrast with full-optimization, where the unit is purpose-built for a refrigerant. For details, see Section 1.2.

Table 2. Baseline and lower-GWP alternative refrigerant characteristics for the R-22 Unit

Refrigerant	Manufacturer	ASHRAE Safety Class	GWP [a]		System GWP (CO_2 equivalent, AR5) kg
			AR4	AR5	
R-22 (Baseline)	-	A1	1,810	1,760	2,495
N-20B[b]	Honeywell	A1	988	904	1,886
DR-3[b]	Chemours	A2L	148	146	293
ARM-20B[b]	Arkema	A2L	251	251	398
L-20A (R-444B)[b]	Honeywell	A2L	295	295	462
R-290	-	A3	3	3	*To be included in the final report*

[a] Sources: IPCC AR4, 2007[2]; IPCC AR5, 2013[3]

[b] GWP values for refrigerant blends not included in IPCC reports are calculated as a mass weighted average using manufacturer-supplied compositions.

For all refrigerants, efficiency degrades with increased ambient temperature. Further, since the test units were not designed specifically for the alternative refrigerants, but were soft-optimized, the alternative refrigerants are not expected to perform at identical levels as the baseline. Consistent with these expectations, at each of the tested ambient conditions, the preliminary results from the R-22 unit tests indicate that the alternative refrigerants delivered lower performance than R-22 (both in terms of COP and cooling capacity).

Table 3 summarizes the results at moderate ambient temperatures (AHRI Standard 210/240 A and B).

Table 3. ORNL interim test result at AHRI Standard 210/240 A, B (performance change from baseline in parentheses)[a]

	AHRI B Outdoor: 27.8°C (82°F)		**AHRI A** Outdoor: 35.0°C (95°F)	
	COP	**Capacity**	**COP**	**Capacity**
R-22 (Baseline)	3.48	6.26	3.07	6.10
N-20B	3.04 (-12.6%)	5.42 (-13.4%)	2.68 (-12.5%)	5.25 (-14.0%)
DR-3	2.88 (-17.1%)	5.52 (-11.8%)	2.57 (-16.1%)	5.4 (-11.5%)
ARM-20B	3.06 (-12.2%)	6.05 (-3.3%)	2.71 (-11.8%)	5.91 (-3.1%)
L-20A (R-444B)	3.02 (-13.3%)	5.53 (-11.6%)	2.72 (-11.3%)	5.58 (-8.6%)

[a] Shading legend – blank: < 5% degradation; yellow: 5–10% degradation; orange: >10% degradation

Table 4 summarizes the results at high ambient temperatures (Hot and Extreme).

---

[2] IPCC, 2007: *Climate Change 2007: The Physical Science Basis. Contribution of Working Group I to the Fourth Assessment Report of the Intergovernmental Panel on Climate Change* [Solomon, S., D. Qin, M. Manning, Z. Chen, M. Marquis, K.B. Averyt, M. Tignor and H.L. Miller (eds.)]. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA; section 2.10.2: Direct Global Warming Potentials. Available: https://www.ipcc.ch/publications_and_data/ar4/wg1/en/contents.html

[3] IPCC, 2013. Myhre, G., D. Shindell, F.-M. Bréon, W. Collins, J. Fuglestvedt, J. Huang, D. Koch, J.-F. Lamarque, D. Lee, B. Mendoza, T. Nakajima, A. Robock, G. Stephens, T. Takemura and H. Zhang, 2013: Anthropogenic and Natural Radiative Forcing. In: *Climate Change 2013: The Physical Science Basis. Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change* [Stocker, T.F., D. Qin, G.-K. Plattner, M. Tignor, S.K. Allen, J. Boschung, A. Nauels, Y. Xia, V. Bex and P.M. Midgley (eds.)]. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA. Available: https://www.ipcc.ch/pdf/assessment-report/ar5/wg1/WG1AR5_Chapter08_FINAL.pdf.

Table 4. ORNL interim test results at Hot and Extreme (performance change from baseline in parentheses)[a]

	Hot Ambient Outdoor: 52°C (125.6°F)		**Extreme Ambient** Outdoor: 55°C (131°F)	
	COP	**Capacity**	**COP**	**Capacity**
R-22 (Baseline)	1.98	5.00	1.82	4.76
N-20B	1.77 (-10.7%)	4.26 (-14.8%)	1.64 (-9.7%)	4.10 (-13.9%)
DR-3	1.70 (-14.3%)	4.41 (-11.7%)	1.55 (-14.8%)	4.21 (-11.7%)
ARM-20B	1.76 (-11.2%)	4.84 (-3.2%)	1.61 (-11.2%)	4.62 (-3.0%)
L-20A (R-444B)	1.85 (-7.0%)	4.79 (-4.1%)	1.69 (-6.9%)	4.59 (-3.7%)

[a]Shading legend – blank: < 5% degradation; yellow: 5–10% degradation; orange: >10% degradation

At the highest ambient-temperature test condition ("extreme" 55°C outdoor temperature), two alternative refrigerants demonstrated only modest performance degradation relative to R-22. The highest performing refrigerant, R-444B, had a COP 6.9% lower and a cooling capacity 3.7% lower than the baseline refrigerant.[4]

These efficiency and capacity results would be expected to improve through design modifications that manufacturers would normally conduct prior to introducing a new product to market. However, because the scope of this study only covered soft-optimization testing, no assessment can be made as to the extent of potential improvements through design changes. The ORNL test plan involved relatively few optimizations, including refrigerant charge optimization, lubricant change, and capillary tube/flow control changes. Thus, additional performance improvements would be expected through more thorough engineering design, including compressor and heat exchanger optimization.

Many countries have already undertaken the transition from R-22 to R-410A and have faced many similar challenges in maintaining capacity and performance using the new refrigerants. That transition involved design and other modifications to account for differences in properties between the two fluids, including a reduction in COP of R-410A of up to approximately 6% at typical rating conditions, depending on equipment type and level of optimization.[5] Through engineering design and optimization, some performance losses can be overcome.

This document is an interim working paper and includes only a limited set of experimental data for a subset of the alternative refrigerants to be evaluated under this program. Upon completion, a final report will be published that describes the experimental results for all the alternative refrigerants evaluated under this program, including R-290 in the R-22-designed unit, as well as all of the baseline and alternative refrigerants in the R-410A unit.

---

[4] Based on the uncertainty analysis, the air-side capacity has an uncertainty of ±3.6% and the air-side COP also has an uncertainty of 3.6%. As such, refrigerants whose performance values are within 5% of the baseline should be a good match to R-22. Furthermore, values within 10% of the baseline should indicate an acceptable match that requires incremental soft-optimization to reach parity with R-22 performance.

[5] COP data based on ductless split system A/C performance from ARI AREP Reports #212 and #213: *New Optimized System for Refrigerant Blend R-410A* (Daikin Industries, Ltd., February 1996). Although these systems were not fully optimized, they involved more substantial design changes than were undertaken in the ORNL evaluation. Additional AREP results show varying performance results, accounting for many variables; some tests found no change in efficiency.

1. INTRODUCTION

Hydrofluorocarbon (HFC) refrigerants are non-ozone-depleting fluids that are used as working fluids in air-conditioning and refrigeration equipment as substitutes for ozone-depleting substances (ODS) which have been or are being phased out under the Montreal Protocol. However, due to concerns regarding the direct global warming impact of HFCs, there has been increasing pressure to phase down HFC usage as well, especially for HFCs with high global warming potential (GWP). While HFCs currently account for only 1% of greenhouse gas emissions, their use is growing rapidly by as much as 10 to 15% per year, primarily due to their use as replacements for ODS and the increasing use of air conditioners globally. Consequently, some countries have proposed using the Montreal Protocol to govern a phase down of the production and consumption of HFCs. A phase-down could prevent global warming by up to 0.1°C by 2050 and up to 0.5°C by 2100.[6]

One key technical barrier hindering progress toward an HFC phase-down agreement is concern over the performance of the most commonly proposed low-GWP refrigerants. Of particular concern is the performance degradation that lower-GWP refrigerants might experience at high-ambient-temperature conditions. In order to address this issue, the US Department of Energy (DOE), in cooperation with Oak Ridge National Laboratory (ORNL), has established an evaluation program to assess the performance of low-GWP alternative refrigerants under high-ambient-temperature conditions. Carrier Corporation, international refrigerant suppliers, and technical experts from various countries are also participating. The program evaluates the performance of air conditioners under high-ambient-temperature conditions using low-GWP refrigerants. The objective is to assess whether it is possible to achieve similar or better energy efficiency and cooling capacity with low-GWP refrigerants compared to current baseline refrigerants such as R-22 and R-410A. This program is guided by an expert panel consisting of members of government, academia, and industry from interested countries.

Other evaluation programs aimed at understanding the performance of low-GWP refrigerants at high ambient temperatures are currently under way. The United Nations Environment Programme (UNEP) and the United Nations Industrial Development Organization (UNIDO) are running two separate programs funded by the Multilateral Fund for the Implementation of the Montreal Protocol (MLF): Promoting Low-GWP Alternative Refrigerants in the Air-Conditioning Industry for High-Ambient Conditions (PRAHA) and Egyptian Program for Promoting Low-GWP Refrigerants' Alternatives (EGYPRA).[7,8] PRAHA was developed in association with several high-ambient-temperature nations and is evaluating purpose-built prototypes, while EGYPRA is based in Egypt and is also testing purpose-built prototypes. Sections 3.2 and 3.3 of this report discuss PRAHA and EGYPRA, respectively, in more depth. In addition to those efforts, participants in the Air-Conditioning, Heating, and Refrigeration Institute's (AHRI) Low-GWP

---

[6] Xu Y., Zaelke D., Velders G. J. M., & Ramanathan V. (2013) *The role of HFCs in mitigating 21st century climate change*, ATMOS. CHEM. PHYS. 13:6083-6089; see also Hare B. et al. (2012) *CLOSING THE 2020 EMISSIONS GAP: ISSUES, OPTIONS AND STRATEGIES*; and Ramanathan V. & Xu Y. (2010) *The Copenhagen Accord for limiting global warming: Criteria, constraints, and available avenues*, PROC. NAT'L ACAD. SCI. U.S.A. 107:8055-8062

[7] UNEP & UNIDO Presentation on EGYPRA, June 2015.

[8] El-Talouny, A. & Nielsen, O. (2014) *Promoting Low-GWP Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA)*, UNEP and UNIDO, OzonAction Fact Sheet - November 2014, available at http://www.unep.org/ozonaction/Portals/105/documents/events/MOP26/Fact%20Sheet%20Promoting%20low-GWP%20Refrigerants%20for%20Air-Conditioning%20Sectors%20in%20High-Ambient%20Temperature%20Countries%20(PRAHA).pdf

Alternative Refrigerants Evaluation Program (Low-GWP AREP) testing program are conducting high-ambient-temperature testing.[9]

1.1 PURPOSE/OBJECTIVES

The objective of the program is to evaluate the performance and help determine the viability of using low-GWP refrigerants for mini-split air-conditioning units under high ambient temperatures.

1.2 SCOPE AND COVERAGE

The program evaluates the performance of mini-split (ductless) air-conditioning units originally designed to use HCFC-22 and HFC-410A, both when using the baseline refrigerants and when using low-GWP alternatives. The primary objective of the evaluation is to determine whether it is possible, using the lower-GWP alternatives, to achieve comparable or better performance than with R-22 and R-410A. Ductless mini-split air conditioners were chosen as the equipment to be evaluated because they are the most common type of air conditioners used in residential and light commercial applications in most high-ambient-temperature regions.

The evaluation is being performed at ORNL in Oak Ridge, Tennessee, USA, using a range of fluorinated and non-fluorinated low-GWP refrigerants, which are tested and compared to two baselines – R-22 (an HCFC with GWP=1,760) and R-410A (an HFC with GWP=1,924).[10] There is currently a global effort to transition from R-22, as agreed under the Montreal Protocol. Many are also transitioning from R-410A due to its high GWP. These transitions are at various stages in different parts of the world, so including both refrigerants as baselines can provide a point of reference regardless of where particular countries stand in the transition process.

Testing of the baseline refrigerants was first carried out on the original equipment provided by the manufacturer. The units were then soft-optimized for use with the alternative refrigerants. Soft-optimized equipment can be modified with standard production line components, which differentiates it from drop-in tests (where only minor adjustments are allowed) and from purpose-built prototype testing (where units are custom-designed to work with a specific alternative refrigerant). Drop-in tests are the simplest to conduct, while purpose-built prototypes are the most complex. Soft-optimization is considered as an intermediate step. Purpose-built prototypes have the potential of achieving higher efficiency levels, but the process of designing and manufacturing is more complex and time-consuming compared to soft-optimization and drop-in tests.

1.3 PARTICIPANTS

1.3.1 Oak Ridge National Laboratory (ORNL)

ORNL has been involved in the research and development (R&D) of space-conditioning equipment and appliances for nearly 40 years.[11] Building Technologies Research and Integration Center (BTRIC) partnerships with industry have resulted in successful introduction of products such as high-efficiency refrigerator-freezers, heat pump water heaters, high-efficiency supermarket refrigeration systems, and

---

[9] For details, refer to http://www.ahrinet.org/site/514/Resources/Research/AHRI-Low-GWP-Alternative-Refrigerants-Evaluation

[10] IPCC AR5 GWP values. See Section 4.2 for discussion of refrigerants and GWP values (and sources).

[11] ORNL's website includes detailed information on their history of work in space conditioning and appliances; available at: http://web.ornl.gov/sci/buildings/

hybrid desiccant/vapor compression air-conditioning systems.[12] Nine of these products have won the prestigious R&D 100 Award.

The BTRIC User Facility at ORNL is the premier US DOE research facility devoted to the development of technologies that improve the energy efficiency and environmental compatibility of residential and commercial building equipment. BTRIC's mission is to identify, develop, and deploy energy-efficient technologies by forming partnerships between DOE and industry for technology development and analysis, well-characterized laboratory and field experiments, and market outreach. The experimental facilities for building equipment research are ISO14001 certified for environmental compliance.

BTRIC is a leading center for the development of innovative air conditioners, heat pumps, water heaters, and appliances. The ORNL Heat Pump Design Model (HPDM) is one of the most frequently used heat pump models and is currently being used by several original equipment manufacturers (OEMs) in their sizing and selection software tools.[13,14] Furthermore, ORNL led the development of integrated heat pumps (air source and ground source) as well as heat-pump water heaters.[15]

BTRIC also has decades of experience in the research, design, and development of advanced heat exchangers. Its expertise in this area includes the measurement of heat transfer coefficients for zeotropic refrigerant mixtures and methods for improvement; evaluation of microchannel heat exchangers; and computational fluid dynamics (CFD) modeling to improve the performance of heat exchangers in heating, ventilation, and air-conditioning equipment by reducing maldistribution of air across the heat exchanger and of refrigerant inside the heat exchanger. In addition, ORNL has recently been involved in the application of rotary heat exchangers for refrigeration applications.[16]

Finally, BTRIC has decades of experience in alternative refrigerant evaluation programs. User facilities and flagship modeling capabilities were used during the CFC to HCFC transition, the HCFC to HFC transition, and are currently being leveraged as part of the transition from high-GWP HFCs to lower GWP refrigerants. This work has produced numerous publications in this field. Select examples include the following.

- *CFC Phase-out* – a strategy development project concerned with containing existing refrigerant and retrofitting or replacing CFC-based chillers with alternative refrigerants[17]
- *Global Warming Impacts of Ozone-Safe Refrigerants and Refrigeration, Heating, and Air-Conditioning Technologies* – an analysis of the contributions of various refrigerants in major applications to global warming[18]
- *Development of Low Global Warming Potential Refrigerant Solutions for Commercial Refrigeration Systems Using a Life Cycle Climate Performance (LCCP) Design Tool* – an LCCP

---

[12] For more information on BTRIC, see the website at: http://www.ornl.gov/user-facilities/btric

[13] For more information on ORNL’s HPDM, see their website at: http://web.ornl.gov/~wlj/hpdm/MarkVI.shtml

[14] For a list of relevant reports on HPDM, see http://web.ornl.gov/~wlj/hpdm/Related_Reports.html

[15] Example program report: Rice et al. (1993) *Thermodynamic Cycle Evaluation Model for R-22 Alternatives in Heat Pumps – Initial Results and Comparisons*, available: http://web.ornl.gov/~webworks/cppr/y2001/pres/67370.pdf

[16] For a list of capabilities, see ORNL’s Experimental Capabilities and Apparatus Directory at: http://web.ornl.gov/sci/buildings/docs/buildings_catalog.pdf

[17] CFC project phase-out summary available: http://homer.ornl.gov/sesa/environment/ods/ornl.pdf

[18] Fischer et al. (1997) *Global Warming Impacts of Ozone-Safe Refrigerants and Refrigeration, Heating, and Air-Conditioning Technologies*, available: http://www.osti.gov/scitech/biblio/555370

analysis of the performance of typical commercial refrigeration systems with alternative refrigerants and minor system modifications[19]

- *Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies* – a comparative analysis of the global warming impacts of alternative technologies using total equivalent warming impact (TEWI)[20]

1.3.2 Industry

As part of this program, major refrigerant producers such as Arkema, Chemours (formerly DuPont), Honeywell, and Mexichem provided sample prototype refrigerants with a lower GWP compared to existing refrigerants. They supplied ORNL with refrigerants that are being considered as alternatives to R-22 and R-410A at high ambient conditions. Carrier, a US-based air conditioner manufacturer, donated equipment for testing, including both mini-split air-conditioning units specially designed for the high-ambient-temperature conditions. One unit is designed for R-22 and the other for R-410A.

1.3.3 Expert Panel

A group of HVAC experts was assembled to provide input and guidance to the evaluation program, including design of the program and review of the test results, the interim working paper, and the final report. The investigators conducting the testing recognized that, given the international implications of the results of the evaluation program, it was essential that this panel consist of individuals from various nations, especially countries with hot climates. Accordingly, a number of governments were contacted to recommend technical personnel who, whether from government, academia, or industry, would act independently, on their own behalf (i.e., not formally representing a government or an industrial entity) in providing guidance for this effort. In addition, representatives from UNEP and UNIDO were also asked to join the panel, given the significant involvement of both these UN organizations in projects aimed at developing solutions for the replacements of HCFC and high-GWP HFC refrigerants in the air-conditioning sector. The panel met for the first time via teleconference on March 23, 2015, for a second time in a face-to-face meeting at the United Nations Conference Centre in Bangkok, Thailand, on April 19, 2015, and for a third time via teleconference on June 30, 2015. Additional teleconferences and one more face-to-face meeting are planned.

1.3.3.1 Mandate

The panel is tasked with providing technical input for this study: *Alternative Refrigerant Evaluation for High-Ambient-Temperature Environments: R-22 and R-410A Alternatives for Mini-Split Air Conditioners*. The technical input requested from the panel includes recommending alternative refrigerants to be tested, commenting on appropriate test procedures, assessing results, and reviewing the interim working paper and the final report.

---

[19] Abdelaziz et al. (2012) *Development of Low Global Warming Potential Refrigerant Solutions for Commercial Refrigeration Systems using a Life Cycle Climate Performance Design Tool*, available: http://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=2352&context=iracc

[20] Sand et al. (1997) report to US DOE, *Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies*, available: http://web.ornl.gov/sci/ees/etsd/btric/eere_research_reports/electrically_driven_heat_pumps/fluids_development/cfc_and_hcfc_replacements/tewi_3/tewi_3.pdf

1.3.3.2 Member Composition

Brief biographies of the panel members are included below.

Dr. Radhey Agarwal (India)

Radhey Agarwal is a Mechanical Engineer and received his Ph.D. from the Indian Institute of Technology Delhi (India) in 1975. He specializes in refrigeration, air-conditioning, and alternative refrigerants to CFCs and HCFCs. He is a former Deputy Director (Faculty), Dean of Industrial Research & Development and Chairman, Department of Mechanical Engineering, IIT Delhi. He was the Co-Chair, UNEP Technical Options Committee on Refrigeration, Air-conditioning and Heat Pumps (RTOC) and a member of the Technology and Economics Assessment Panel (1996–2008) of the Montreal Protocol. He has been actively contributing towards efforts to protect the ozone layer as part of the Technology and Economics Assessment Panel (UNEP TEAP) since 1989. He is the recipient of the 1998 US Environmental Protection Agency (EPA) Stratospheric Ozone Protection Award for Technical Leadership in CFC-Free Refrigeration and the 2007 US EPA Stratospheric Ozone Protection Award Best of the BEST.
Dr. Agarwal was the Vice-President of IIR, Commission-B2 and member of the scientific committee of the IIR. He is a member of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) and the Indian Society of Heating, Refrigerating, and Air-Conditioning Engineers (ISHRAE).

Dr. Karim Amrane (USA)

Karim Amrane is Senior Vice President of Regulatory and International Policy at the Air-Conditioning, Heating, and Refrigeration Institute (AHRI). He manages the industry's cooperative research program and is responsible for the development and implementation of AHRI's regulatory and international policy. He holds a Ph.D. in Mechanical Engineering from the University of Maryland at College Park (Maryland, USA) where he currently is a part-time faculty member. Dr. Amrane has over 25 years of experience in the air-conditioning and refrigeration industry. He is a member of (ASHRAE), the International Institute of Refrigeration (IIR), and the American Society of Mechanical Engineers (ASME).

Dr. Enio Bandarra (Brazil)

Enio P. Bandarra Filho is an associate professor of mechanical engineering from the Federal University of Uberlandia (Brazil). He received his BS degree from the State University of Sao Paulo (Brazil) in 1994, and his MS and Ph.D. degrees from the University of Sao Paulo, in thermal sciences, in 1997 and 2002, respectively. In 2007–2008 he was a visiting professor in the heat and mass transfer laboratory at the Ecole Polytechnique Federale de Lausanne, Switzerland, working with oil-refrigerant mixtures in two-phase flow. Currently, he has 1 postdoc, 8 Ph.D., and 4 M.S. students in different areas, such as refrigeration and air-conditioning, heat transfer of nanofluids, heat exchangers, single-phase flow, control in refrigeration systems, and related topics. He published more than 260 papers in journals, book chapters, and conferences, including some awards received by the best-presented papers.

Dr. J. Bhambure (India)

Jitendra Bhambure, who is presently the Executive Vice President - R&D and Technology at Blue Star, received a degree in Electrical Engineering in 1979 from Bombay University (India) and a Post-Graduate degree in Management Studies from Mumbai University (India) in 1983. He joined Rallis India Ltd. in 1979 as a trainee engineer and worked there for 13 years. He was head of R&D before he left Rallis India. He joined Blue Star in 1992, and worked in various operations, before taking charge of R&D in 2000. He has trained in the United States, London Business School, IIM-A'bad, and at Tel Aviv University.

Dr. Bhambure was the founder and President of ISHRAE Thane Sub Chapter, which over the course of 3 years has become an independent chapter. He is a member of the ISHRAE Technical Committee and an active member of the Refrigeration & Air-Conditioning Manufacturers Association (RAMA) to represent Industry on Energy Efficiency and new refrigerants with Bureau of Energy Efficiency (BEE), Bureau of Indian Standards (BIS) and Ozone Cell under the Ministry of Environment & Forest. He is also the Chairperson of the ozone-depleting substances committee of RAMA.

Dr. Suely Machado Carvalho (co-chair; Brazil)

Suely Carvalho is a physicist and received her Ph.D. from Purdue University (USA). She was a postdoctoral researcher at the National Superconducting Cyclotron Laboratory (NSCL), Department of Energy (DOE), at Michigan State University, East Lansing, USA (1980). As former director of the Montreal Protocol Unit and Principal Technical Adviser for Chemicals for the United Nations Development Programme in New York (2002–2013), she led the implementation of projects in over 100 developing countries to replace ozone-depleting substances in several sectors. She was the UNEP TEAP co-chair for 10 years. As the former director of Technology Transfer at the São Paulo State Environment Protection Agency, CETESB (1985–1987), she established the Climate and Ozone Protection programs at the state level. She has been involved with the Montreal Protocol nationally and internationally for 25 years. Dr. Carvalho is currently adviser to the Superintendent at the Instituto de Pesquisas Energéticas e Nucleares, IPEN-CNEN, Ministry of Science, Technology and Innovation, MCTI, São Paulo, Brazil.

Mr. Ayman El-Talouny (UNEP)

Mr. El-Talouny joined the Compliance Assistance Programme (CAP) at UNEP's Regional Office for West Asia, financed by the MLF of the Montreal Protocol, starting in January 2003. He has been working in the field of refrigeration and air-conditioning since 1992. His experience in this field and its different sectors and applications qualified him to join the Egyptian Ozone Unit as a Technical Advisor in 1999, which is closely related to the Montreal Protocol. During the last 3 years he was involved directly in the preparation, implementation, and monitoring of the Egyptian Refrigerant Management Program as well as refrigeration compliance projects.

Dr. Tingxun Li (China)

Tingxun Li received his Ph.D. from SHANGHAI JIAOTONG University (China). He has been engaged in alternative refrigerant activities since 1995. As an associate professor in Sun Yat-sen University, he teaches courses in refrigeration and conducts research on air-conditioning and cryogenics. He led the conversion from R-22-based to propane-based room air conditioner manufacturing at Guangdong Midea Group in a demonstration project that was funded by MLF of the Montreal Protocol. As a member of the Refrigeration and Air-Conditioning Technical Options Committee (RTOC) of UNEP, he is one of the authors of the 2014 RTOC report and the report of the task force. He is also a member of IEC SC61D and has been engaged in a revision of the standard IEC 60335-2-40 since 2013.

Dr. Samuel Yana Motta (Peru)

Samuel F. Yana Motta received his BS degree from the National Engineering University in his native Peru, and his Ph.D. from the Catholic University (Brazil), all in mechanical engineering. Following a guest researcher appointment at the National Institute of Standards (Thermal Machinery Group – Gaithersburg, MD, USA), he joined Honeywell as a scientist in the Buffalo Research Laboratory in 2000. At Honeywell, he participated in the development of new environment-friendly refrigerants such as HFO-1234yf and HFO-1234ze. He has also assumed positions of increasing responsibility and leadership in the development of such fluids. He now leads the Global R&D teams responsible for developing new heat transfer fluids.

Mr. Ole Nielsen (UNIDO)

Mr. Nielsen graduated from the Technical University of Copenhagen (Denmark) in 1988 as a mechanical engineer specializing in energetics and refrigeration. He worked in the Danish refrigeration industry until 1996. Afterwards, Mr. Nielsen worked as an independent technical consultant on the Montreal Protocol project formulation and implementation until 2003. He then returned to the private sector as sales manager for refrigeration equipment. He joined the UNIDO Montreal Protocol team in 2011. Currently he is acting as Chief of the Montreal Protocol Unit. Mr. Nielsen has been involved with the Montreal Protocol since 1993 through consultancy, the private sector, and most recently through an implementing agency, with a specialty in the refrigeration and air-conditioning sectors.

Mr. Tetsuji Okada (Japan)

Tetsuji Okada received his BS degree from the University of Tokyo (Japan) and his MS degree from the University of California, Berkeley (USA), all in mechanical engineering. He joined Mitsubishi Electric Corporation in 1980. He was engaged in the design of domestic air conditioners and the development of finned tubed–type heat exchangers until 1995. From 1995 to 1998 he researched radiation air conditioning in the company's laboratory. He was the department manager of heat pump hot water heater development using CO_2 refrigerant from 2000 to 2009. He was transferred to the commercial air conditioners factory in Scotland as the vice president (2010–2012). Mr. Okada was the general manager of the Brussels office of Mitsubishi Electric Europe from 2012–2014 and is now the president of the Japan Refrigeration and Air Conditioner Industry Association.

Dr. Alaa Olama (Egypt)

Alaa Olama received his M.Sc. and Ph.D. from King's College, London University (England), in mechanical engineering, specializing in refrigeration and air-conditioning. He is the founder, board of directors' member, and past vice chair of the first district cooling company in Egypt, GasCool. He is a member of the RTOC of UNEP. Dr. Olama is the head of the committee writing the first District Cooling code for Egypt and a member of the committee writing the Egyptian code of Air Conditions, Refrigeration & Automatic Control and the Arab Refrigeration and Air Conditioning Code. He is the past president of the Board of Directors of ASHRAE Cairo Chapter 2002–2003 and general Chair, ASHRAE, of the Second Regional Conference of Refrigeration (ARC) Region-At-Large in Cairo, September 2003. He is a member of the international reviewers' panel of the low-GWP refrigerants testing program of PRAHA and the technical advisor of EGYPRA. Dr. Olama is an independent consultant.

Dr. Alessandro Giuliano Peru (Italy)

Dr. Peru was a researcher at the University Consortium CUEIM where he co-authored several research papers and technical reports for the protection of the environment. He has worked for more than 15 years in the ozone protection field. He was in charge of national plans for the phase out of ozone-depleting and high-GWP substances from 2000 to 2010. Starting in 2006, he was in charge, as financial expert, of the mobilization of financial resources and budget of the multi-environmental agreements and a member of the executive committee of the MLF. Dr. Peru has chaired and coordinated many technical and working groups at both the European Union and international level.

In addition, he was in charge of several bilateral and multilateral cooperation programs. In 2011, he received a Ph.D. in Economics and Finance in the Government Enterprise with a thesis on the Carbon

Footprint as a means of emerging Corporate Social Responsibility. In 2014, he was President of the European Union for the Montreal Protocol during the Italian Presidency of the Council of the European Union. In collaboration with the European Commission, he has coordinated the European Union position during the 2015 negotiation of the Montreal Protocol. Dr. Peru is the author of several articles and publications on environmental issues and Professor of Economics and Management at the Faculty of Economics at the University "La Sapienza." He is a member of the Technical Expert Panel of Stewardship and member of the International Expert Panel on High Ambient Temperature. Today he is Coordinator of the European Affairs for the Italian Ministry of the Environment, Land and Sea.

Dr. Patrick Phelan (co-chair; USA)

Patrick Phelan received his BS degree from Tulane University (New Orleans, Louisiana, USA), his MS degree from the Massachusetts Institute of Technology (USA), and his PhD from the University of California, Berkeley (USA), all in mechanical engineering. Following a 2 year postdoctoral fellowship at the Tokyo Institute of Technology (Japan), he started his academic career as an Assistant Professor at the University of Hawaii in 1992. In 1996 he moved to Arizona State University (USA), where he is a Professor of Mechanical & Aerospace Engineering and a Senior Sustainability Scientist. While on leave from Arizona State University, he served as the Director of the National Science Foundation Thermal Transport Processes Program from 2006 to 2008. Dr. Phelan is currently on leave from Arizona State University and is now the Program Manager for Emerging Technologies in the Building Technologies Office, Energy Efficiency and Renewable Energy, US DOE.

2. SELECTION OF ALTERNATIVES AND TESTING CONDITIONS

2.1 ALTERNATIVE REFRIGERANTS SELECTION

Guided by input from the expert panel, the investigators decided that the following criteria (in no particular order) should be considered when selecting alternative refrigerants for testing.

- Refrigerants shall have a lower GWP than the refrigerants being replaced. No strict upper limit on the GWP of alternative refrigerants was specified.
- Refrigerants shall be relatively close to commercial availability, or already commercially available.
- Refrigerants shall have properties that are a relatively close match to the baseline refrigerant that they are replacing. It is notable that temperature glide is an especially important property, in addition to capacity and coefficient of performance (COP).
- Refrigerants shall have readily available information about their characteristics.

In addition, it was decided that this program should include alternative refrigerants currently being evaluated by other high-ambient-temperature testing programs (e.g., Low-GWP AREP, PRAHA, and EGYPRA) and that flammability should not be one of the selection criteria. Nevertheless, the panel felt it important to include at least one alternative with A1 toxicity and flammability classification, provided other conditions were met (see Section 4.2 for a discussion of refrigerant safety classification).

Given the controversies inherent in Life Cycle Climate Performance (LCCP) models, the panel recommended that LCCP not be used as a selection criterion. While the concept of LCCP is generally accepted as a metric for evaluating alternative refrigerants, there is considerable disagreement about accurate LCCP values, largely due to uncertainties about refrigerant leakage rates.

2.2 TESTING CONDITIONS

Testing of all refrigerants was performed at each of the environmental conditions described in Table 5.

Table 5. Test conditions

Test Condition	**Outdoor** [a]	**Indoor**			
	Dry-Bulb Temp.	Dry-Bulb Temp.	Wet-Bulb Temp.	Dew Point Temp. [b]	Relative Humidity
	°C (°F)	°C (°F)	°C (°F)	°C (°F)	%
AHRI B [c]	27.8 (82)	26.7 (80.0)	19.4 (67)	15.8 (60.4)	50.9
AHRI A [c]	35.0 (95)	26.7 (80.0)	19.4 (67)	15.8 (60.4)	50.9
T3* [d]	46 (114.8)	26.7 (80.0)	19 (66.2)	15.8 (60.4)	50.9
T3	46 (114.8)	29 (84.2)	19 (66.2)	13.7 (56.6)	39
Hot	52 (125.6)	29 (84.2)	19 (66.2)	13.7 (56.6)	39
Extreme	55 (131)	29 (84.2)	19 (66.2)	13.7 (56.6)	39

[a] There is no specification for the outdoor relative humidity as it has no impact on the performance.
[b] Dew-point temperature at 0.973 atm (14.3 psi)
[c] Per AHRI Standard 210/240
[d] T3* is a modified T3 condition where the indoor settings are similar to the AHRI conditions.

3. OTHER HIGH-AMBIENT-TEMPERATURE EVALUATION EFFORTS

This section describes three other high-ambient-temperature evaluation programs for alternative refrigerants.

- PRAHA is a joint effort by UNEP and UNIDO and funded by the MLF, whose objective is to investigate alternative refrigerants for high ambient temperatures in air-conditioning applications.[21]
- EGYPRA is also a joint effort by UNEP and UNIDO but with a focus on Egypt.[22]
- Low-GWP AREP is an industry-wide effort on alternative refrigerants run by AHRI. While low-GWP AREP's focus is not on high-ambient-temperature testing, the program is conducting testing at high-ambient-temperature conditions.[23]

Table 6 compares the ORNL Evaluation Program with these three other high-ambient-temperature testing programs. Low-GWP AREP includes testing of a wide range of equipment; Table 6 covers only those low-GWP AREP tests that are directly pertinent to this report and that are discussed in more detail in Appendix A.

---

[21] El-Talouny, A. & Nielsen, O. (2014) *Promoting Low-GWP Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA)*, UNEP and UNIDO, OzonAction Fact Sheet - November 2014.
[22] UNEP & UNIDO Presentation on EGYPRA, June 2015.
[23] For details, refer to http://www.ahrinet.org/site/514/Resources/Research/AHRI-Low-GWP-Alternative-Refrigerants-Evaluation

Table 6. Comparison of high-ambient-temperature testing programs

	ORNL Evaluation Program		**EGYPRA (UNEP, UNIDO, Egypt)**				
Type of Test	Soft-optimized tests, comparing with base units: R-22 and R-410A		Individual test prototypes, comparing with base units: R-22 and R-410A				
Number of Prototypes	2 commercially available units, soft-optimized to compare with base refrigerants: R-22, R-410A		36 prototypes, each specific to one capacity and one refrigerant, compared with base units: R-22, R-410A				
Categories	60 Hz		50Hz				
	Split unit 18 MBH (designed for R-22)	Split unit 18 MBH (designed for R-410A)	Split 12 MBH	Split 18 MBH	Split 24 MBH	Central 120 MBH	Central micro Channel 120 MBH
Testing Conditions	ANSI/AHRI Standard 210/240 and ISO T3 condition, 52°C, and 55°C		EOS 4814 and 3795 (ISO 5151), T1 conditions plus one point in T3 conditions				
Prototypes Supplied & Tests Performed	ORNL (Oak Ridge National Laboratory), one supplier – soft-optimization in situ		Prototypes built at eight OEMs, test at NREA (Local test laboratory in Egypt)				
Refrigerants Tested	R-32, R-290, HFC/HFO blends (4 Types) vs. to R-22 HFC/HFO blends (4 Types) vs. to R-410A		R-32, R-290, HFC/HFO blends (3 Types) vs. to R-22 HFC/HFO blends (3 Types) vs. to R-410A				
Expected Delivery Dates	Preliminary report, July 2015 Final Report October 2015		Early 2016				
Constraints	To change some components of the two prototypes to accommodate the different refrigerants characteristics, within a soft-optimization process		To build new prototypes with dedicated compressors for the selected refrigerants with the condition to meet the same design capacities of the selected models in comparison to the R-22 or R-410A designs				
Other Components	N/A		N/A				

Table 6. (continued)

	PRAHA (UNEP,UNIDO, high-ambient temperature countries)				Low-GWP AREP (AHRI)[a]			
Type of Test	Individual test prototypes, comparing with base units: R-22 and R-410A				Soft-optimization and drop-in tests. Baseline units vary by application; for the air-conditioning results presented in this report, the baseline was R-410A			
Number of Prototypes	22 prototypes, each specific to one capacity and one refrigerant, compared with base units: R-22, R-410A				5 mentioned in this report, but tests at high ambient temperatures with other prototypes were performed within AREP			
Categories	60 Hz		50 Hz		60 Hz			
	Window 18 MBH	Decorative Split 24 MBH	Ducted 36 MBH	Packaged 90 MBH	Split central AC 36 MBH	Split central HP 36 MBH	Split central HP 42 MBH	VRF HP 96 MBH
Testing Conditions	ISO 5151 at T1, T3 and T3+ (50°C) and a continuity test for 2 hr at 52°C				ANSI/AHRI Standard 210/240, additional tests at 46.1°C & 51.7°C	ASHRAE Standard 116	ANSI/AHRI Standard 210/240, additional test at 46.1°C	AHRI 1230 & ASHRAE 37
Prototypes Supplied & Tests Performed	Prototypes built at 7 OEMs, test at Intertek				Units were manufactured or obtained by each party and tested at each party's facilities			
Refrigerants Tested	R-32, R-290 HFC/HFO blends (2 Types) vs. to R-22 HFC/HFO blends (2 Types) vs. to R-410A				The test results discussed in this report include R-32, D2Y60, L41A and R-1234yf			
Expected Delivery Dates	4th quarter of 2015				Phase I results have already been published; Phase II results are currently being published on a rolling basis			
Constraints	To build new prototypes with dedicated compressors for the selected refrigerants with the condition to meet the same design capacities of the selected models in comparison to the R-22 or R-410A designs				To conduct drop-in system tests and soft-optimized tests with any modifications clearly indicated in the test reports			
Other Components	The project includes other non-testing elements to assess relevant issues of EE standards, technology transfer and economics in addition to special reporting on the potential of District Cooling to reduce the use of high-GWP alternatives.				Compressor calorimeter tests and heat transfer tests are also performed			

[a] Only includes results available in this interim working paper (see Appendix A); does not include all Low-GWP AREP high-temperature results.

3.1 LOW-GWP ALTERNATIVE REFRIGERANTS EVALUATION PROGRAM (AREP)

The objective of Low-GWP AREP is to identify suitable alternatives to high-GWP refrigerants. The intent of the program is to help industry select promising alternative refrigerants, understand the technical challenges involved in applying those refrigerants, and identify what areas require further research which would lead to the use of these refrigerants. Ultimately, the objective is to identify potential replacements for the high-GWP refrigerants currently in use in the industry and present the performance of those replacements in a consistent manner. Low-GWP AREP is strongly supported by industry to assess the research needs, accelerate industry's response to environmental challenges raised by the use of high-GWP refrigerants, and avoid duplicative pre-competitive work by individual equipment manufacturers. In order to achieve these goals, the program has tested a broad spectrum of equipment, including air conditioners, heat pumps, dehumidifiers, chillers, water heaters, ice makers, and refrigeration equipment and has published 42 test reports and a literature review to date.

Low-GWP AREP has carried out compressor calorimeter tests, equipment drop-in tests, and soft-optimized equipment tests. Drop-in tests only allow for minor adjustments to the equipment being tested so that it may be used with the alternative refrigerant instead of the baseline refrigerant for which it was designed. Soft-optimized equipment can be modified using standard production line components, provided that the changes are indicated in the report; furthermore, the overall heat exchanger area must remain constant, but the area ratio between the condenser and the evaporator may be optimized. Compressor calorimeter tests were performed according to the conditions set forth in ASHRAE 23-2010.

Phase I of Low-GWP AREP ended in 2013 and led to 40 reports pertaining to a total of 38 refrigerants. A few conclusions can be drawn from the results of Phase I.[24]

- Several alternative refrigerants had performed similarly to the baseline refrigerants they replaced.
- It is unlikely that a single refrigerant will replace R-22, R-134a, R-404A, and R-410A. Most likely, the alternative replacement used will depend on the application.
- AREP focused on drop-in replacements and soft-optimized equipment. It is possible that further improvements could be attained by further soft-optimization or by full-optimization of the equipment for the alternative replacement.
- There were inconsistencies in the test results, which may have been caused by the comparison across product types, sizes, and manufacturers and from using different testing facilities.

Phase I did not initially plan to include high-ambient-temperature testing, but some testing parties chose to include high-ambient-temperature results in their reports. Phase II began in 2014 and includes new refrigerants and an increased focus on high-ambient-temperature testing. Twenty-five refrigerants were proposed for Phase II, of which 15 will be tested. Eight of the test plans include high-ambient-temperature conditions, and in total, nine alternative refrigerants are being tested at high ambient temperatures. The test matrix is listed in Tables 7 and 8. See Appendix A for relevant results to date. The results include data from one of the Phase II reports and from four of the Phase I reports.

For individual test reports and further details, please refer to the AREP website.[25]

---

[24] Wang, X. & Amrane, K. (2014) *AHRI Low Global Warming Potential Alternative Refrigerants Evaluation Program (Low-GWP AREP)*, JRAIA International Symposium 2014.

[25] For test reports and further details, refer to http://www.ahrinet.org/site/514/Resources/Research/AHRI-Low-GWP-Alternative-Refrigerants-Evaluation

Table 7. Low-GWP AREP Phase I high-ambient-temperature test matrix

Product	Test Companies	High-Ambient Conditions	Report	Baseline Refrigerant	R-1234yf	R32	D2Y60	L-41a	L-40	N-40b	AC5	N-13a	L-20	D-52Y
3.5 TR split system HP[a]	Lennox	46.1°C (115°F)	AREP Report 10	R-410A	×									
3.5 TR split system HP[a]	Lennox	46.1°C (115°F)	AREP Report 4	R-410A		×								
3 TR split system HP[a]	Uni. of Maryland	46.1°C (115°F)	AREP Report 20	R-410A		×	×	×						
3 TR split system HP[a]	Uni. of Maryland	46.1°C (115°F)	AREP Report 32	R-410A			×							
Ice machine (self-contained)	Manitowoc	43.3°C (110°F)	AREP Report 2	R-404A					×	×				
Ice machine (split system)	Manitowoc	48.9°C (120°F)	AREP Report 2	R-404A					×	×				
Bus AC system	ThermoKing	48.9°C (120°F)	AREP Report 12	R-134A							×	×		
Bus AC system	ThermoKing	48.9°C (120°F)	AREP Report 13	R-407C									×	×
VRF Multi-split HP[b]	Daikin	18.3°C (65°F), 20.0°C (68°F), 26.7°C (80°F), 35.0°C (95°F)	AREP Report 15	R-410A										×

[a] Results included in Appendix A.
[b] No high-ambient-temperature testing but included in Appendix A due to similarity with the units in this report.

Table 8. Low-GWP AREP Phase II high-ambient-temperature test matrix

Product	Test Companies	High-Ambient Conditions	Current Testing Status	Baseline Refrigerant	ARM-20b	ARM-71a	DR-5A	HPR2A	L-41-1	L-41-2	N-40c	R-32
10kW water chiller	Armines	46.1°C (115°F)	In testing	R-410A		×	×		×	×		
11.3 EER 10 TR rooftop unit	Carrier	51.7°C (125°F)	In testing	R-410A			×		×	×		×
14 SEER 3 TR HP	Carrier	51.7°C (125°F)	In testing	R-410A		×	×	×	×	×		
13 SEER 3 TR HP	Danfoss	46.1°C (115°F), 51.7°C (125°F)	In testing	R-410A			×			×		×
14 SEER 3 TR split AC[a]	Goodman	46.1°C (115°F), 51.7°C (125°F)	Completed	R-410A								×
Commercial package unit	Lennox	46.1°C (115°F) 51.7°C (125°F)	Completed	R-410A		×	×	×		×		×
Split ice machine	Manitowoc	48.9°C (120°F)	Completed	R-404A	×						×	
4 TR packaged rooftop unit	Trane	51.7°C (125°F)	In testing	R-410A			×					×
Rooftop packaged unit	Zamilac	51.7°C (125°F)	In testing	R-410A								×

[a] Results included in Appendix A.

3.2 PROMOTING LOW-GWP REFRIGERANTS FOR THE AIR-CONDITIONING SECTORS IN HIGH-AMBIENT-TEMPERATURE COUNTRIES (PRAHA)

PRAHA is a regional project approved by the Executive Committee of the Multilateral Fund and under implementation by UNEP and UNIDO. The primary objective of the project is to investigate sustainable refrigerant technologies for high-ambient-temperature countries. The project also aims to support technical and policy decisions, share information about demonstration projects, encourage the development of regional standards, and link regional energy efficiency to the adoption of a long-term low-GWP alternative.

The project includes three components: building and testing of prototypes, study of long-term feasible technologies, and coordination of the phase-out requirements with Minimum Energy Performance Standards (MEPS) programs. Seven local manufacturers from Saudi Arabia, Bahrain, Kuwait, and the UAE are participating in these efforts, as well as six international technology providers. Furthermore, PRAHA and AREP have created a joint declaration for the promotion of low-GWP alternatives and the exchange of relevant technical information. [26]

PRAHA is investigating a total of six refrigerants. Two are HFC/HFO blends meant to replace R-22, while two others are HFC/HFO blends designed to replace R-410A. R-32, an HFC, and R-290, commonly known as propane, are also being tested. The refrigerants are tested according to the conditions in ISO 5151. A 2 hr continuity test at 52°C is also performed. The products being tested are window room air conditioners, decorative split systems, ducted split systems, and packaged units. Each unit is a prototype specifically designed for a given refrigerant and capacity. Table 9 summarizes the PRAHA test plan.[27]

Table 9. PRAHA test plan summary

	Number of Prototypes per Unit Type						
	Window 220V/60Hz/ 1-phase	**Decorative 220V/60Hz/ 1-phase**	**Ducted 380V/50Hz/ 3-phase**	**Packaged 380V/50Hz/ 3-phase**	**Total Prototypes**	**# of Test Conditions per Prototype**	**Total # of Tests**
R-22	3	2	1	2	8	3	24
HFC Base	NA	2	2	NA	4	3	12
R32	NA	2	2	NA	4	3	12
HFO1	2	2	1	2	7	3	21
HFO2	2	1	1	1	5	3	15
R-290	1	1	NA	NA	2	3	6
				Total	**30**		**90**

See results to date in Appendix B.

---

[26] El-Talouny, A. & Nielsen, O. (2014) *Promoting Low-GWP Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA)*, UNEP and UNIDO, OzonAction Fact Sheet - November 2014.

[27] Elassaad, B. (2014) *Alternative Refrigerants for High-ambient Countries; Risk Assessment of Future Refrigerants in Production, Installation, and Service* Presentation at the 4th Symposium Low GWP Alternatives for High Ambient on October 28, 2014; Dubai, UAE; available: http://4thhighambient.com/presentations/4th%20sypm%20Day%201/Session-III%20P03%20UNEP-UNIDO%20PRAHA.pdf

3.3 EGYPTIAN PROGRAM FOR PROMOTING LOW-GWP REFRIGERANTS' ALTERNATIVES (EGYPRA)

EGYPRA is another regional project approved by the Executive Committee of the MLF and under implementation by UNEP and UNIDO. The project is focused on Egypt. Its objective is to test purpose-built ductless mini-split and central ducted air-conditioning prototypes using new refrigerants and compare their performance to R-22 and R-410A baseline units. Testing will include mini-splits with capacities of 3.5 kW (1 refrigeration ton, or TR), 5.25 kW (1.5 TR), and 7 kW (2 TR), and a 35 kW (10 TR) central air-conditioning system, with both common heat exchangers and with microchannel heat exchangers.

EGYPRA is investigating a total of eight types of refrigerants, three of which are HFC/HFO blends equivalent to R-22, three HFC/HFO blends equivalent to R-410A, as well as R-32 and R-290. Testing is done in accordance with EOS 4814 and EOS 3795 (ISO 5151) and will include data collection at T1 and T3 temperature conditions (refer to Table 5). Thirty-six prototypes are being built with dedicated compressors for each refrigerant. The project is currently building prototypes, and results are expected to be available early in 2016.[28]

See results to date in Appendix C.

3.4 COMPARISON OF HIGH-AMBIENT-TEMPERATURE TESTING PROGRAMS

The results from PRAHA and EGYPRA complement the test results of the ORNL test program because they are based on prototypes, while the ORNL test program is testing soft-optimized production units. Additionally, the ORNL test program includes refrigerants that were not tested in PRAHA or EGYPRA, further complementing the results from these programs. These programs also use multiple equipment types. While the ORNL evaluation program is focused on ductless mini-splits, EGYPRA also includes central air-conditioning units (including those with microchannel heat exchangers), and PRAHA includes window units, ducted units, and packaged units.

Low-GWP AREP also offers some results that complement the results from PRAHA, EGYPRA, and the ORNL evaluation program. However, many of the Low-GWP AREP tests did not cover high-ambient-temperature conditions on similar equipment and are therefore not directly applicable. Nevertheless, a number of test reports that have already been published included high-ambient-temperature testing, and more are expected in Phase II. The results to date that are relevant to this report are discussed in Appendix A.

4. EXPERIMENTAL FACILITIES AND EQUIPMENT

4.1 MINI-SPLIT AIR-CONDITIONING UNITS

ORNL performed drop-in tests for two baseline mini-split systems: a 5.25 kW_{th} (1.5 TR) R-22 system and a 5.25 kW_{th} (1.5 TR) R-410A system. The R-22 unit is from the product family Eco Plus, model number 42KHRT18-308. The corresponding condensing unit is 38MKR18US30-03. The R-410A unit is from the product family Xpression Elite, model number 42KHL0183P. The corresponding condensing unit is 38KHL0183. Figure 1 shows the two baseline units provided by Carrier.

---

[28] UNEP and UNIDO presentation on EGYPRA, June 2015.

- Carrier has donated two Split Condensing Units (60 Hz)
  - R-22 unit
  - EER 9.5
  - R-410A unit
  - EER 11.5

Figure 1. Baseline equipment provided by Carrier designed for high-ambient-temperature conditions.

Note: Detailed specifications are not available at this time. ORNL is continuing its outreach to gather additional technical information on the units under evaluation.

4.2 REFRIGERANTS

4.2.1 Refrigerants for the R-22 Unit

The panel selected five alternative refrigerants for testing in the R-22 unit. In addition, ORNL's evaluation schedule allowed for time to test DR-93, a refrigerant with a higher GWP than the five primary alternatives but whose results may also provide useful insights. Table 10 shows the details for each of the five primary alternative refrigerants, the baseline (R-22), and DR-93.[29] The refrigerant manufacturers provided the data to ORNL via NIST REFPROP files. All of the selected alternatives are ASHRAE Standard 34 safety class A refrigerants, meaning they are of low toxicity. For flammability, the alternatives include ASHRAE safety class 1, 2L, and 3, where higher numbers indicate higher flammability. Class 2L is a subgroup of mildly flammable class 2 refrigerants with a maximum burning velocity of 10 cm/sec.

The charge size is determined during the soft-optimization process. See Section 5 for an additional discussion of the process.

The expert panel did not recommend a priority order for conducting these tests. The panel strongly recommended, however, that the baseline refrigerant (R-22) be tested again upon completion of all the alternatives in order to ensure that the unit's operating conditions remain unchanged.

The R-22 unit is being tested both with mineral oil (ATMOS M60 (A)), the OEM-specified lubricant for use with R-22, and with POE oil (ISO 68), the lubricant used for the alternative refrigerants. The expert panel's consensus recommendation was that R-22 with mineral oil be the baseline since that is how the unit was designed and shipped by the manufacturer. (See Appendix E for results of R-22 with POE oil.)

---

[29] For thermodynamic cycle calculations for each alternative refrigerant compared to R-22, refer to the *Low-GWP AREP Participants' Handbook* (April 17, 2015) by the Air-Conditioning, Heating and Refrigeration Institute, pages 70 (R-290), 86 (ARM-20B), 89 (DR-3), 90 (L-20A and DR-93) and 91 (N-20B). Available: http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/Participants_Handbook2015-04-17.pdf

Table 10. Baseline and alternative refrigerant data for the R-22 unit

Refrigerant	Manufacturer	ASHRAE Safety Class	GWP[a]		Charge Mass kg (oz.)	CO_2 Equivalent (AR5) kg
			AR4	AR5		
R-22 (Baseline)	-	A1	1,810	1,760	1.417 (50)	2,495
N-20B[b]	Honeywell	A1	988	904	2.087 (73.6)	1,886
DR-3[b]	Chemours	A2L	148	146	2.007 (70.8)	293
ARM-20B[b]	Arkema	A2L	251	251	1.588 (56)	398
L-20A (R-444B)[b]	Honeywell	A2L	295	295	1.568 (55.3)	462
DR-93[b]	Chemours	A1	1,258	1,153	1.828 (64.5)	2,108
R-290	-	A3	3	3	*To be included in the final report*	

[a] Sources: IPCC AR4, 2007[30]; IPCC AR5, 2013[31]

[b] GWP values for refrigerant blends not included in IPCC reports are calculated as a weighted average using manufacturer-supplied compositions.

4.2.2 Refrigerants for the R-410A Unit

Table 11 shows the five R-410A alternatives selected by the panel for evaluation.[32] All these selected alternatives for this unit are ASHRAE safety class A2L (nontoxic, mildly flammable).

Table 11. Baseline and alternative refrigerant data for the R-410A unit

Refrigerant	Manufacturer	ASHRAE Safety Class	GWP [a]		Charge Size kg (oz)	CO_2 Equivalent (AR5) kg
			AR4	AR5		
R-410A (Baseline)	-	A1	2088	1924	*To be included in the final report*	
ARM-71A[b]	Arkema	A2L	460	461		
R-32	Daikin	A2L	675	677		
DR-55[b]	Chemours	A2L	698	676		
L-41 (R-447A)[b]	Honeywell	A2L	583	572		
HPR-2A[b]	Mexichem	A2L	600	593		

[a] Sources: IPCC AR4, 2007[30]; IPCC AR5, 2013[31]

[b] GWP values for refrigerant blends not included in IPCC reports are calculated as a weighted average using manufacturer-supplied compositions.

---

[30] IPCC, 2007: *Climate Change 2007: The Physical Science Basis. Contribution of Working Group I to the Fourth Assessment Report of the Intergovernmental Panel on Climate Change* [Solomon, S., D. Qin, M. Manning, Z. Chen, M. Marquis, K.B. Averyt, M. Tignor and H.L. Miller (eds.)]. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA; section 2.10.2: Direct Global Warming Potentials. Available: https://www.ipcc.ch/publications_and_data/ar4/wg1/en/contents.html

[31] IPCC, 2013. Myhre, G., D. Shindell, F.-M. Bréon, W. Collins, J. Fuglestvedt, J. Huang, D. Koch, J.-F. Lamarque, D. Lee, B. Mendoza, T. Nakajima, A. Robock, G. Stephens, T. Takemura and H. Zhang, 2013: Anthropogenic and Natural Radiative Forcing. In: *Climate Change 2013: The Physical Science Basis. Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change* [Stocker, T.F., D. Qin, G.-K. Plattner, M. Tignor, S.K. Allen, J. Boschung, A. Nauels, Y. Xia, V. Bex and P.M. Midgley (eds.)]. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA. Available: https://www.ipcc.ch/pdf/assessment-report/ar5/wg1/WG1AR5_Chapter08_FINAL.pdf

[32] For thermodynamic cycle calculations for each alternative refrigerant compared to R-410A, refer to the *Low-GWP AREP Participants' Handbook* (April 17, 2015) by the Air-Conditioning, Heating and Refrigeration Institute, pages 56 (R-32), 83 (ARM-71A), 84 (DR-55), 85 (L-41) and 86 (HPR-2A). Available: http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/Participants_Handbook2015-04-17.pdf

The amount of refrigerant charge is based upon the value determined during the soft-optimization process. See Section 5 for additional discussion of the process.

The expert panel did not recommend a priority order for conducting these tests. The panel strongly recommended, however, that the baseline refrigerant (R-410A) be tested again upon completion of all alternatives, in order to ensure that the unit's operating conditions remain unchanged.

4.3 EXPERIMENTAL FACILITIES

The ORNL Multi-Zone Environmental Chambers, shown in Figure 2, were used for this project. This facility characterizes the performance of multi-zone electric or gas HVAC systems for residential and light commercial use. The "outdoor" chamber is 6.1 × 4.6 m (20 × 15 ft.); the 8.5 m (28 ft.) square "indoor" chamber can be divided into up to four spaces controlled at different conditions to represent separate zones. Dry-bulb temperature can be controlled at −23 to 55°C (−10 to 131°F) and relative humidity at 30 to 90%. Utilities include 480 V, three-phase power at 225 A with step-down to 240, 208, and 120 V. In this project, the indoor side was split into two chambers, each 8.5×4.25 m such that we can evaluate two systems in parallel. The chambers are equipped with two code testers—one that can supply and measure airflow up to 5100 m^3/hr (3000 cfm) and the other up to 11,900 m^3/hr (7000 cfm). The code testers have the required duct mixers and temperature sampling trees.

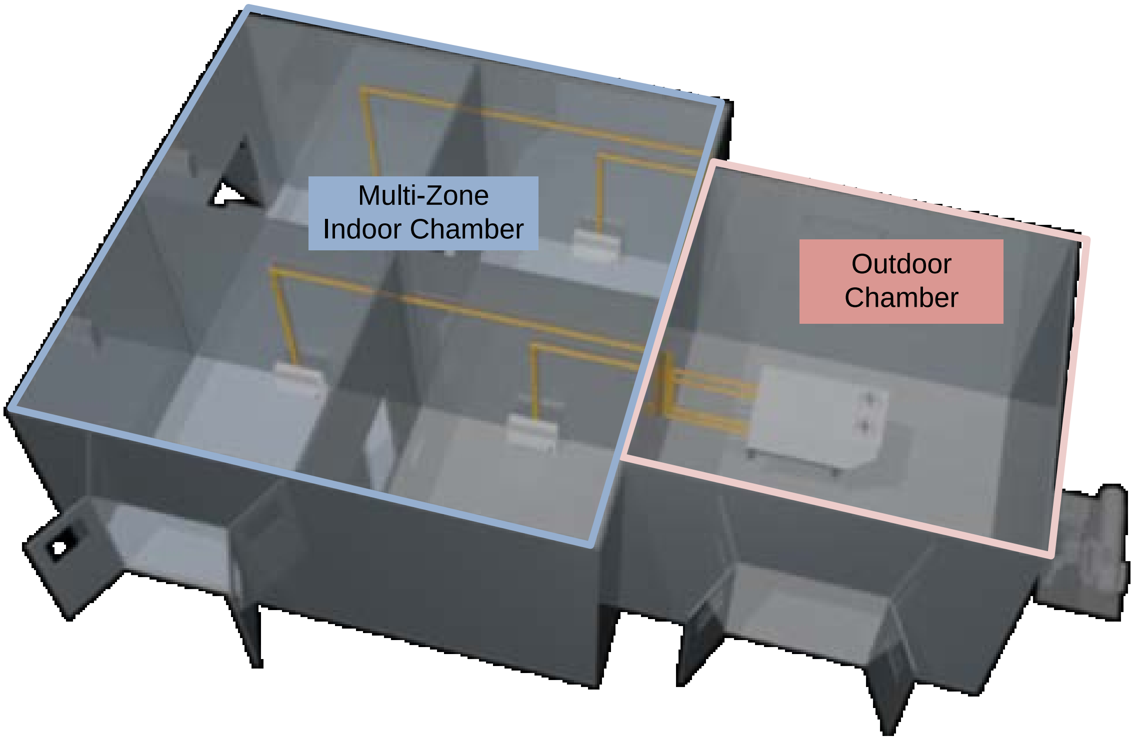


Figure 2. Multi-zone environmental chambers.

4.4 EXPERIMENTAL SETUP AND INSTRUMENTATION

A comprehensive experimental facility was designed and built to comply with ANSI/AHRI Standard 210/240-2008 and ANSI/ASHRAE Standard 37-2009. The Air Enthalpy method is used to evaluate the performance of the indoor unit, and the Refrigerant Enthalpy Method is used as a secondary means of evaluating the system performance in order to establish energy balance and assess measurement accuracy. For an overview of the experiment test setup, refer to Appendix D.

Table D.1 in Appendix D summarizes the instrumentation used on the baseline R-22 unit. All of the instrumentation provides better accuracy than required by ASHRAE Standard 37-2009 (Table 2b). The data are collected to satisfy Table 3 of the ASHRAE Standard 37-2009 for both the Indoor Air Enthalpy Method column and the Refrigerant Enthalpy Method column. Additional data were recorded to increase the level of understanding of the alternative refrigerants, including compressor shell temperature and additional surface thermocouples on the liquid line and the compressor suction line.

4.5 ALTERNATIVE REFRIGERANT EVALUATION EXPERIMENTAL DESIGN

ORNL is evaluating the R-22 unit with five primary alternative refrigerants plus DR-93, which ORNL added to the test plan as a secondary priority for testing only where the schedule allowed. ORNL is also evaluating the R-410A unit with five alternative refrigerants. For each alternative refrigerant, ORNL is evaluating the performance at six test conditions each, for a total of 66 tests with alternative refrigerants. Additionally, each unit is being tested with its baseline refrigerant 12 times (six times before and six times after testing the alternative refrigerants). Consequently, the total number of tests is 90. Table 12 summarizes the test plan.

Table 12. ORNL test plan summary[a]

	Base	N-20B	DR-3	ARM-20B	L-20A (R-444B)	DR-93	R-290	ARM-71A	R-32	DR-55	L-41 (R-447A)	HPR-2A	Total # of Tests
R-22 Unit	I	I	I	I	I	I[b]	F						**48**
R-410A Unit	F							F	F	F	F	F	**42**

[a] I = Completed for the interim report; F=To be completed for the final report

[b] DR-93 was not included in the original list of priority alternative refrigerants but was added only when ORNL's test schedule permitted as a potential opportunity for gathering additional insights.

5. EXPERIMENTAL PROCEDURE

5.1 OVERALL PROCEDURE

ORNL followed ANSI/ASHRAE Standard 37-2009 to test 2 mini-split 5 kW_{th} (1.5 TR) air-conditioning systems designed for high-ambient-temperature conditions. The first unit is an R-22 unit, and the second unit is designed for R-410A. The indoor side of the multi-zone environmental chambers was divided into two chambers. Each indoor chamber houses the indoor unit of the corresponding mini-split system connected with the associated air-enthalpy tunnel, while both outdoor units are installed in the outdoor chamber. The following steps were taken to evaluate the equipment and refrigerant combinations.

1. Perform charge optimization at the AHRI A conditions.[33] See Section 5.2.

---

[33] ORNL performed charge optimization at AHRI A conditions (35°C [95°F] outdoor and 26.7°C [80.0°F] indoor) because it is the closest of the test conditions in this study to manufacturers' reported rating conditions (ISO T1 conditions – 35°C [95°F] outdoor and 27°C [80.6°F] indoor). It is assumed that this is therefore also the condition for which manufacturers do their system design and analysis.

2. At the optimum charge, evaluate the performance at T3 conditions. If adequate subcooling and superheat are available, proceed with testing. Otherwise, adjust the charge to ensure 100% liquid entering the capillary tube, based on the mass flow meter measurement, and avoid evaporator flooding.

3. Run the test matrix (each refrigerant at each test condition) as summarized in Table 12. Collect steady-state data for 30 min at each condition.[34] If the difference in dew point temperature between the inlet and outlet of the indoor-airside is less than 1.7°C (3°F), rely on the water condensate measurement for latent capacity.[35]

4. To ensure system performance is maintained over the test period, the unit is retested with the baseline refrigerant to verify the system performance stability after finishing all alternative refrigerant tests.

The baseline units were modified slightly to allow for adequate instrumentation. First, the liquid line was diverted to outside the outdoor unit housing to allow installation of the Elite Coriolis mass flow meter (CMF025). A capillary tube header was also placed after the Coriolis mass flow meter, where up to five capillary tubes can be tested. Finally, pressure and in-stream thermocouples (for the R-22 unit) or in-stream RTD (for the R-410A unit) were used to evaluate the refrigerant enthalpy at the liquid line just before the capillary tube and before and after the evaporator. In addition, a capillary tube header was used to select appropriate capillary tubes for the alternative refrigerants. See Figure 3 for a high-level schematic diagram of the system and Appendix D for more details.

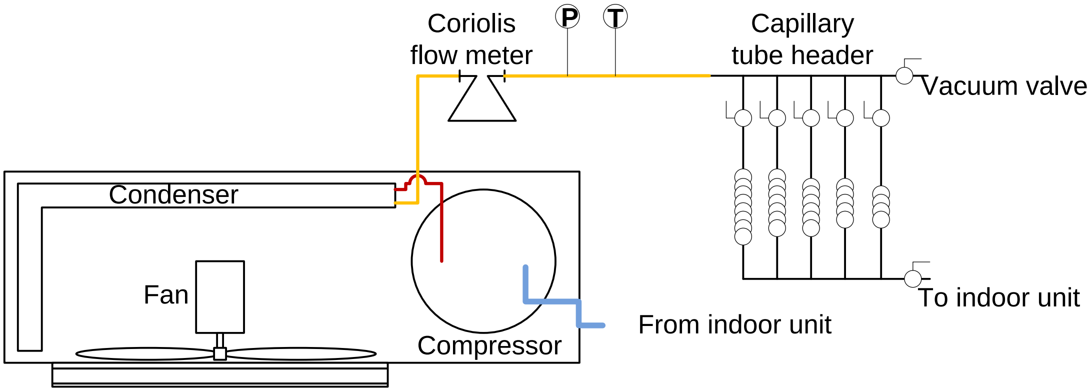


Figure 3. Outdoor unit configuration.

5.2 PROCESS FOR SOFT-OPTIMIZATION

ORNL's soft-optimization process consisted primarily of adjusting the capillary tube to control flow to the evaporator in the indoor unit and performing charge-mass optimization. The baseline capillary tube had an inner diameter (ID) of 2.00 mm (0.079 in.), an outer diameter (OD) of 3.2 mm (1/8 in.), and a length of 508 mm (20 in.). Unfortunately, at the time of testing, ORNL was only able to obtain capillary

---

[34] Steady state in this case refers to the temperature conditions in the test chamber (Table 5); the temperatures must be maintained within ±0.5°C; if conditions are not maintained, the investigators wait until stabilization is achieved before commencing data collection.

[35] When the dew-point temperature difference is less than 1.7°C (3°F), the latent capacity calculation uncertainty becomes significant. Hence we rely on the water condensate measurement for latent capacity instead.

tubes with an ID of 1.65 mm (0.065 in.). As such, ORNL first identified the appropriate alternative capillary tube length to be 254 mm (10 in.). The same capillary tube spool with ID of 1.65 mm (0.065 in.) was used for the soft-optimization and selection of the appropriate flow restriction. The processes for capillary tube length optimization and charge optimization are as follows (see Table 13 for nomenclature).

1. Size capillary tubes using appropriate correlation and fabricate. (±50.8 mm (2 in)., ±25.4 mm (1 in)., and exact size per calculations)[36]
2. Charge the system with $M_{opt,ref\#} = M_{opt,ref\#} * (\rho_{ref\#,\,liq} / \rho_{R\text{-}22liq})$
3. Run charge optimization campaign at the AHRI A condition: collect steady-state[37] data for 10 minutes at each of the following conditions.
   a) $M_{opt,ref\#}$ and exact capillary tube length
   b) $M_{opt,ref\#}$ and 25.4 mm (1 in). shorter capillary tube length; if better performance, proceed to c), or else proceed to d)
   c) $M_{opt,ref\#}$ and 50.8 mm (2 in). shorter capillary tube length, skip to f)
   d) $M_{opt,ref\#}$ and 25.4 mm (1 in). longer capillary tube length; if better performance, proceed to e), or else proceed to f)
   e) $M_{opt,ref\#}$ and 50.8 mm (2 in). longer capillary tube length
   f) Add/subtract refrigerant charge (approximately 2 oz. at a time), go back to a). NOTE: if refrigerant is to be removed from the unit, it is done by removing liquid (as indicated by the mass flow meter) and returning it to the same cylinder to avoid fractionation.
4. Run the unit with $M_{opt,ref\#}$ and the selected capillary tube at T3 conditions to ensure adequate subcooling and superheating; if not, adjust the charge accordingly (approximately 57 g (2 oz.) at a time with 10 min of steady-state data collected).
5. Evaluate the system performance for all test conditions listed in Table 5.

Table 13. Test process nomenclature, subscripts and symbols

Symbol	Meaning
M	Refrigerant Charge
opt	Optimized
ref#	Alternative Refrigerant
liq	Liquid
R-22	Refrigerant
ρ	Density

Soft-optimization, as defined by Low-GWP AREP, allows for minor refinements of the system being tested with a particular alternative refrigerant, provided commonly available components are used. Examples of potential changes include “compressor displacement and/or motor size; flow control; heat transfer circuiting; use of a liquid-line/suction-line heat exchanger; amount of refrigerant charge; use of a variable speed compressor motor; diameter / size of the tubing to adapt to the refrigerant volume flow and

---

[36] *ASHRAE Refrigeration Handbook*, Chapter 11 – Refrigerant Control Devices, Section Adiabatic Capillary Tube Selection Procedure, 2014. For details, see Wolf, D.A., R.R. Bittle and M.B. Pate, *Adiabatic capillary tube performance with alternative refrigerants*, ASHRAE Research Project RP-762, Final Report, 1995.

[37] Steady state is established when the average dry-bulb temperatures at the inlet of the indoor and outdoor heat exchangers are within 0.28°C (0.5°F) of the desired conditions, and the individual readings of each instrument at the inlet and outlet of each heat exchanger are within 0.56°C (1.0°F) of the average values of these quantities. Furthermore, the average wet-bulb temperature at the inlet of the indoor heat exchanger must be within 0.17°C (0.3°F) of the desired conditions with the individual readings within 0.56°C (1.0°F) of the average value, and the airflow rate must be within 1% of the desired value.

pressure drop; size of accumulators; and lubricant. In addition, the heat transfer area of the soft-optimized system's evaporator and condenser may be changed, provided that the sum total area remains the same as the baseline system."

With fewer modifications, the testing may be considered "drop-in" testing, which can include "only minor modifications, if any" and may include optimization of refrigerant charge quantity, adjustment of expansion device (if adjustable), and adjustment of compressor speed.[38]

As part of the ORNL evaluation program, the lubricants were changed, charges optimized, and capillary tube/flow control changes implemented. Additional changes, still within the scope of "soft-optimization," could be made for future testing, with the potential to increase performance.

5.3 PROCESS FOR CHANGING REFRIGERANTS AND THE LUBRICANTS

The following steps were followed to change refrigerants between sets of tests.

1. The refrigerant is reclaimed in either the original cylinders or empty cylinders.
2. The system is put under vacuum for an extended period of time (minimum of 3 hr) to ensure all the refrigerant is dissolved from the oil; a vacuum gauge is used to ensure system is evacuated to 300 microns.
3. Refrigerant is slowly charged from the liquid port through the refrigerant suction line.

In the case of the R-22 unit, further modifications were required after baseline testing and prior to initiating testing of the alternative refrigerants.

1. Replace mineral oil with POE oil
2. Replace capillary tube with a capillary tube tree consisting of three lengths (exact length and ±50.8 mm (2 in). of the exact length)
3. Adjust the refrigerant charge: $M_{opt} = M_{opt,R\text{-}22} + \rho_{R\text{-}22liq} * V_{cap\text{-}tube\ header}$
4. Run test at AHRI A conditions with similar capillary tube; compare performance with baseline performance (if COP and capacity are within ±5%, skip to #7)
5. Compare the performance with the other capillary tubes (shorter and longer)
6. Use the capillary tube that provides the closest performance to the baseline refrigerant and run charge optimization
7. Run test at T3 conditions
8. Drain oil (POE + traces of mineral oil)
9. Charge with fresh POE oil
10. Evacuate system to 300 microns for at least 3 hr and proceed with the alternative refrigerant evaluation

6. SYSTEM MODELING

For incorporation in the final report.

---

[38] AHRI, *Participants' Handbook: AHRI Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP)*, April 17, 2015, pp24-25; available: http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/Participants_Handbook2015-04-17.pdf

7. RESULTS AND DISCUSSION

7.1 R-22 UNIT RESULTS

ORNL's first round of evaluation (i.e., those evaluations conducted for this interim working paper) included tests of one baseline refrigerant (R-22, using mineral oil as a lubricant) and five alternative refrigerants in the R-22 unit: N-20B, DR-3, ARM-20B, L-20A (R-444B), and DR-93, all using POE oil as a lubricant. R-290 test data are not available at this time but will be included in the final report. As discussed in Section 4.2.1, the expert panel recommended using R-22 with mineral oil as the baseline; all results in this section reflect this decision. In order to evaluate the unit durability after the retrofit, a partial test with R-22/POE oil was performed before testing any of the alternative refrigerants and then a repeat test with R-22/POE oil was performed after testing the five primary alternative refrigerants (DR-93 tests were conducted after the repeat R-22/POE tests once the test chamber became available – see Section 4.2 for additional discussion). The results of the tests with R-22/POE are included only in Appendix E, together with detailed results for all the refrigerants.

This section describes the results at all test conditions (AHRI A and B, ISO T3, T3*, Hot, and Extreme) and then presents the results for the AHRI A, ISO T3, Hot, and Extreme test conditions in more detail to show performance trends as ambient temperature changes.

Based on the uncertainty analysis described in Section 7.3, the air-side capacity and COP have an uncertainty of ±3.6%. Thus, refrigerants whose performance values are within 5% of the baseline may be considered a good match to R-22, while refrigerants within 10% of the baseline may only require additional soft-optimization to achieve the same performance as the baseline.

Table 14 summarizes the results of testing the baseline refrigerant and alternatives in the R-22 unit at moderate ambient temperatures (AHRI B and A conditions).

Table 14. Test results at moderate ambient temperatures (performance change from baseline in parentheses)[a]

	AHRI B Outdoor: 27.8°C (82°F)		**AHRI A** Outdoor: 35.0°C (95°F)	
	COP	**Capacity**	**COP**	**Capacity**
R-22 (Baseline)	3.48	6.26	3.07	6.10
N-20B	3.04 (-12.6%)	5.42 (-13.4%)	2.68 (-12.5%)	5.25 (-14.0%)
DR-3	2.88 (-17.1%)	5.52 (-11.8%)	2.57 (-16.1%)	5.4 (-11.5%)
ARM-20B	3.06 (-12.2%)	6.05 (-3.3%)	2.71 (-11.8%)	5.91 (-3.1%)
L-20A (R-444B)	3.02 (-13.3%)	5.53 (-11.6%)	2.72 (-11.3%)	5.58 (-8.6%)
DR-93	3.00 (-13.7%)	5.92 (-5.4%)	2.63 (-14.2%)	5.7 (-6.6%)

[a] Shading legend – blank: < 5% degradation; yellow: 5 – 10% degradation; orange: >10% degradation

Table 15 summarizes the results of testing the baseline refrigerant and alternatives in the R-22 unit at high ambient temperatures.

Table 15. Test results at high ambient temperatures (performance change from baseline in parentheses)[a]

	Hot Ambient Outdoor: 52°C (125.6°F)		**Extreme Ambient** Outdoor: 55°C (131°F)	
	COP	**Capacity**	**COP**	**Capacity**
R-22 (Baseline)	1.98	5.00	1.82	4.76
N-20B	1.77 (-10.7%)	4.26 (-14.8%)	1.64 (-9.7%)	4.10 (-13.9%)
DR-3	1.70 (-14.3%)	4.41 (-11.7%)	1.55 (-14.8%)	4.21 (-11.7%)
ARM-20B	1.76 (-11.2%)	4.84 (-3.2%)	1.61 (-11.2%)	4.62 (-3.0%)
L-20A (R-444B)	1.85 (-7.0%)	4.79 (-4.1%)	1.69 (-6.9%)	4.59 (-3.7%)
DR-93	1.70 (-14.2%)	4.63 (-7.4%)	1.54 (-14.9%)	4.38 (-8.1%)

[a] Shading legend – blank: < 5% degradation; yellow: 5–10% degradation; orange: >10% degradation

Figure 4 shows the cooling COP for each refrigerant at each test condition. For all refrigerants, the efficiency degrades with increases in ambient temperature. The percentage of efficiency degradation associated with increasing ambient temperature is roughly consistent across all alternative refrigerants; the cooling COP degrades approximately 40% as the ambient temperature increases from AHRI A to Extreme conditions. For detailed data on cooling COP relative to the cooling COP at AHRI A conditions as a function of ambient temperature, refer to Figure E.3 in Appendix E. At all test conditions, the baseline, R-22, yielded a higher COP than the alternative refrigerants, as expected; the equipment is not purpose-built for the alternative refrigerants and, as a result, is not expected to perform at the same levels.

At the Extreme test conditions, R-444B fared best of all the alternative refrigerants with a COP 6.9% lower than the baseline refrigerant. ARM-20B, N-20B, and DR-93 had an efficiency similar to that of R-444B at moderate temperature conditions but degraded more at higher ambient temperatures; their COPs were, respectively, 11.2, 9.7, and 14.9% lower than the baseline under the Extreme test conditions, compared to the aforementioned 6.9% for R-444B.

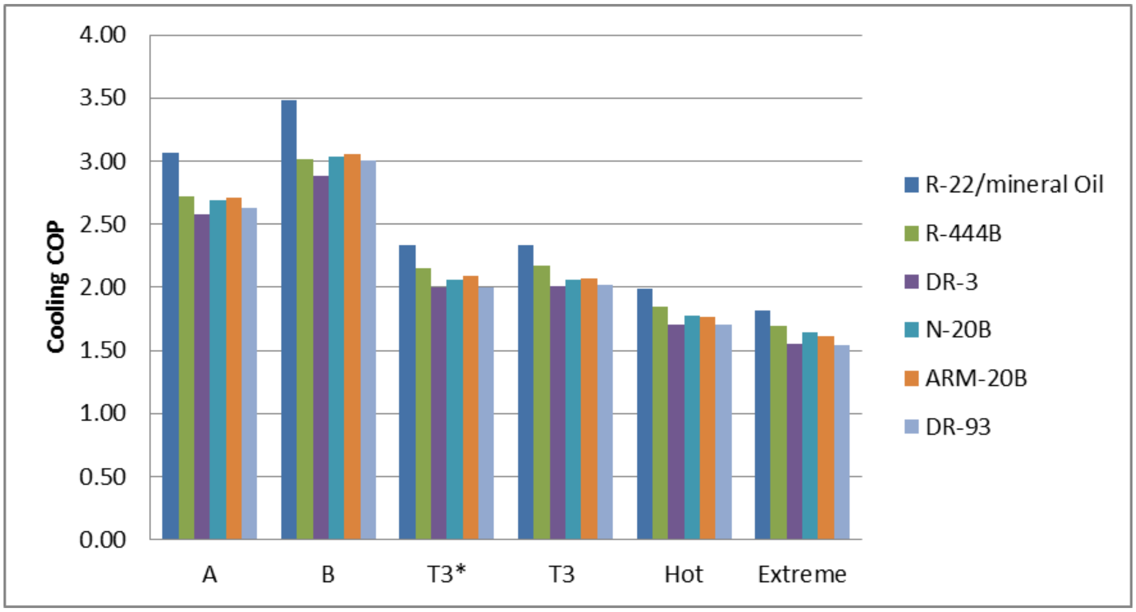


Figure 4. Cooling COP for each refrigerant at each test condition.

Figure 5 shows the cooling capacity for each refrigerant at each test condition. For all tested refrigerants, the cooling capacity degraded as the ambient temperature increased. The amount of capacity degradation varies by refrigerant (for a detailed view of cooling capacity as a function of ambient temperature, refer to Figure E.4 in Appendix E). At all test conditions, the baseline refrigerant yielded a higher cooling capacity than the alternatives. ARM-20B provided the highest cooling capacity of all the alternatives at moderate temperature conditions, with only a 3% loss from the baseline, followed closely by DR-93 with a 6% loss from the baseline. Under high ambient temperatures (both Hot and Extreme), R-444B and ARM-20B yielded similar capacities (4% and 3% degradation compared to the baseline, respectively). DR-93 performed 7.4 and 8.1% below the capacity of the baseline at Hot and Extreme conditions, respectively. ARM-20B yielded the highest capacity of all alternative refrigerants at each test condition.

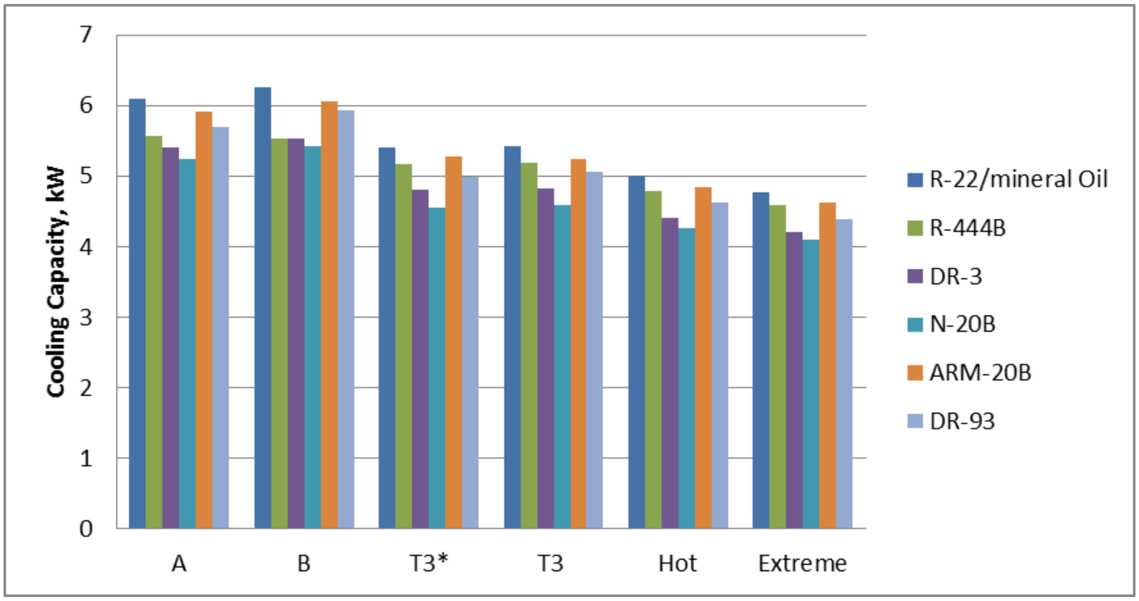


Figure 5. Cooling capacity for each refrigerant at each test condition.

Figure 6 compares the cooling COP and capacity of the alternative refrigerants to the baseline under the AHRI A test conditions. ARM-20B nearly matched the cooling capacity of the baseline (3% degradation), though with a 12% drop in cooling COP. The COP was approximately 11% lower for R-444B when compared to the baseline, while its cooling capacity was more than 90% of the baseline. DR-93 achieved 93% of the capacity of the baseline, but only 86% of the COP. DR-3 and N-20B both yielded less than 90% of both the COP and of the capacity of the baseline.

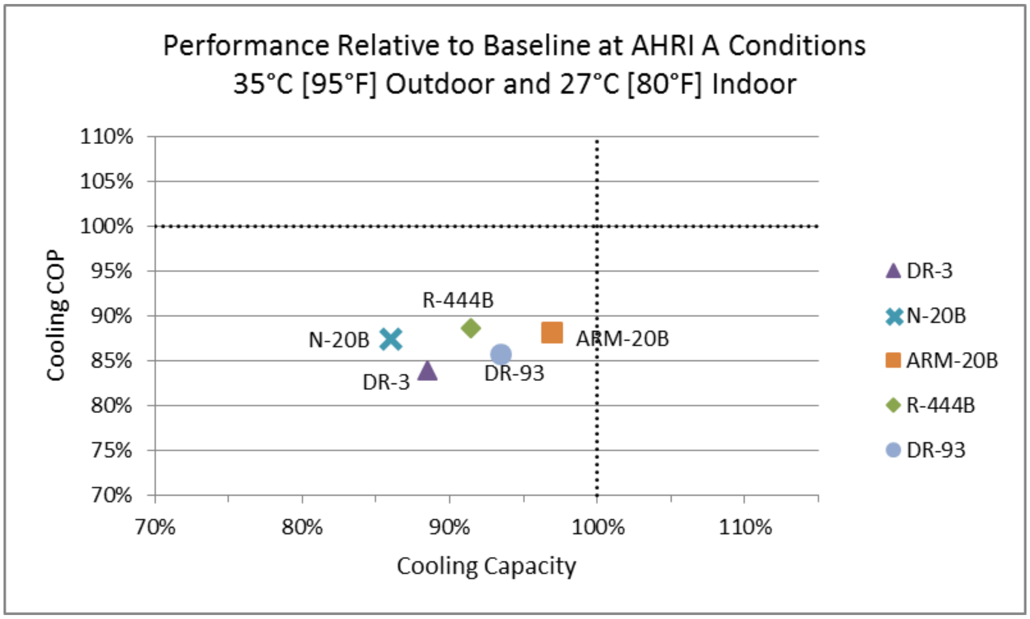


Figure 6. Performance of alternative refrigerants compared to R-22 (mineral oil) at AHRI A test conditions (outdoor temperature 35°C and indoor temperature 27°C).

Figure 7 compares the cooling COP and capacity of the alternative refrigerants to R-22 (mineral oil) under the ISO T3 test conditions. Under the ISO T3 test conditions, where the outdoor temperature is 11°C higher than under the AHRI A conditions, ARM-20B and R-444B performed very close to the baseline in terms of capacity (higher than 95% of the baseline capacity), but their cooling COP was still 11 and 7% below that of the baseline, respectively. The relative performance of DR-93 and N-20B compared to the baseline remained approximately the same as in AHRI A conditions, both in terms of COP and capacity, while the COP of DR-3 improved slightly relative to the baseline.

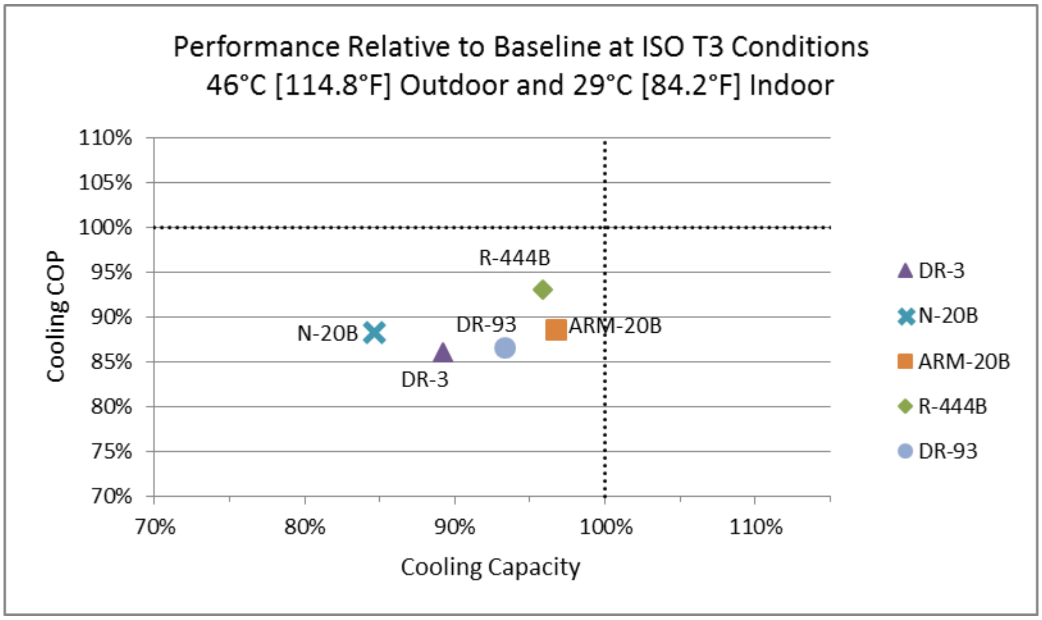


Figure 7. Performance of alternative refrigerants compared to R-22 (mineral oil) at ISO T3 (outdoor temperature 46°C and indoor temperature 29°C).

Figure 8 compares the cooling COP and capacity of the alternative refrigerants to R-22 (mineral oil) under the Hot test conditions. Under the Hot test conditions, the relative performance compared to the baseline remained approximately constant for all alternative refrigerants, both in terms of capacity and COP.

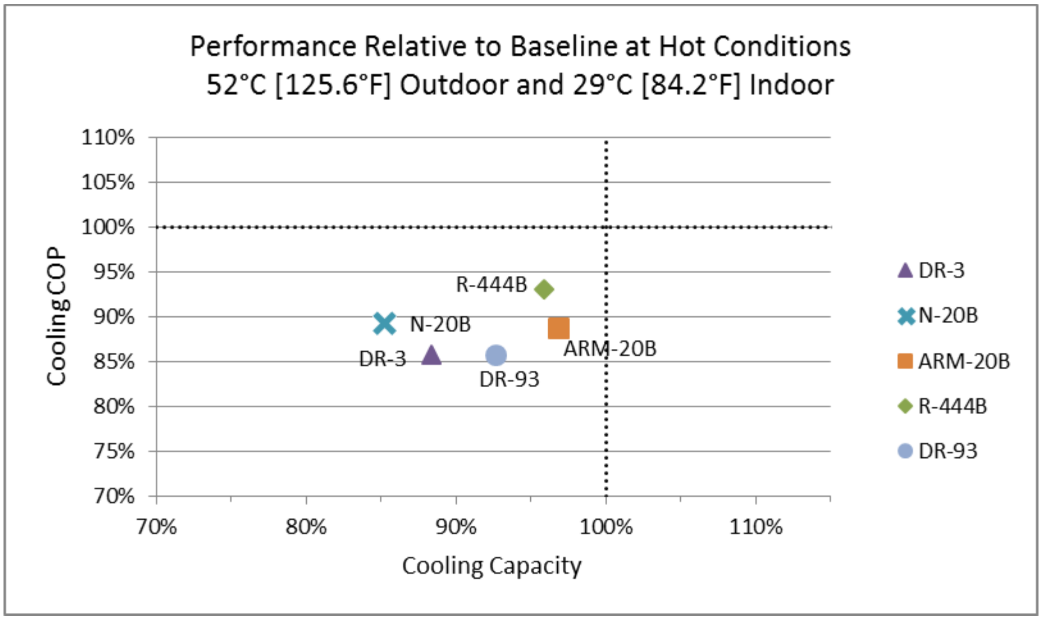


Figure 8. Performance of alternative refrigerants compared to R-22 (mineral oil) at Hot test conditions (outdoor temperature 52°C and indoor temperature 29°C).

Figure 9 compares the cooling COP and capacity of the alternative refrigerants to R-22 (mineral oil) under the Extreme test conditions. ARM-20B yielded a higher cooling capacity but lower COP than R-444B, a trend which was also apparent under T3 and Hot conditions but not as clear under the AHRI A conditions. For the AHRI A, T3, Hot, and Extreme conditions, DR-3 yielded a higher cooling capacity and lower COP than N-20B. The relative COP of DR-93 compared to the baseline remained roughly constant throughout all test conditions, but the results indicate a marginal drop in capacity relative to the baseline.

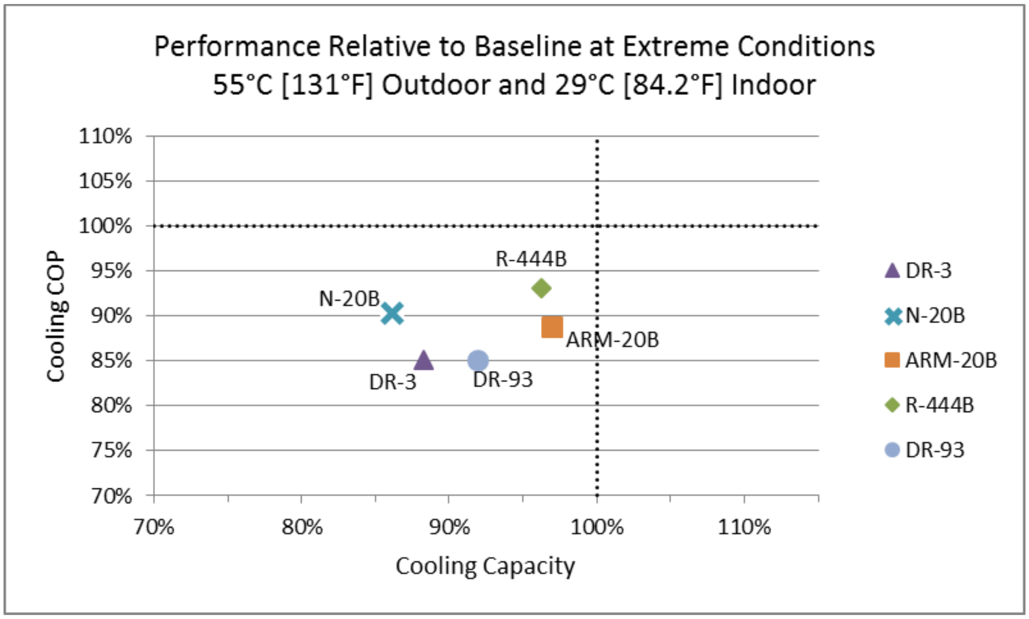


Figure 9. Performance of alternative refrigerants compared to R-22 (mineral oil) at Extreme test conditions (outdoor temperature 55°C and indoor temperature 29°C).

See Appendix E for additional results, including detailed data tables.

7.2 R-410A UNIT RESULTS

ORNL’s first round of testing (i.e., those tests included in the interim working paper) did not include any tests of the R-410A unit. The final report will include results of all testing of the R-410A unit, including all baseline refrigerant data and all alternative refrigerant data.

Appendix F will include detailed tables of operating temperatures and pressures.

7.3 ERROR ANALYSIS

The experimental uncertainty was calculated based on the uncertainties of each of the measured variables which are propagated into the value of the calculated quantity. The method for determining this uncertainty propagation is described in NIST Technical Note 1297.[39] Assuming the individual measurements are uncorrelated and random, the uncertainty in the calculated quantity can be determined as

$$U_Y = \sqrt{\sum_i \left(\frac{\partial Y}{\partial X_i}\right)^2 {U_{X_i}}^2}\ ,$$

where Y is the calculated quantity and X_i is the measured variable.

Based on the uncertainty analysis, the air-side capacity has an uncertainty of ±3.6% and the air-side COP also has an uncertainty of 3.6%. As such, refrigerants whose performance values are within 5% of the

---

[39] Taylor B.N. and Kuyatt, C.E., *Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results*, National Institute of Standards and Technology Technical Note 1297, 1994.

baseline should be a good match to R-22. Furthermore, values within 10% of the baseline should indicate an acceptable match that requires incremental soft-optimization to reach parity with R-22 performance.

8. PRELIMINARY CONCLUSIONS

As expected, the absolute performance of all refrigerants, including the baseline refrigerant, degraded as outdoor temperature increased. The relative performance of each of the alternative refrigerants compared to the baseline is generally maintained with increases in outdoor temperature. R-444B shows the lowest performance reduction of all tested refrigerants, relative to the baseline. Figures 7–9 show that the performance of R-444B approaches that of the baseline (R-22 with mineral oil) at the higher ambient conditions (T3, Hot, and Extreme). In other words, while all refrigerants, including the baseline refrigerant, performed progressively worse as the ambient temperature increased, the drop in performance relative to the baseline refrigerant was less pronounced for R-444B.

These efficiency and capacity values would be expected to improve through design modifications that manufacturers would conduct prior to introducing a new product to market. However, given that the scope of this study only covered soft-optimized testing, no detailed assessment can be made as to the extent of potential improvements through design changes. Within the bounds of what is possible as far as optimization for soft-optimized tests, the ORNL test plan includes only minor optimizations, including refrigerant charge optimization, capillary tube length optimization, and lubricant change, which likely indicate that these are conservative results. It can be expected that further soft-optimization would yield better performance results, and further optimization beyond that, as would be performed for purpose-built units, would likewise increase those results.

This is an interim working paper that presents the progress of the program thus far and other testing programs. The results discussed here represent only part of a complete set of results, which will be presented in the final report.

9. ACKNOWLEDGEMENT

We would like to acknowledge Dr. Patrick Phelan, Emerging Technologies Program Manager at the U.S. Department of Energy Building Technologies Office; Dr. Suely Machado Carvalhofor, adviser to the Superintendent at the Instituto de Pesquisas Energéticas e Nucleares, IPEN-CNEN, Ministry of Science, Technology and Innovation, MCTI, São Paulo, Brazil; and Ms. Jaqueline Wong, Senior Policy Analyst – Energy and Climate Change at the White House Domestic Policy Council; for their support and guidance. We would also like to acknowledge Mr. Antonio Bouza; HVAC&R Technology Manager at the US Department of Energy Building Technologies Office for his continued support.

APPENDIX A. RELEVANT LOW-GWP AREP TEST RESULTS TO DATE

APPENDIX A. RELEVANT LOW-GWP AREP TEST RESULTS TO DATE

Low-GWP AREP Phase I, though not primarily focused on high-ambient temperature testing, included a series of tests at high-ambient temperature conditions. High-ambient temperature testing is more prominent in Low-GWP AREP Phase II, and one report including high-ambient temperature testing has already been published. The high-ambient temperature tests performed within AREP so far included residential split-system ducted air conditioners and heat pumps (as well as other unrelated equipment). The results are informative, though not necessarily conclusive for the mini-split equipment under investigation in this study. A multi-split heat pump was also tested within Low-GWP AREP, with the results shown below; however, the tests were not performed at high ambient temperatures and cannot be said to represent performance in those conditions.

In this section, the testing temperature conditions are noted in the following form: outdoor temperature and indoor/room-side temperature. For example, 46.1°C and 28.9°C (115°F and 84°F) means that the outdoor temperature was maintained at 46.1°C while the indoor temperature was maintained at 28.9°C. All temperature values were rounded to the nearest integer in Fahrenheit due to the fact that AREP tests are usually based on AHRI and ASHRAE standards. Each plot presented below shows the cooling COP and the cooling capacity of the alternative refrigerants relative to the baseline refrigerant for each application. Accordingly, the results are normalized to the baseline data, to make comparisons easier to understand. The dashed lines, at 1 on each axis, represent the performance of the equipment using the baseline refrigerant. All results shown here, including the ones pertaining to heat pumps, were obtained in cooling mode.

In test report 42, results for a drop-in replacement of R-32 into an R-410A split-system air conditioner are presented.[40] These results are shown in Figure A.1.

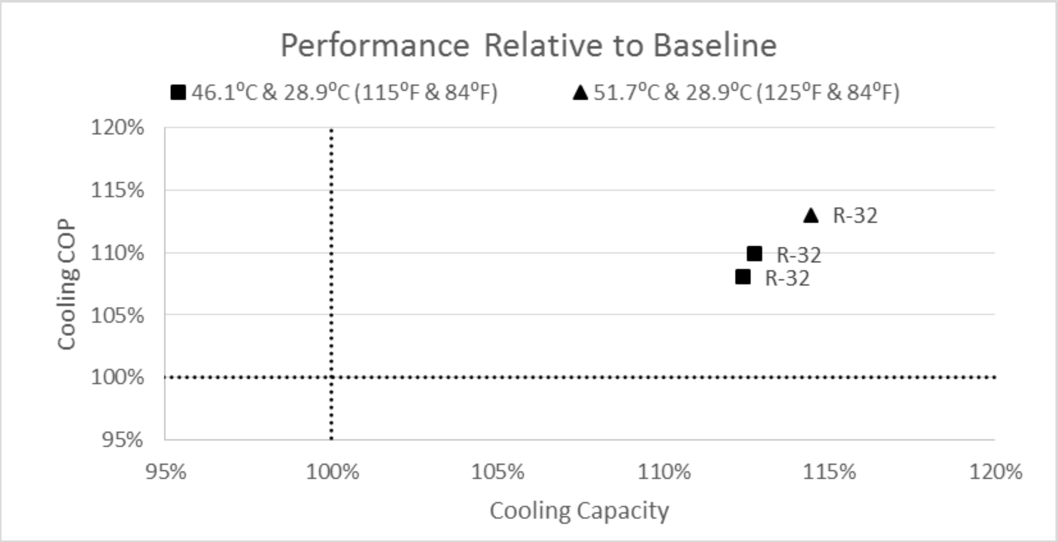


Figure A.1. Cooling COP as a function of capacity for R-32 compared to R-410A (drop-in replacement) in split-system air conditioners, 3-TR nominal capacity.

---

[40] Li, H., By, R. (2015) *System Soft-Optimization Tests of Refrigerant R-32 in a 3-ton Split System Air-Conditioner*, Goodman Manufacturing, AREP Test Report #42.

The results shown in Figure A.1 indicate that R-32 can achieve higher efficiency and capacity than R-410A for high-ambient-temperature split-system air-conditioning applications. The relative performance of R-32 was higher at the higher outdoor temperature tested (51.7°C) than at the lower outdoor temperature of 46.1°C.

In test reports 4, 20, and 32, results for R-410A split-system heat pumps are described.[41,42,43] All tests were performed on drop-in replacements with the exception of one, which was soft-optimized. These results are shown in Figure A.2.

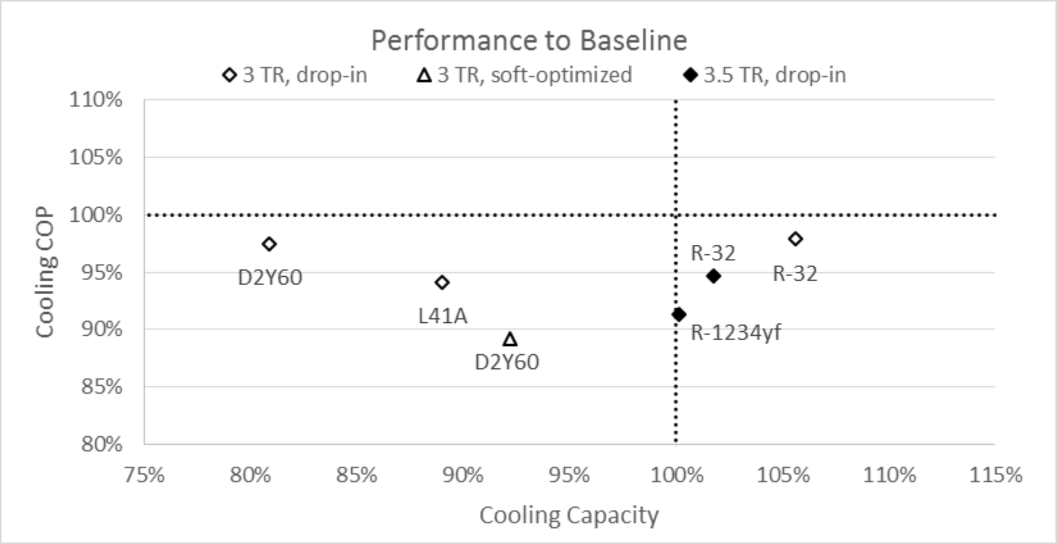


Figure A.2. Cooling COP as a function of capacity for alternative refrigerants compared to R-410A in split-system heat pumps tested at 46.1°C and 26.7°C (115°F and 80°F), 3 TR and 3.5 TR nominal capacities.

In these tests, R-1234yf matched the capacity of R-410A while R-32 surpassed it, but both refrigerants achieved lower COP values than the baseline. The other refrigerants tested (D2Y60 and L41A) performed worse than R-410A.

In test report 15, a variable refrigerant flow (VRF) multi-split heat pump was tested with a drop-in replacement of R-32 replacing R-410A.[44] Although the system was not tested at high ambient temperatures, the results are relevant to this report in that they pertain to a similar kind of unit (VRF multi-split). The unit was tested under the standard rating conditions for this kind of system in the United States and at 75%, 50%, and 25% cooling capacity. This was achieved by using the same indoor temperature (26.7°C) for all tests but with a different outdoor temperature for each test. The results are shown in Figure A.3.

---

[41] Crawford, T., Uselton, D. (2013) *System Drop-In Test of Refrigerant R-32 in Split System Heat Pump*, Lennox Industries Inc., AREP Test Report #4.

[42] Alabdulkarem, A., Hwang, Y., and Radermacher, R. (2013) *System Drop-In Tests of Refrigerants R-32, D2Y-60, and L41a in Air Source Heat Pump*, Center for Environmental Energy Engineering – University of Maryland, AREP Test Report #20.

[43] Alabdulkarem, A., Hwang, Y., and Radermacher, R. (2013) *System Soft-Optimized Test of Refrigerant D2Y60 in Air Source Heat Pump*, Center for Environmental Energy Engineering – University of Maryland, AREP Test Report #32.

[44] Tsujii, H., Imada, H. (2013) *System Drop-In Test of Refrigerant R-32 in a VRF Multi-split Heat Pump*, Daikin Industries Ltd., AREP Test Report #15.

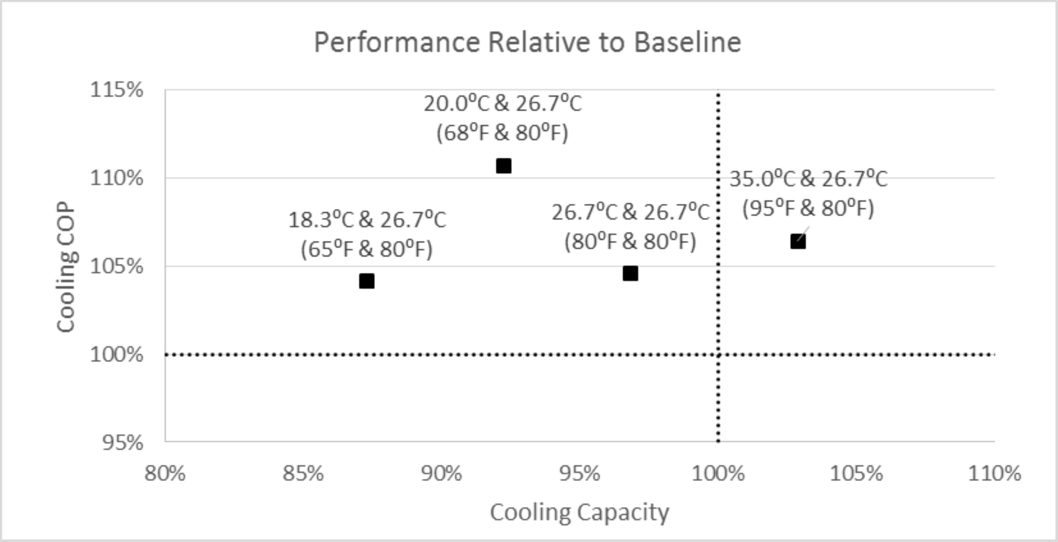


Figure A.3. COP as a function of capacity for R-32 compared to R-410A (drop-in replacement) in a multi-split heat pump, 8 TR nominal capacity.

The results indicate that the relative capacity of R-32 improves when compared to R-410A as the outdoor temperature is increased. Furthermore, the results show that the COP of R-32 is generally higher than that of R-410A in the range of conditions tested.

APPENDIX B. PRAHA RESULTS TO DATE

APPENDIX B. PRAHA RESULTS TO DATE

Results from PRAHA were not available at the time of writing of this interim report. Results may be included in the final report, if available.

APPENDIX C. EGYPRA RESULTS TO DATE

APPENDIX C. EGYPRA RESULTS TO DATE

Results from EGYPRA were not available at the time of writing of this interim report. Results may be included in the final report, if available.

APPENDIX D. EXPERIMENTAL TEST SETUP

APPENDIX D. EXPERIMENTAL TEST SETUP

Figures D.1 and D.2 provide an overview of the ORNL experimental test setup, with the measurement locations indicated. The as-installed system is shown in Figures D.3 and D.4. Finally, the fully instrumented outdoor unit is shown as installed in Figure D.5, and the details of the capillary tube installation are shown in Figure D.6.

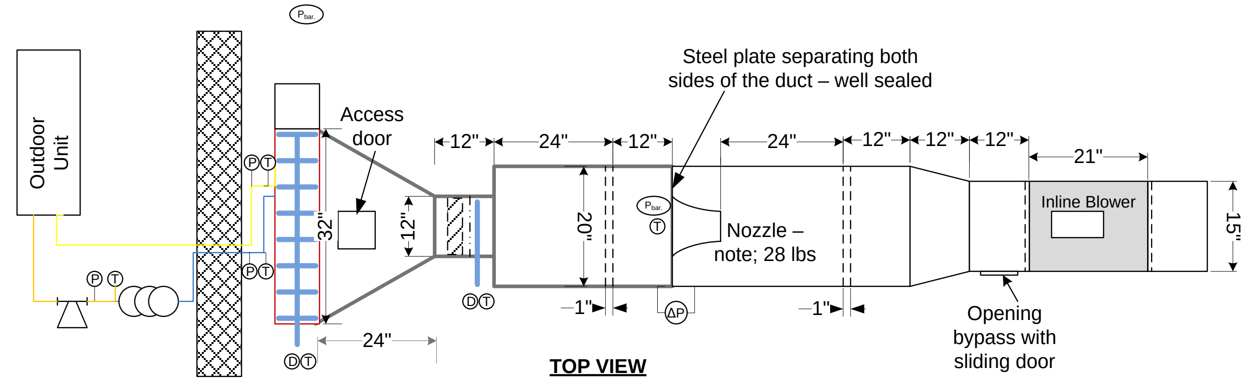


Figure D.1. Top view of the R-22 baseline unit experimental setup showing both the indoor side and the outdoor side along with instrumentation locations and design of the air enthalpy tunnel. For line legend, please refer to Figure D.2.

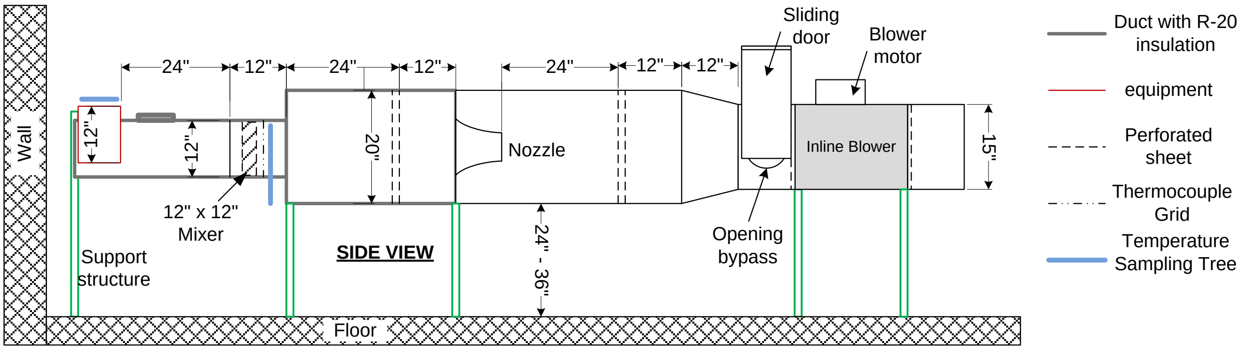


Figure D.2. Side view of the air enthalpy tunnel showing additional details and legend for the lines.

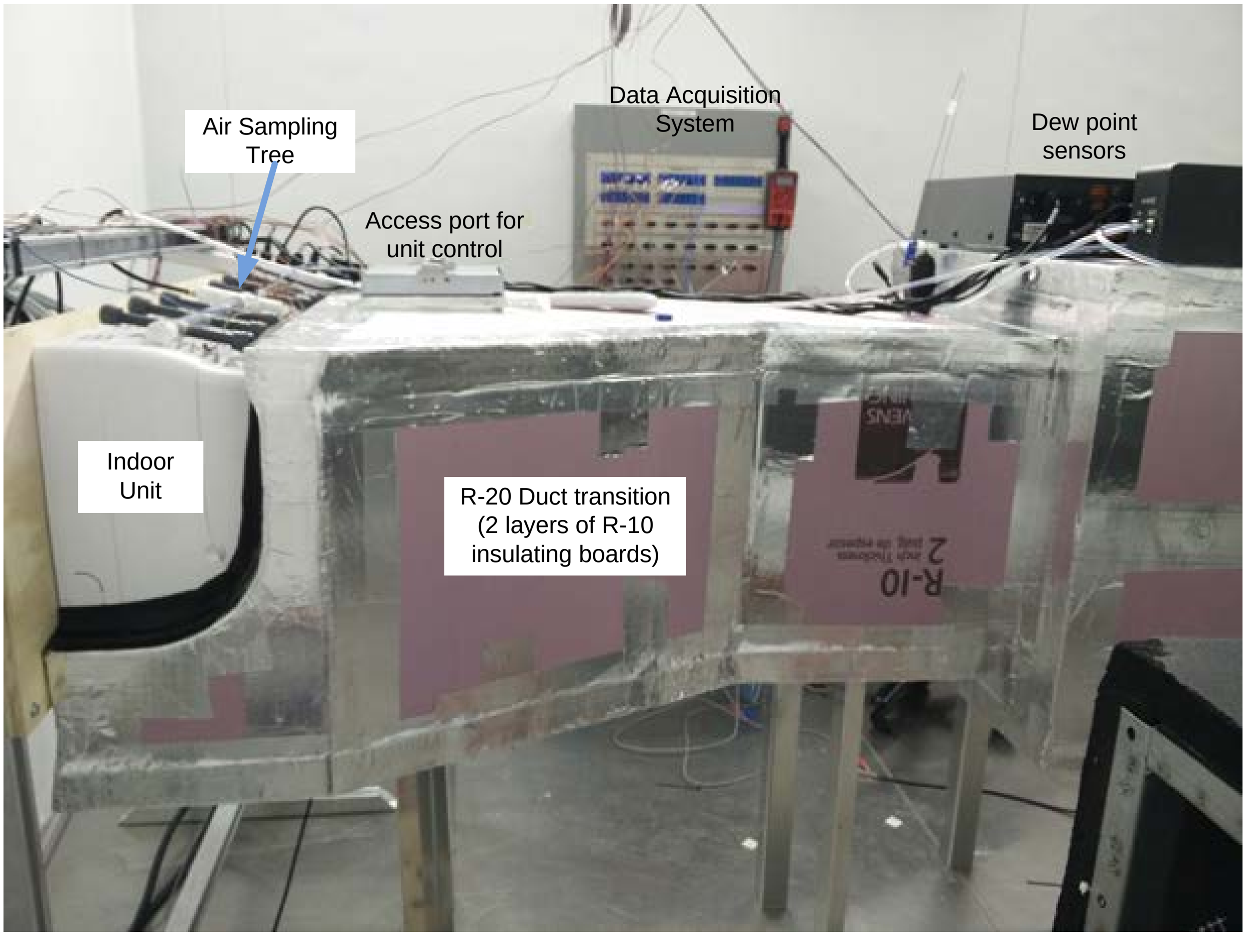


Figure D.3. As installed indoor unit, showing the sampling tree on the return air and the two-layer insulation (R-20 effective insulation).

Figure D.4. Indoor air enthalpy tunnel fully instrumented and connected to the data acquisition system.

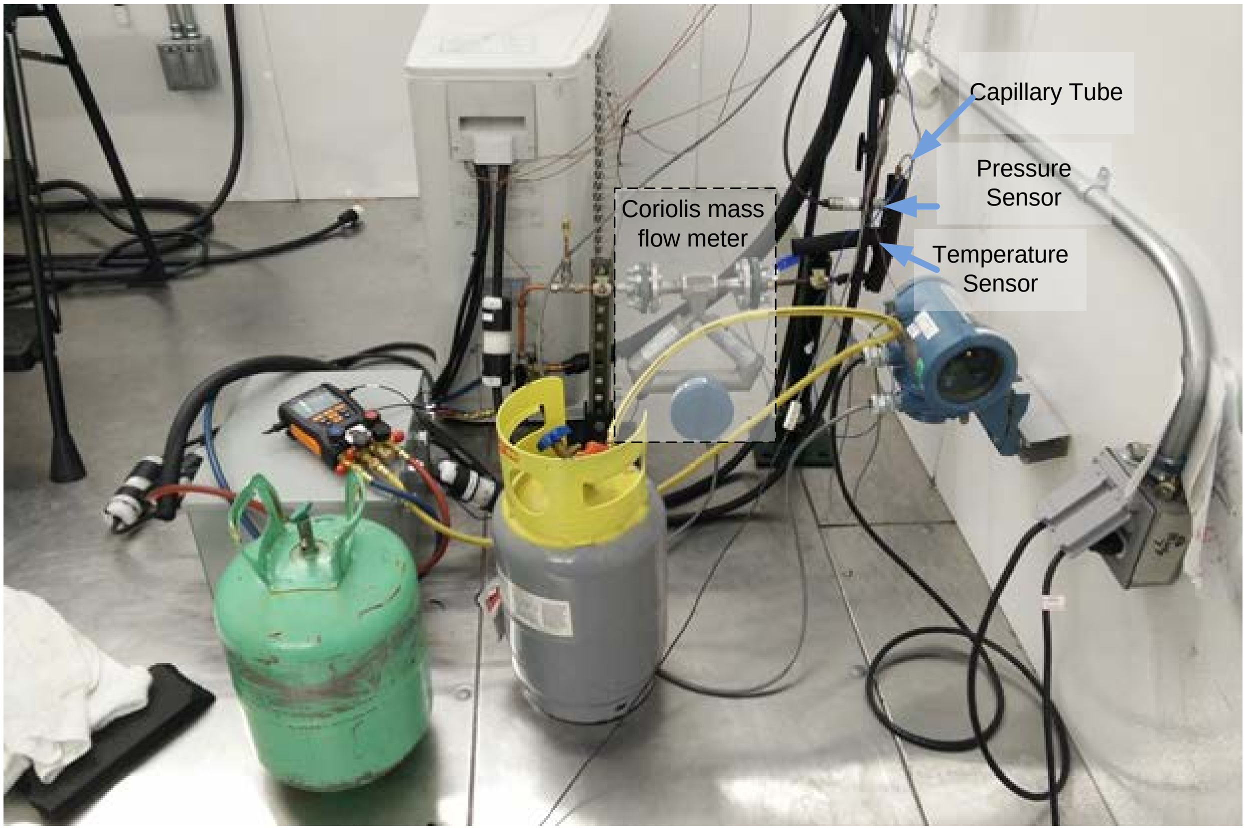


Figure D.5. Outdoor unit showing Coriolis mass flow meter, pressure and temperature sensors, and capillary tube.

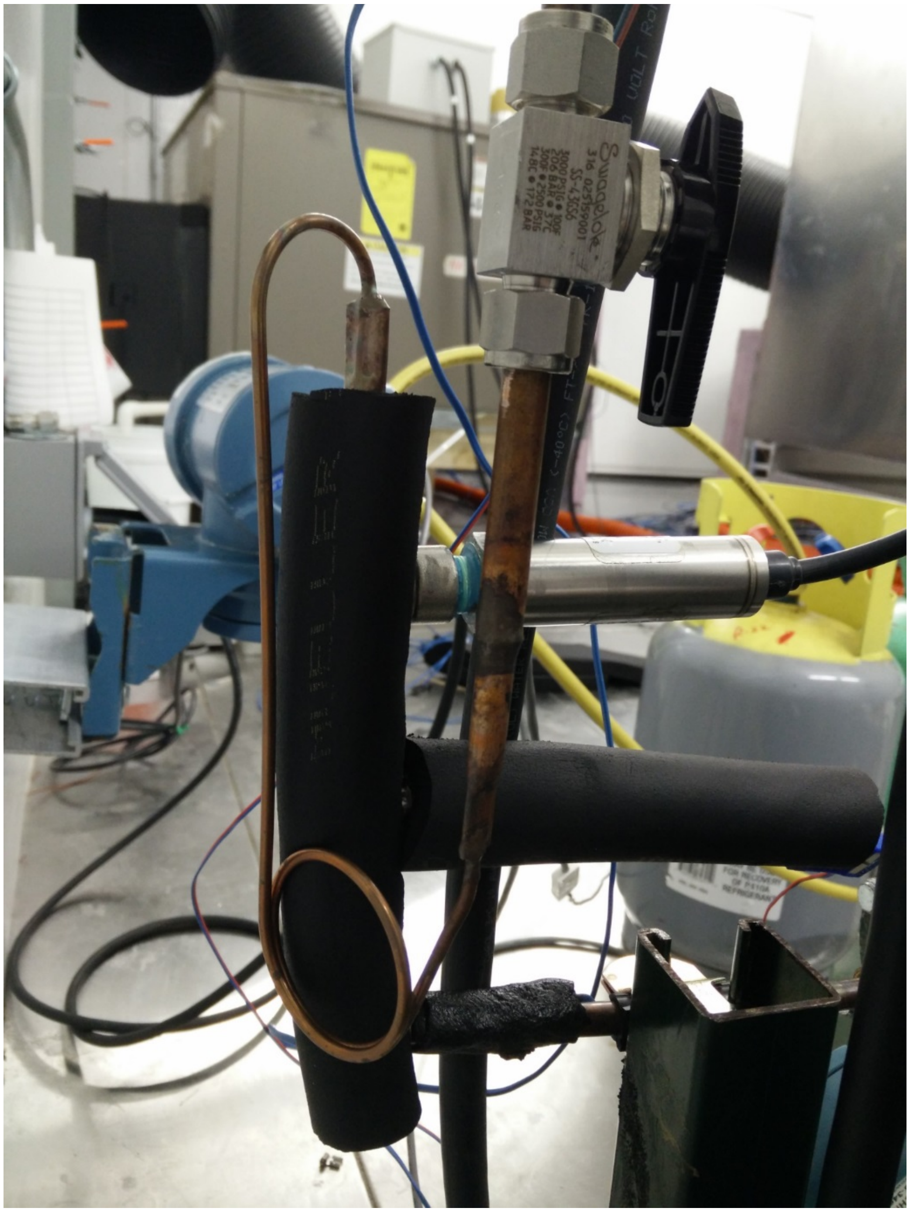


Figure D.6. Capillary tube downstream of the pressure and temperature sensors.

Table D.1 provides a summary of the instrumentation used for this testing.

Table D.1. R-22 unit instrumentation

Data	Instrument	Range and Accuracy	Comments
Indoor unit air inlet temperature	T-type thermocouple	1.7 to 79.4°C, 0.28°C (35 to 175°F, ±0.5°F)	Using aspirated sampling temperature tree
Indoor unit air inlet dew point	DewMaster EdgeTech Chilled mirror Hygrometer	-40 to 95°C, ±0.2°C (-40 to 203°F, ±0.36°F)	
Indoor unit air outlet temperature	T-type thermocouple	1.7 to 79.4°C, ±0.28°C (35 to 175°F, ±0.5°F)	Using aspirated sampling temperature tree
Indoor unit air outlet dew point	DewMaster EdgeTech Chilled mirror Hygrometer	-40 to 95°C, ±0.2°C (-40 to 203°F, ±0.36°F)	
Airflow rate	127 mm (5 in). nozzle		Air mass flow rate measurement
Barometric pressure upstream of the nozzle	Setra model 278 barometric pressure sensor	800 to 1100 hPa/mb, ±0.6 hPa/mb	
Temperature upstream of the nozzle	T-type thermocouple	1.7 to 79.4°C, 0.28°C (35 to 175°F, ±0.5°F)	
Pressure drop across the nozzle	Setra Model 239 differential pressure sensor	0 to 5 in. H_2O, ±0.073% FS	
Liquid line pressure	Omega Pressure Transducer PX409 absolute pressure sensor	0-750 psiA, ±0.08% BSL	Used to evaluate evaporator inlet enthalpy
Liquid line temperature	In-stream T-type thermocouple	1.7 to 79.4°C, 0.28°C (35 to 175°F, ±0.5°F)	
Evaporator inlet pressure	Omega Pressure Transducer PX409 absolute pressure sensor	0 to 250 psiA, ±0.08% BSL	Used to evaluate evaporator inlet enthalpy for refrigerant mixtures with significant glide
Evaporator inlet temperature	In-stream T-type thermocouple	1.7 to 79.4°C, 0.28°C (35 to 175°F, ±0.5°F)	
Evaporator inlet pressure	Omega Pressure Transducer PX409 absolute pressure sensor	0 to 250 psiA, ±0.08% BSL	Used to evaluate evaporator outlet enthalpy
Evaporator inlet temperature	In-stream T-type thermocouple	1.7 to 79.4°C, 0.28°C (35 to 175°F, ±0.5°F)	
Refrigerant mass flow rate	Micro Motion Elite CMF025 Coriolis mass flow meter	0 to 0.19 kg/s (0 to 25 lb/min), ±0.1% of rate	
Compressor power	Power meter	0–4 kW, ±0.2% reading	
Outdoor unit fan power	Power meter	0–4 kW, ±0.2% reading	
Indoor unit power	Power meter	0–1 kW, ±0.2% reading	

APPENDIX E. DETAILED R-22 TEST DATA

APPENDIX E. DETAILED R-22 TEST DATA

This appendix provides additional details of the testing documented in Section 7.1.

Figure E.1 shows the cooling COP for each refrigerant at each test condition, including R-22 using POE as a lubricant. Note that ORNL conducted two sets of tests with R-22 and POE oil: one set following the baseline tests in which only AHRI A, ISO T3, and Extreme ambient conditions were tested, as well as a set of “rerun” tests conducted at all ambient conditions. By comparing the results with POE before and after testing the alternative refrigerants, it is possible to establish the performance reliability using the alternative refrigerants.

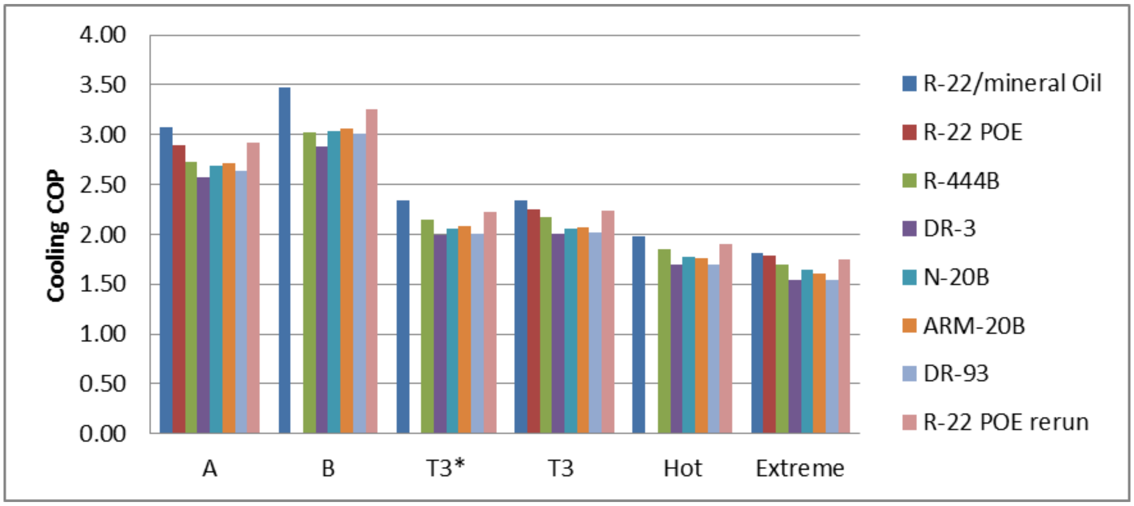


Figure E.1. Cooling COP for each refrigerant at each test condition, including R-22 using POE as a lubricant.

Figure E.2 shows the cooling capacity for each refrigerant at each test condition, including R-22 using POE as a lubricant.

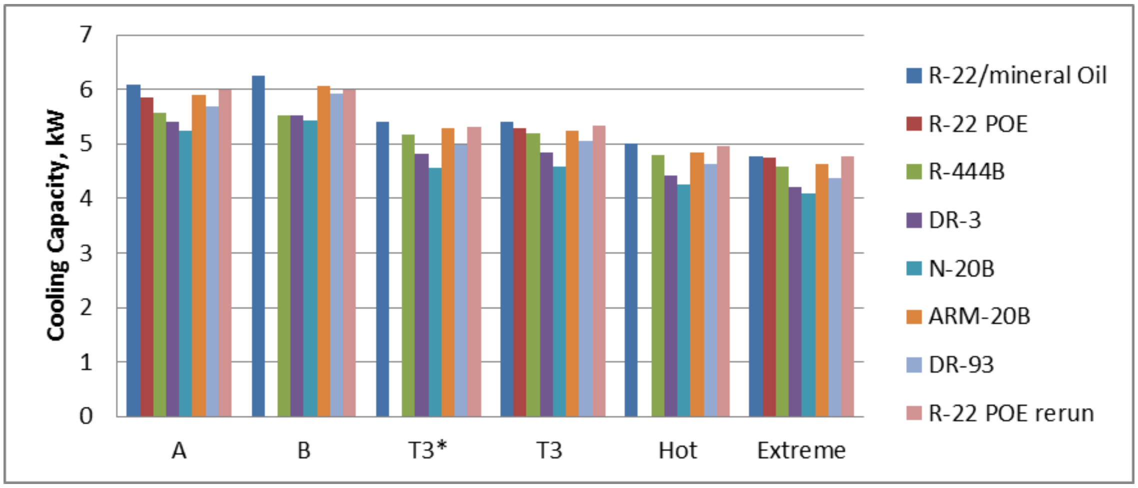


Figure E.2. Cooling capacity for each refrigerant at each test condition, including R-22 using POE as a lubricant.

Figure E.3 shows the cooling COP of each refrigerant at each test condition, relative to the cooling COP of that refrigerant at AHRI A conditions. Both runs with R-22 using POE are included.

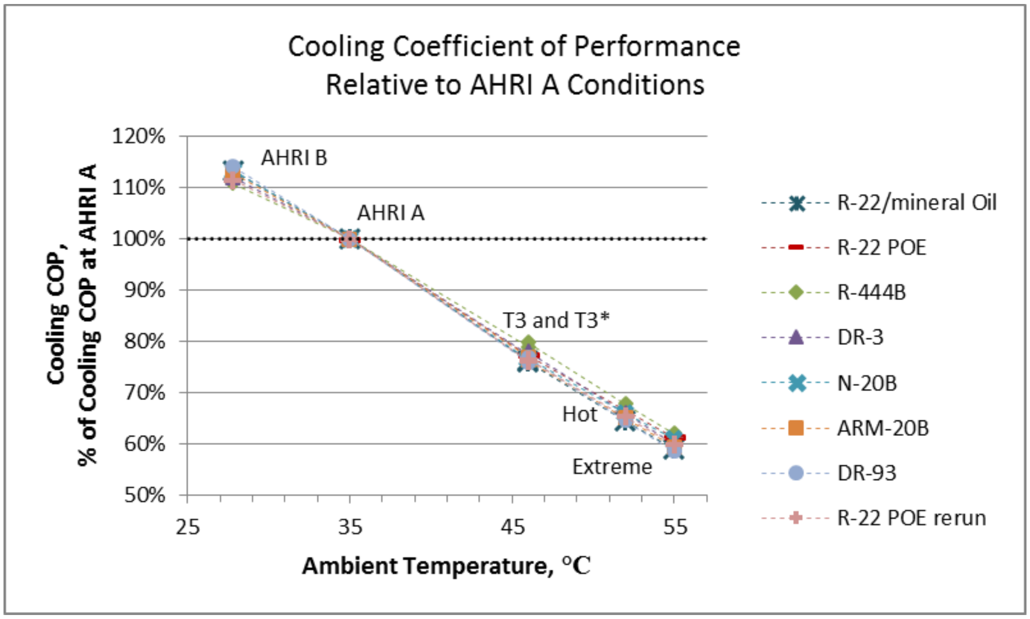


Figure E.3. Cooling COP of each refrigerant at each test condition, relative to the cooling COP of that refrigerant at AHRI A conditions.

Figure E.4 shows the cooling capacity of each refrigerant at each test condition, relative to the cooling capacity of that refrigerant at AHRI A conditions. Both runs with R-22 using POE are included.

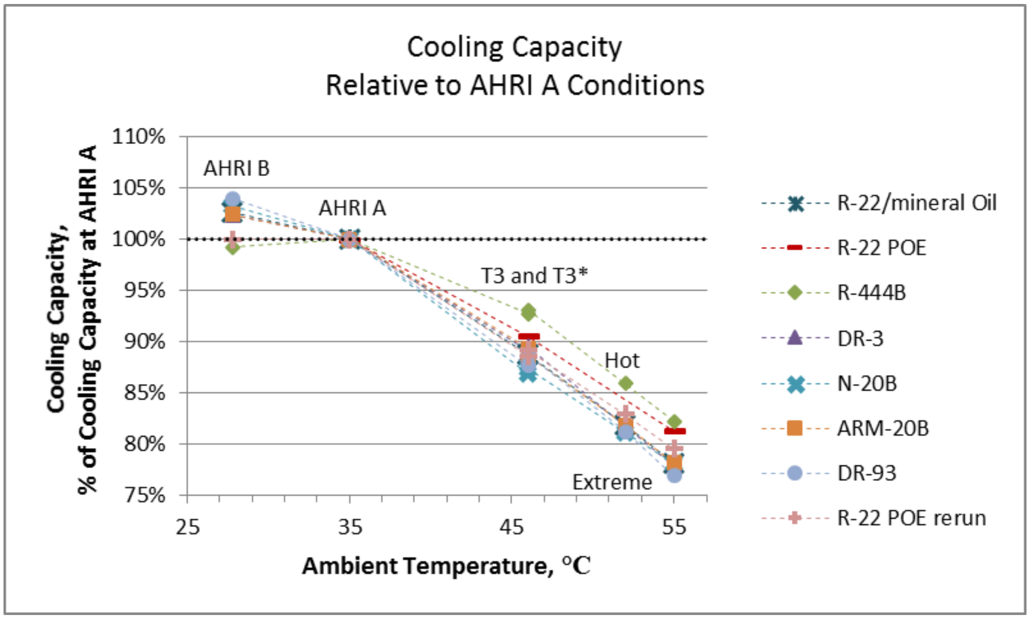


Figure E.4. Cooling capacity of each refrigerant at each test condition, relative to the cooling capacity of that refrigerant at AHRI A conditions.

Figure E.5 shows the condenser subcooling for each refrigerant under each of the test conditions.

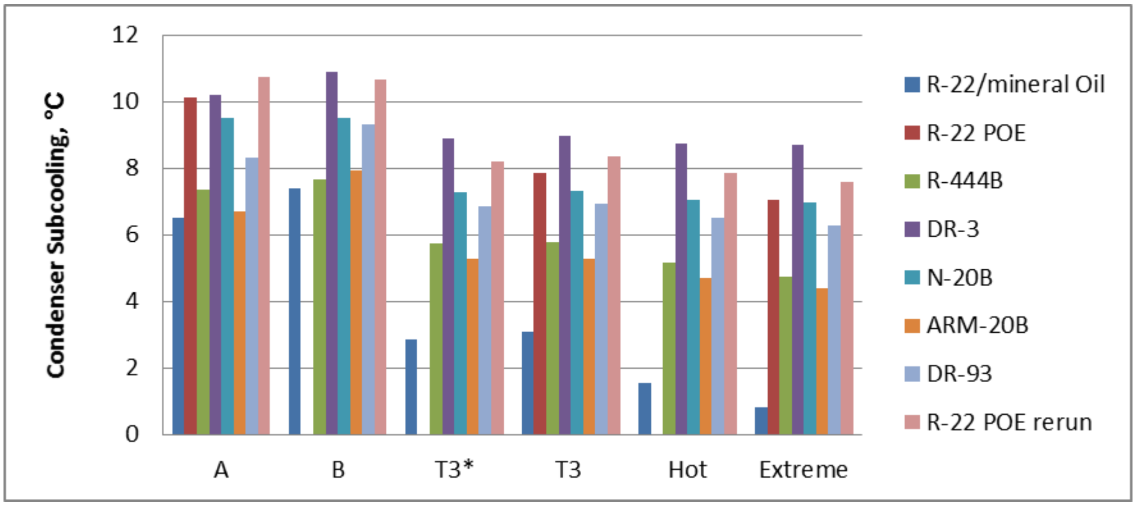


Figure E.5. Condenser subcooling for each refrigerant at each test condition.

Figure E.6 shows the evaporator superheat for each refrigerant under each of the test conditions.

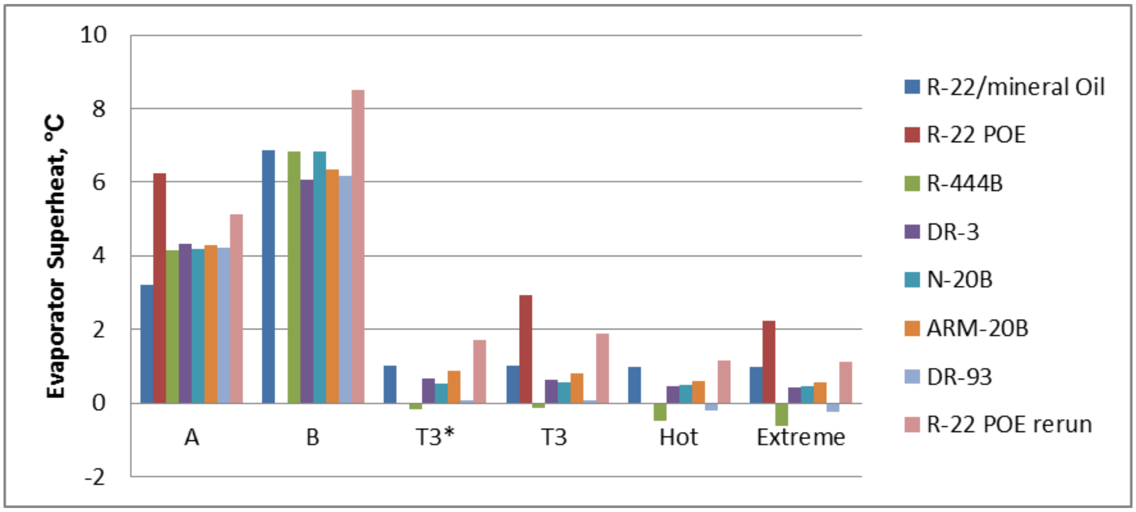


Figure E.6. Evaporator superheat for each refrigerant at each test condition.

Figure E.7 shows the compressor discharge temperature for each refrigerant under each of the test conditions.

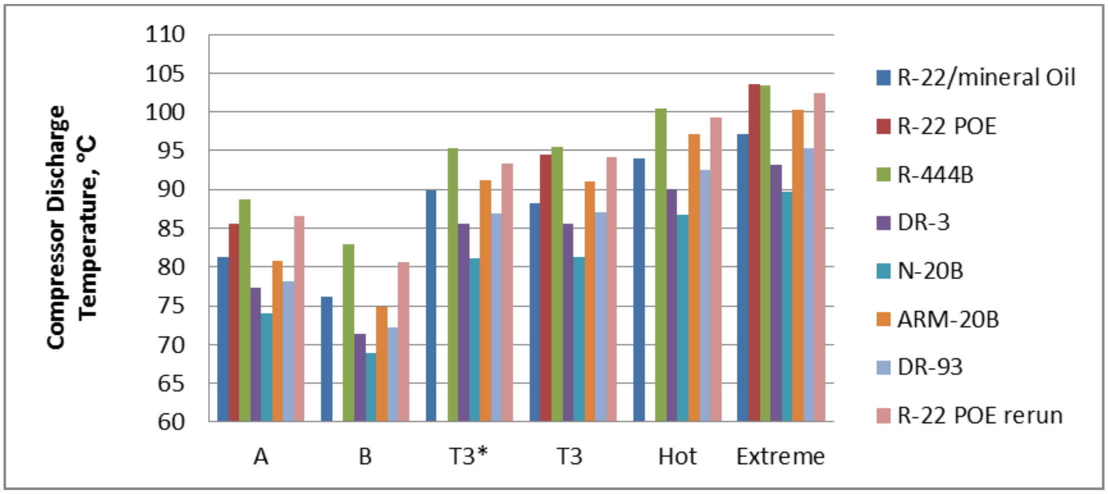


Figure E.7. Compressor discharge temperature for each refrigerant at each test condition.

Table E.1 shows additional test data, including

- air-side cooling capacity,
- air-side cooling COP,
- liquid line temperature and pressure,
- evaporator inlet temperature and pressure,
- evaporator outlet temperature and pressure, and
- compressor suction temperature.

Table E.1. Additional test data for the R-22 unit

	Refrigerant	Test Conditions					
		A	B	T3*	T3	Hot	Extreme
Cooling Capacity (air-side), kW	R-22 (mineral oil)	6.10	6.26	5.41	5.42	5.00	4.76
	R-22 (POE)	5.85	-	-	5.29	-	4.75
	R-444B	5.58	5.53	5.17	5.19	4.79	4.59
	DR-3	5.40	5.52	4.81	4.83	4.41	4.21
	N-20B	5.25	5.42	4.56	4.59	4.26	4.10
	ARM-20B	5.91	6.05	5.28	5.24	4.84	4.62
	DR-93	5.70	5.92	4.99	5.05	4.63	4.38
	R-22 (POE) rerun	5.98	5.99	5.31	5.34	4.96	4.76
Cooling COP (air-side)	R-22 (mineral oil)	3.07	3.48	2.34	2.34	1.98	1.82
	R-22 (POE)	2.90	-	-	2.24	-	1.78
	R-444B	2.72	3.02	2.15	2.17	1.85	1.69
	DR-3	2.57	2.88	1.99	2.01	1.70	1.55
	N-20B	2.68	3.04	2.05	2.06	1.77	1.64
	ARM-20B	2.71	3.06	2.09	2.07	1.76	1.61
	DR-93	2.63	3.00	2.00	2.02	1.70	1.54
	R-22 (POE) rerun	2.92	3.25	2.22	2.23	1.91	1.75
Liquid Line Temperature, °C	R-22 (mineral oil)	38.6	30.6	51.2	51.0	57.9	61.4
	R-22 (POE)	36.7	-	-	47.9	-	57.1
	R-444B	36.7	29.2	48.8	48.4	54.9	58.3
	DR-3	36.3	28.9	47.5	47.3	53.2	56.5
	N-20B	36.3	29.0	47.5	47.5	53.5	56.5
	ARM-20B	37.1	29.6	48.6	48.6	54.8	58.0
	DR-93	36.9	29.4	48.2	48.1	54.3	57.4
	R-22 (POE) rerun	36.8	29.4	48.0	48.0	54.2	57.3
Liquid Line Pressure, MPa (absolute)	R-22 (mineral oil)	1.73	1.46	2.13	2.13	2.40	2.55
	R-22 (POE)	1.80	-	-	2.21	-	2.65
	R-444B	1.92	1.62	2.44	2.42	2.75	2.93
	DR-3	1.96	1.68	2.43	2.42	2.72	2.90
	N-20B	1.76	1.48	2.15	2.15	2.44	2.60
	ARM-20B	2.03	1.75	2.55	2.55	2.88	3.06
	DR-93	1.99	1.71	2.49	2.49	2.82	2.99
	R-22 (POE) rerun	1.84	1.54	2.24	2.24	2.54	2.69

Table E.1. (continued)

	Refrigerant	Test Conditions					
		A	B	T3*	T3	Hot	Extreme
Evaporator Inlet Temperature, °C	R-22 (mineral oil)	12.1	10.1	14.3	14.5	15.9	16.5
	R-22 (POE)	11.3	-	-	13.8	-	15.6
	R-444B	5.1	1.8	8.9	8.7	10.4	11.3
	DR-3	8.8	6.2	12.0	12.1	13.9	15.1
	N-20B	10.6	8.0	13.5	13.6	15.5	16.5
	ARM-20B	7.8	5.4	10.5	10.4	12.1	12.9
	DR-93	8.8	6.5	11.7	11.8	13.6	14.6
	R-22 (POE) rerun	11.5	9.0	13.7	14.0	15.2	15.9
Evaporator Inlet Pressure, MPa (absolute)	R-22 (mineral oil)	0.726	0.686	0.773	0.778	0.808	0.823
	R-22 (POE)	0.709	-	-	0.762	-	0.762
	R-444B	0.666	0.605	0.738	0.735	0.766	0.783
	DR-3	0.720	0.675	0.775	0.777	0.809	0.829
	N-20B	0.672	0.625	0.722	0.724	0.759	0.778
	ARM-20B	0.769	0.720	0.821	0.820	0.853	0.870
	DR-93	0.751	0.704	0.806	0.809	0.845	0.867
	R-22 (POE) rerun	0.714	0.661	0.761	0.767	0.795	0.810
Evaporator Outlet Temperature, °C	R-22 (mineral oil)	9.8	12.0	8.8	9.0	9.8	10.2
	R-22 (POE)	12.4	-	-	10.8	-	11.3
	R-444B	13.0	13.0	11.3	11.2	11.9	12.2
	DR-3	12.8	12.7	10.8	10.9	11.6	12.1
	N-20B	12.5	13.2	10.1	10.3	11.2	11.9
	ARM-20B	11.9	12.1	10.2	10.0	10.8	11.1
	DR-93	11.9	12.2	9.3	9.4	10.2	10.8
	R-22 (POE) rerun	11.7	12.8	9.6	10.0	10.1	10.6
Evaporator Outlet Pressure, MPa (absolute)	R-22 (mineral oil)	0.614	0.586	0.637	0.641	0.657	0.665
	R-22 (POE)	0.605	-	-	0.638	-	0.661
	R-444B	0.576	0.525	0.626	0.624	0.643	0.654
	DR-3	0.591	0.559	0.623	0.625	0.641	0.653
	N-20B	0.548	0.515	0.571	0.573	0.592	0.604
	ARM-20B	0.653	0.615	0.688	0.686	0.706	0.715
	DR-93	0.617	0.585	0.649	0.651	0.672	0.686
	R-22 (POE) rerun	0.614	0.571	0.638	0.644	0.660	0.670

Table E.1. (continued)

	Refrigerant	Test Conditions					
		A	B	T3*	T3	Hot	Extreme
Compressor Suction Temperature, °C	R-22 (mineral oil)	12.6	13.8	9.5	9.5	9.6	9.8
	R-22 (POE)	14.4	-	-	12.2	-	11.8
	R-444B	16.1	15.5	12.1	12.0	11.8	12.2
	DR-3	14.9	14.3	13.0	13.1	11.8	11.8
	N-20B	14.7	14.8	11.9	12.2	12.0	12.4
	ARM-20B	14.6	14.1	13.2	12.9	12.5	12.2
	DR-93	14.1	13.9	12.4	12.6	11.3	11.0
	R-22 (POE) rerun	14.7	15.1	10.6	11.1	10.0	10.4
Saturation Temperature, Liquid Line, °C	R-22 (mineral oil)	45.1	38.0	54.0	54.1	59.5	62.2
	R-22 (POE)	46.8	-17.8	-17.8	55.8	-17.8	64.1
	R-444B	44.1	36.9	54.5	54.2	60.0	63.0
	DR-3	46.5	39.8	56.4	56.2	61.9	65.2
	N-20B	45.8	38.5	54.8	54.8	60.6	63.5
	ARM-20B	43.8	37.5	53.9	53.9	59.5	62.4
	DR-93	45.2	38.8	55.1	55.0	60.8	63.7
	R-22 (POE) rerun	47.5	40.1	56.2	56.4	62.1	64.9
Saturation Temperature, Evaporator Inlet, °C	R-22 (mineral oil)	12.2	10.2	14.3	14.5	15.8	16.4
	R-22 (POE)	11.3	-17.8	-17.8	13.8	-17.8	15.6
	R-444B	13.4	10.4	16.7	16.6	17.9	18.6
	DR-3	14.9	12.8	17.4	17.5	18.9	19.7
	N-20B	14.8	12.5	17.2	17.3	18.9	19.7
	ARM-20B	12.9	10.8	15.1	15.1	16.4	17.0
	DR-93	13.9	11.9	16.3	16.4	17.9	18.7
	R-22 (POE) rerun	11.6	9.0	13.8	14.0	15.2	15.9
Saturation Temperature, Evaporator Outlet, °C	R-22 (mineral oil)	6.6	5.1	7.8	8.0	8.8	9.2
	R-22 (POE)	6.1	-17.8	-17.8	7.8	-17.8	9.0
	R-444B	8.9	6.1	11.5	11.4	12.3	12.8
	DR-3	8.5	6.7	10.1	10.3	11.1	11.7
	N-20B	8.3	6.4	9.6	9.7	10.8	11.4
	ARM-20B	7.6	5.8	9.3	9.2	10.2	10.6
	DR-93	7.7	6.1	9.3	9.4	10.4	11.0
	R-22 (POE) rerun	6.6	4.3	7.9	8.1	9.0	9.4

Table E.1. (continued)

	Refrigerant	**Test Conditions**					
		A	B	T3*	T3	Hot	Extreme
Saturation Temperature, Compressor Discharge, °C	R-22 (mineral oil)	45.1	38.0	54.0	54.1	59.5	62.2
	R-22 (POE)	46.8	-17.8	-17.8	55.8	-17.8	64.1
	R-444B	51.4	44.6	61.2	60.9	66.3	69.0
	DR-3	52.6	46.3	61.8	61.6	66.8	69.7
	N-20B	50.2	43.2	58.8	58.8	64.3	67.1
	ARM-20B	48.9	42.9	58.6	58.5	63.9	66.5
	DR-93	49.8	43.7	59.2	59.2	64.6	67.3
	R-22 (POE) rerun	47.5	40.1	56.2	56.4	62.1	64.9
Mass Flow Rate, kg/min	R-22 (mineral oil)	2.32	2.20	2.42	2.44	2.47	2.48
	R-22 (POE)	2.21	-	-	2.32	-	2.37
	R-444B	1.70	1.57	1.86	1.85	1.90	1.92
	DR-3	2.28	2.17	2.40	2.41	2.47	2.50
	N-20B	2.26	2.13	2.36	2.36	2.43	2.46
	ARM-20B	2.11	2.00	2.20	2.20	2.25	2.26
	DR-93	2.33	2.22	2.44	2.44	2.50	2.53
	R-22 (POE) rerun	2.24	2.09	2.34	2.35	2.39	2.41
Capillary Tube Length, mm	R-22 (mineral oil)	Original without header					
	R-22 (POE)	254					
	R-444B	356					
	DR-3	178					
	N-20B	152					
	ARM-20B	178					
	DR-93	152					
	R-22 (POE) rerun	254					

APPENDIX F. DETAILED R-410A TEST DATA

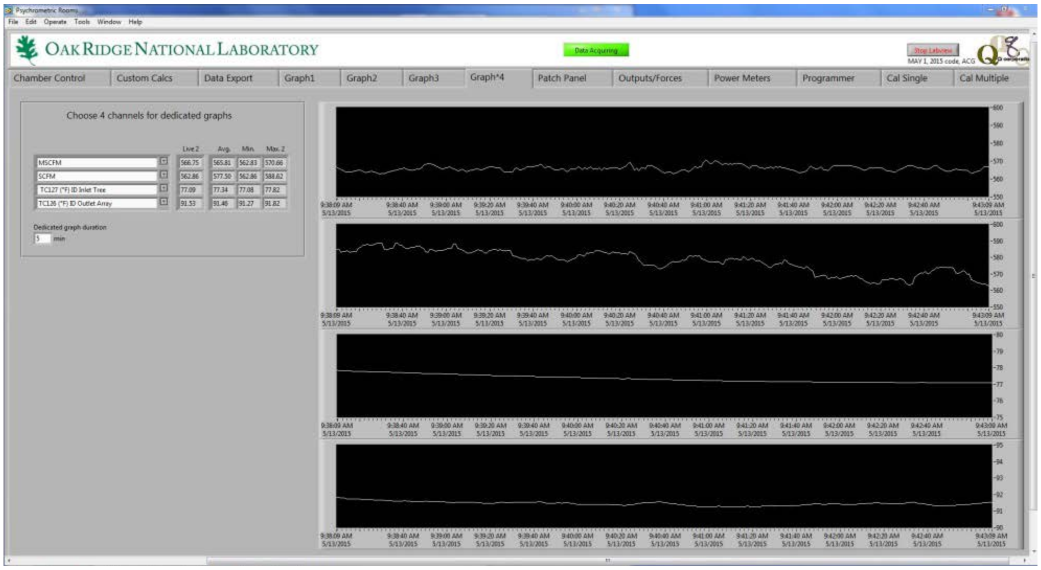
APPENDIX F. DETAILED R-410A TEST DATA

For inclusion in the final report.

APPENDIX G. DATA REDUCTION METHODOLOGY

APPENDIX G. DATA REDUCTION METHODOLOGY

The measured data obtained using the instrumentation listed in Appendix D were recorded using a National Instrument Data Acquisition system. Data were recorded continually at 5 second intervals. LabView® code was developed to allow for real-time data visualization and performance monitoring, as shown in Figures G.1 and G.2. REFPROP property calculations were included in the code, as shown in Figure G.3, in order to facilitate the real-time evaluation of the refrigerant side properties, e.g., saturation properties and capacity. The airflow, air-side capacity, refrigerant-side capacity, and refrigerant subcooling and superheat calculations are presented in the following sections. Table G.1 shows the symbols, and Table G.2 shows the subscripts used in the calculations.

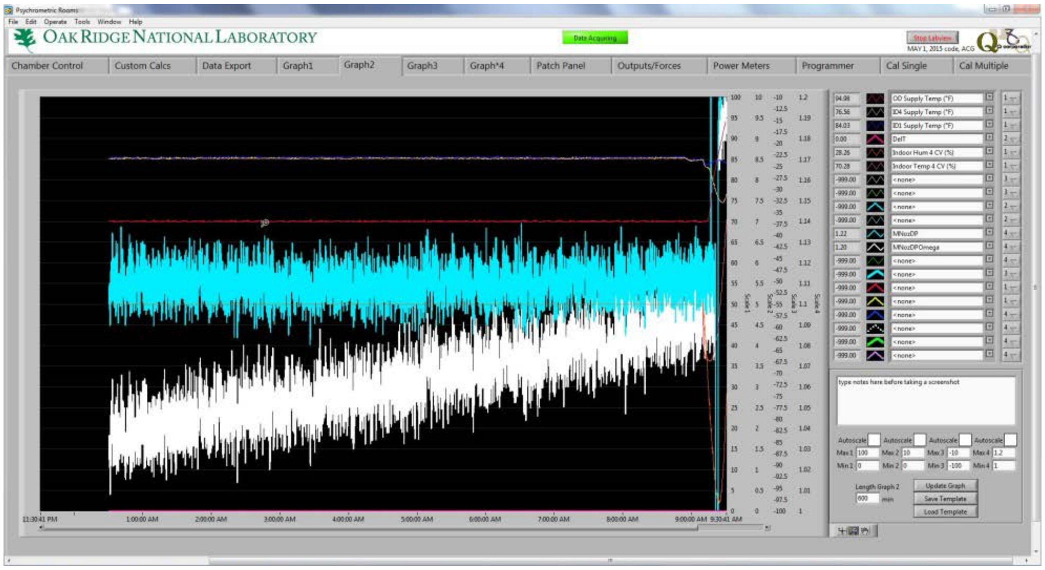


Choose 4 channels for dedicated graphs

	Live 2	Avg.	Min.	Max. 2
MSCFM	566.75	565.81	562.83	570.66
SCFM	562.86	577.50	562.86	588.62
TC127 (°F) ID Inlet Tree	77.09	77.34	77.08	77.82
TC126 (°F) ID Outlet Array	91.53	91.46	91.27	91.82

Dedicated graph duration 5 min

Figure G.1. LabView® display of room temperature and fan flow rate.



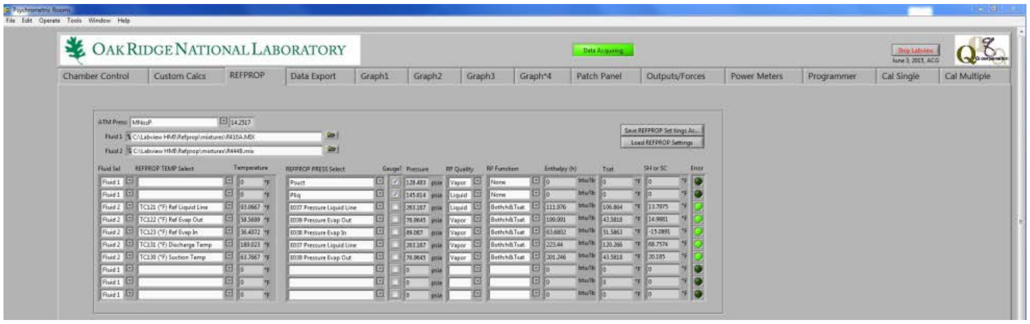
94.98	OD Supply Temp (°F)	1
76.56	ID4 Supply Temp (°F)	1
84.03	ID1 Supply Temp (°F)	1
0.00	DelT	2
28.26	Indoor Hum 4 CV (%)	1
70.28	Indoor Temp 4 CV (%)	1
-999.00	<none>	3
-999.00	<none>	3
-999.00	<none>	2
-999.00	<none>	2
1.22	MNozDP	4
1.20	MNozDPOmega	4
-999.00	<none>	4
-999.00	<none>	3
-999.00	<none>	1
-999.00	<none>	1
-999.00	<none>	4
-999.00	<none>	4
-999.00	<none>	4
-999.00	<none>	4

type notes here before taking a screenshot

Autoscale ☐ Autoscale ☐ Autoscale ☐ Autoscale ☐
Max 1 100 Max 2 10 Max 3 -10 Max 4 1.2
Min 1 0 Min 2 0 Min 3 -100 Min 4 1

Length Graph 2 600 min

Figure G.2. LabView® display of various monitored parameters.



ATM Press: MNosP 14.2517

Fluid 1: C:\Labview HMB\Refprop\mixtures\R410A.MIX

Fluid 2: C:\Labview HMB\Refprop\mixtures\R444B.mix

Fluid Sel	REFPROP TEMP Select	Temperature		REFPROP PRESS Select	Gauge?	Pressure		RP Quality	RP Function	Enthalpy (h)		Tsat		SH or SC		Error
Fluid 1		0	°F	Psuct	☑	128.483	psia	Vapor	None	0	btu/lb	0	°F	0	°F	●
Fluid 1		0	°F	Pliq	☑	145.814	psia	Liquid	None	0	btu/lb	0	°F	0	°F	●
Fluid 2	TC121 (°F) Ref Liquid Line	93.0667	°F	E037 Pressure Liquid Line	☐	283.187	psia	Liquid	Bothh&Tsat	111.976	btu/lb	106.864	°F	13.7975	°F	●
Fluid 2	TC122 (°F) Ref Evap Out	58.5699	°F	E039 Pressure Evap Out	☐	78.9645	psia	Vapor	Bothh&Tsat	199.901	btu/lb	43.5818	°F	14.9881	°F	●
Fluid 2	TC123 (°F) Ref Evap In	36.4872	°F	E038 Pressure Evap In	☐	89.087	psia	Vapor	Bothh&Tsat	83.6802	btu/lb	51.5863	°F	-15.0891	°F	●
Fluid 2	TC131 (°F) Discharge Temp	189.023	°F	E037 Pressure Liquid Line	☐	283.187	psia	Vapor	Bothh&Tsat	223.44	btu/lb	120.266	°F	68.7574	°F	●
Fluid 2	TC130 (°F) Suction Temp	63.7667	°F	E039 Pressure Evap Out	☐	78.9645	psia	Vapor	Bothh&Tsat	201.246	btu/lb	43.5818	°F	20.185	°F	●
Fluid 1		0	°F		☐	0	psia			0	btu/lb	0	°F	0	°F	●
Fluid 1		0	°F		☐	0	psia			0	btu/lb	0	°F	0	°F	●
Fluid 1		0	°F		☐	0	psia			0	btu/lb	0	°F	0	°F	●

Figure G.3. LabView® display of built-in REFPROP calculation.

Table G.1. Data reduction methodology symbols

Symbol	Description	Unit
A_{nozzle}	Area of cross section at the nozzle throat	ft²
C_{nozzle}	Nozzle discharge coefficient	-
C_p	Specific heat	Btu/lbm·°F
D	Diameter	In
h	Enthalpy	Btu/lbm
L	Side	in
$\dot{m}$	Flow rate	lbm/h
P	Absolute pressure	inH_2O
q	Heat capacity	Btu/h
Q	Airflow rate	cfm
Q_s	Standard airflow rate	scfm
Re	Reynolds number	-
T	Dry bulb temperature	°F
Y	Expansion Factor	-
W	Electric Power	Watt
α	Ratio of the absolute pressure at exit from the nozzle to the absolute pressure entering the nozzle	-
β	Ratio of nozzle throat diameter to duct diameter	-
Δ	Differential	inH_2O, °F
μ	Moist air humidity ratio	lb H_20/lb air
ρ	Moist air density	lbm/ft^3

Table G.2. Data reduction methodology subscripts

Subscript	Description
air	Air side
$compressor$	Compressor
$condensate$	Condensate collected from the evaporator
$condenser\ fan$	Condenser fan
$duct$	Related to the duct
$evaporator\ fan$	Evaporator fan
$evap, in$	Evaporator inlet conditions
$evap, out$	Evaporator outlet conditions
$exit$	Exiting the nozzle
$latent$	Latent capacity
$nozzle$	Nozzle condition
ref	Refrigerant side
$coil$	At the coil
$return$	Return air to the indoor unit
sat	Saturated conditions at the equilibrium
$subcooling$	Degrees of subcooling
$superheat$	Degrees of superheat
$sensible$	Sensible capacity
$supply$	Supply air exiting the indoor unit
$total$	Total capacity
$tree$	From sampling tree
$upstream$	Entering the nozzle

Airflow Rate Calculation:

The airflow rate calculations were performed according to ASHRAE Standard 41.2-87 (RA92). The airflow rate is calculated as shown in equation (1). Equations (2) and (5) are used to calculate the expansion factor, Y, and the nozzle discharge coefficient, C_{nozzle}, respectively; both of them are inputs to Equation (1). The expansion factor is in turn a function of two parameters: α, the ratio of absolute pressures at the exit and the inlet of the nozzle, and β, the ratio of nozzle throat diameter to duct diameter. α and β are calculated as shown in Equations (3) and (4), respectively. The nozzle discharge coefficient, C_{nozzle}, is calculated based on *Re*, which can be approximately calculated using Equation (6) since the airflow velocity is not known. Finally, in order to evaluate the standard airflow rate, we use Equation (7) to normalize using the standard dry air density at 70°F and 14.696 psia of 0.075 lbm/ft^3.

$$Q = 1096 \times Y \times \sqrt{\Delta P_{nozzle}/\rho_{nozzle}} \times C_{nozzle} \times A_{nozzle} \qquad (1)$$

$$Y = 1 - (0.548 + 0.71 \times \beta^4)(1 - \alpha) \qquad (2)$$

$$\alpha = \frac{P_{nozzle, exit}}{P_{nozzle, upstream}} \qquad (3)$$

$$\beta = \frac{D_{nozzle}}{D_{duct}} = \left(\frac{D_{nozzle}}{\sqrt{\frac{4}{\pi} \times L_{duct}^2}}\right) = \left(\frac{5}{\sqrt{\frac{4}{\pi} \times 20^2}}\right) = 0.22155673 \tag{4}$$

$$C_{nozzle} = 0.9986 - \left(\frac{7.006}{\sqrt{Re}}\right) + \left(\frac{134.6}{Re}\right) \tag{5}$$

$$Re = 1{,}363{,}000 \times D_{nozzle} \times \sqrt{\frac{\Delta P \times \rho_{nozzle}}{1-\beta^4}} \tag{6}$$

$$Q_S = \frac{Q \times \rho_{nozzle}}{(1+\mu_{nozzle})} / 0.075 \tag{7}$$

In the above calculations, ρ_{nozzle} is calculated based on barometric pressure, temperature, and dew-point measurement upstream of the nozzle using equations from the *ASHRAE Handbook of Fundamentals*, 2009, Chapter 1 (Equations 23 and 28). The $P_{nozzle,\,upstream}$ is measured using a barometric pressure sensor and the $P_{nozzle,\,exit}$ is calculated using the $P_{nozzle,\,upstream}$ and the differential pressure drop across the nozzle, which is measured using a differential pressure sensor and can be calculated as shown in Equation (8) below.

$$P_{nozzle,\,exit} = P_{nozzle,\,upstream} - \Delta P_{nozzle} \tag{8}$$

Air-Side Capacity Calculations:

Air-side capacity is calculated according to ANSI/ASHRAE Standard 37-2009 Air Enthalpy Method. The total air-side capacity can be calculated as shown in Equation (9); the supply and return enthalpies used in Equation (9) are calculated using Equations (10) and (11). The factor of 60 is used to convert the airflow rate from cfm to ft^3/h.

$$q_{total} = Q_s \times 0.075 \times 60 \times \left(h_{return} - h_{supply}\right) \tag{9}$$

$$h_{return} = 0.24 \times T_{return,\,tree} + \mu_{return,\,tree} \times \left(1061 + 0.444 \times T_{return,\,tree}\right) \tag{10}$$

$$h_{supply} = 0.24 \times T_{supply,\,tree} + \mu_{supply,\,tree} \times \left(1061 + 0.444 \times T_{supply,\,tree}\right)$$

(11)

The air-side sensible and latent capacities can be calculated as shown in Equations (12) and (15). Also, when the dew-point temperature difference is low, it would be more accurate to use the condensate measurement for latent capacity measurement, as shown in Equation (16).

$$q_{sensible} = Q_s \times 0.075 \times 60 \times \left((CpT)_{return} - (CpT)_{supply}\right) \tag{12}$$

$$(CpT)_{return} = 0.24 \times T_{return,\,tree} + \mu_{return,\,tree} \times 0.444 \times T_{return,\,tree} \tag{13}$$

$$(CpT)_{supply} = 0.24 \times T_{supply,\,tree} + \mu_{supply,\,tree} \times 0.444 \times T_{supply,\,tree} \tag{14}$$

$$q_{latent} = Q_s \times 0.075 \times 60 \times 1061 \times \left(\mu_{return,\,tree} - \mu_{supply,\,tree}\right) \tag{15}$$

$$q_{latent,\,condensate} = \dot{m}_{condensate} \times 1061 \tag{16}$$

Refrigerant-Side Capacity Calculations:

The refrigerant-side capacity is calculated according to ANSI/ASHRAE Standard 37-2009 Refrigerant Enthalpy Method with refrigerant mass flow measurement. It can be calculated as shown in Equation (17) below.

$$q_{ref,\,coil} = \dot{m}_{ref} \times \left(h_{evap,\,out} - h_{evap,\,in}\right) \tag{17}$$

In this equation, the refrigerant flow rate, $\dot{m}_{ref}$, was measured using a Coriolis mass flow meter, and the refrigerant enthalpies were calculated using NIST REFPROP based on pressure and temperature measurements at the evaporator outlet and the condenser liquid line for $h_{evap,\,out}$ and $h_{evap,\,in}$, respectively. To compare both the air-side and the refrigerant-side capacities, the fan power dissipated into the airstream has to be considered: $W_{evaporator\ fan}$.

Efficiency Calculations:

The EER can be calculated based on the air-side or the refrigerant-side measurements by using Equations (18) or (19), respectively. The COP can be obtained from the EER through a unit conversion, as shown in Equations (20) and (21) for the air side and the refrigerant side, respectively.

$$EER_{air} = \frac{q_{total}}{W_{compressor} + W_{condenser\ fan} + W_{evaporator\ fan}} \tag{18}$$

$$EER_{ref} = \frac{q_{ref,\,coil} - W_{evaporator\ fan}}{W_{compressor} + W_{condenser\ fan} + W_{evaporator\ fan}} \tag{19}$$

$$COP_{air} = \frac{EER_{air}}{3.4121} \tag{20}$$

$$COP_{ref} = \frac{EER_{ref}}{3.4121} \tag{21}$$

Subcooling:

The liquid line subcooling was calculated based on the equation below, in which the saturation temperature, T_{sat}, was calculated using NIST REFPROP based on pressure measurements at the liquid line. The temperature, T , was directly measured using an in-stream thermocouple or RTD.

$$\Delta T_{subcooling} = T_{sat} - T \tag{22}$$

Superheat:

The evaporator outlet and compressor inlet superheat were calculated based on the equation below, in which the saturation temperature, T_{sat}, was calculated using NIST REFPROP based on the pressure at the evaporator outlet. The temperature, T, was directly measured at both locations using in-stream thermocouples or RTDs.

$$\Delta T_{superheat} = T - T_{sat} \tag{23}$$

APPENDIX H. DISCLOSURE OF INTEREST

APPENDIX H. DISCLOSURE OF INTEREST

For inclusion in the final report.