1 Country Report: United States

Executive Summary

The HVAC&R industry is at a pivotal juncture, transitioning to low-GWP (Global Warming Potential) refrigerants to combat climate change and adhere to stringent international regulations. This comprehensive report delves into state-of-the-art technologies, presents insightful case studies, reviews significant design optimizations, and offers a forward-looking outlook for the industry through 2030. It emphasizes the need for continuous research, robust regulatory frameworks, and technological advancements to ensure a sustainable future.

State-of-the-Art Technologies:

Refrigerant Innovations: The availability of A1 refrigerants is limited, with CO_2 (R-744) requiring substantial system redesigns and R-466A emerging as a promising candidate, though further validation is needed. A2L refrigerants, especially R-32 and its blends, are extensively researched and increasingly adopted due to their balanced performance and lower GWP.

Component and System Designs: Advanced heat exchanger designs utilizing non-round, shape-optimized tubes and innovative compressor and expansion valve modifications enhance system performance and efficiency. Tools like Approximation-Assisted Optimization (AAO) integrated with Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA) are critical in these advancements.

Regulatory Landscape: The Kigali Amendment to the Montreal Protocol, EU F-gas regulations, and the US EPA SNAP program are pivotal in driving the shift towards low-GWP refrigerants. These regulations set the framework for phasedown schedules and safety standards essential for the adoption of environmentally friendly refrigerants.

Safety and Flammability: Comprehensive safety research is imperative for A2L and A3 refrigerants. Updated safety standards and risk assessments ensure the safe use of these refrigerants in various HVAC&R applications.

Case Studies and Design Guidelines:

Practical Implementations: Case studies on R-516A demonstrate its viability as a low-GWP replacement for R-134a, showing comparable performance with significant emission reductions. Evaluations of unitary air-conditioning systems with refrigerants like R-290, R-32, R-452B, R-454B, and R-466A highlight their potential as alternatives to R-410A.

Design Optimization: Strategies for optimizing heat exchangers and system components are detailed, including the use of small-diameter tubes and advanced simulation tools. Emphasis is placed on enhancing system efficiency, ensuring safety, and meeting regulatory standards.

Design Optimization and LCCP Reduction:

Life Cycle Climate Performance (LCCP): Comprehensive LCCP assessments consider factors such as weather data, power factors, manufacturing emissions, and material usage. The impact of system efficiency and refrigerant leakage on emissions is highlighted, with regional comparisons providing valuable insights.

Key Findings: Annual energy consumption and refrigerant leakage are primary contributors to LCCP. Weather data discrepancies, such as the Urban Heat Island (UHI) effect, can significantly affect LCCP calculations, underscoring the importance of accurate local data.

Outlook for 2030:

Research Directions: Continued evaluations of promising A1 refrigerants, extensive safety research on A2L, like R-454B and A3 refrigerants, and the development of zero or near-zero GWP refrigerants are critical. These efforts aim to identify and mitigate safety challenges while achieving high performance with minimal environmental impact.

Regulatory Trends: Efforts in the US, Europe, and China to promote low-GWP refrigerants are expected to intensify, with ongoing updates to safety standards and policies facilitating the transition.

Technological Advancements: Innovations in system designs and refrigerant blends will drive future advancements, with optimization techniques enhancing component performance and system efficiency.

Conclusion:

The HVAC&R industry's transition to low-GWP refrigerants is imperative for environmental sustainability. This report underscores the necessity of ongoing research, regulatory support, and technological innovation. By continuing these efforts, the industry can achieve significant environmental benefits and meet future sustainability goals, paving the way for a greener, more sustainable future.

1.1 A Responsible Approach to Ensure the Safe Use of Flammable

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1.1.1 Overview

The HVAC&R industry is transitioning to low-GWP refrigerants due to environmental concerns and regulatory requirements[1]. Many of these low-GWP refrigerants are classified as having lower flammability, which has required the update of relevant safety standards and codes to implement these environmentally friendly refrigerants in the field in a timely and cost-effective manner.

Relevant safety standards updates have been underway for many years. Standard revision and code adoption processes are relatively slow, and regulations are advancing quickly. This is, in part, because of a lack of publicly available knowledge about the safe use of these refrigerants. To address this challenge, the US industry, HVACR society, and federal and local governments have collaboratively conducted research to understand refrigerant flammability and the safe use of these refrigerants for several years. The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) has been leading a research program aimed at generating publicly available and sound technical data to support code and standard activities related to the use of flammable refrigerants. This $ 6 million plus program was jointly funded by AHRI, ASHRAE, California Air Resource Board and US Department of Energy. The goal of the research initiative is to deliver scientific findings and produce publicly available technical references to support code and standard activities related to the use of flammable refrigerants.

Underwriters Laboratory and the American Society of Heating and Refrigeration Engineers (ASHRAE), along with stakeholders, including industry experts, have worked to update the safety standards. AHRI surveyed 46 relevant safety standards committees and WGs related to flammable refrigerants for technical data gaps. High-priority projects were identified and initiated to understand the refrigerant flammability risk, refrigerant charge quantity, refrigerant detector technologies, mitigation effectiveness, and equipment installation. The project reports have been shared with the standard communities and the public and have been used to substantiate changes to safety standards. The list of these projects is attached in Exhibit 1.

The industry is taking a systematic and conservative approach to ensure the safe use of flammable refrigerants with a great amount of effort to understand the potential risk. Several whole room-scale refrigerant leak and ignition tests were conducted for air-conditioning and refrigeration

[1] California Air Resources Board is considering a January 1, 2023 transition to low global warming potential refrigerants.

products beyond the worst-case scenarios with the events unlikely happening by using multiple strong ignition sources placed in locations where combustible mixtures were most likely to occur. In addition, the rapid liquid refrigerant release used in testing is an extremely low-probability type of leak. The testing suggested that, if used, refrigerant detection systems would need a faster response time than the 30-second response originally envisioned by the safety standards. This detection response time change has been adopted by relevant safety standards such as ASHRAE Standard 15 and UL 60335-2-40. Additional experimental investigation on potential ignition sources inside domestic residences demonstrated that many household ignition sources cannot ignite mildly flammable refrigerants such as burning cigarettes, sparks from common household appliances and tools, etc.

The industry is continuing its work to assess the use of mildly flammable refrigerants further to inform the continuous upgrading of the safety standards and to confirm previous learnings by other industries already using mildly flammable refrigerants.

Table 1-1 shows the launched projects in the USA. These projects were developed as the result of the survey of the safety standard community. They were prioritized among other identified research needs and considered high-priority projects by a group of experts, including industry experts, major safety standard committee chairmen and members, and representatives from the U.S. Department of Energy (DOE), Environmental Protection Agency (EPA) and National Institute of Standards and Technology (NIST).

Table 1-1: Launched High-Priority Low-GWP Projects in the USA

Projects	Funding Organiza tion	Project Description	Status
AHRTI-9007-01: Benchmarking Risk by Whole Room Scale Leaks and Ignitions Testing of A2L Refrigerants	AHRI	A2L refrigerant leaks and ignition testing under whole room scale conditions was conducted to develop data and insight into the risks associated with using A2L refrigerants versus A1 refrigerants while considering ambient conditions (temperature and humidity) and refrigerant lubricants.	Complet ed
AHRTI-9007-02: Benchmarking Risk by Whole Room Scale Leaks and Ignitions Testing of A3 Refrigerant	CARB	This project is to conduct A3 refrigerant leaks and ignition testing under whole room scale conditions, understand the risk associated with using A3 refrigerants, and provide test data to support future revisions of relevant safety standards associated with using A3 refrigerants.	Report under review
AHRTI-9008: Investigation of Hot Surface Ignition Temperature for A2L Refrigerants	AHRI	The objective of this work is to develop a test methodology to assist in the evaluation of the propensity of A2L refrigerants (R32, R1234ze, and R452B) to ignite on hot surfaces and to carry out testing per the new test methodology.	Complet ed
AHRTI-9009: Leak Detection of A2L Refrigerants in HVACR Equipment	AHRI	A thorough review of sensor technologies was conducted to evaluate available technologies that can be used to meet safety standards requirements of detecting A2L refrigerants and easily integrated into air-conditioning and refrigeration equipment. Infrared (IR) and Metal Oxide Semiconductor (MOS) sensors were found to be the most promising sensor	Complet ed

		technologies.	
AHRTI-9014: Assess Refrigerant Detector Characteristics for Use in HVACR Equipment	AHRI	The objective of the project is to assess refrigerant sensor and refrigerant detector performance requirements for class 2L, 2, 3 flammable refrigerants for use with indoor HVACR equipment, whether in an occupied space or a machinery room.	Ongoing
AHRTI-9015: Assessment of Refrigerant Leakage Mitigation Effectiveness for Air-Conditioning and Refrigeration Equipment	AHRI	The objective of this project is to demonstrate the efficacy of refrigerant leakage mitigation strategies contained within residential split-systems, packaged air-conditioning equipment and commercial refrigeration products.	Ongoing
ASHRAE-1806: Flammable Refrigerants Post-Ignition Risk Assessment	ASHRAE	The objective of this project is to understand the Severity of events where flammable refrigerants are ignited under different scenarios for various HVAC&R products.	ongoing
ASHRAE-1807: Guidelines for flammable refrigerant handling, transporting, storing and equipment servicing and installation	ASHRAE	This project accessed flammable refrigerant safety guidelines and/or requirements that exist domestically and internationally. The assessment will be used to propose requirements/guidelines for the safe handling, storing and transporting of flammable refrigerants	Complet ed
ASHRAE-1808: Servicing and Installing Equipment Using Flammable Refrigerants: Assessment of Field-made Mechanical Joints	ASHRAE	This project tested the leak-tightness of various types of field-made joints used to connect refrigerant piping and system components in HVAC&R equipment. The results of this project provided necessary data to suggest whether or not common types of joints, other than brazed or soldered joints, should be permissible for use in equipment containing flammable refrigerants.	Complet ed
ASHRAE-1855: Determination of the Impact of Combustion byproducts on the Safe Use of Flammable Fluorinated Refrigerants	ASHRAE	The overall objective of the project is to understand the HF and COF2 exposure risk if ignitions of flammable halogenated refrigerants occur and how to clean up following a variety of ignition events, as well as to identify knowledge gaps.	Ongoing
ORNL: Determination of setting charge limits for various types of equipment employing flammable refrigerants	DOE	The primary objective of the project is to examine the currently imposed limits for flammable refrigerant alternatives (A2L, A2, and A3) and identify reasonable adjustments to these limits as appropriate.	Complet ed
ORNL: Experimental Evaluation of Refrigerant Leak Characteristics for Different HVAC&R Equipment Types	DOE	The objective of the project was to quantify actual leak rates and duration for various pieces of equipment by conducting refrigerant leak tests under operating conditions representative of actual applications.	Complet ed
NIST: Modeling Tools for Flammability Ranking of Low-GWP Refrigerant Blends	DOE	The project is to develop modeling tools that can predict the burning velocity of arbitrary mixtures of R32, R125, R134a, R152a, 1234yf, and 1234ze(E), so that flammability of a blend can be minimized, while simultaneously maximizing performance related to other parameters.	ongoing

1.2 Refrigerants for Residential Air-conditioning Systems: A Review of the Recent Development of Alternative Refrigerants

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The low-GWP refrigerant is coming for the AC systems. We have already seen regulatory efforts by the California Energy Commission, as well as major manufacturers developing AC products compatible with low-GWP refrigerants. Meanwhile, researchers continue searching for optimal alternative refrigerants to improve the drawbacks of current candidates, as well as ultra-low-GWP refrigerants for the longer term. This summary provides a comprehensive update on recent regulations and research advancements in bringing low-GWP refrigerants to AC systems. It then tries to answer the following questions, at least partially: where we stand now toward ultra-low-GWP AC systems and how we should proceed to that direction based on past research efforts, including refrigerant mixtures and both A2L and A3 classified refrigerants.

1.2.1 Introduction

The residential sector represents 27% of global energy consumption and 17% of CO_2-eq. emissions [1]. The air conditioning (AC) systems contribute a significant portion of the energy consumption in residential buildings globally, as shown in Figure 1-1. While reducing energy consumption by AC systems continues to be a major pathway to lower global warming impacts, AC systems should use low-GWP refrigerants dominantly for the foreseeable future. The refrigerants used in residential systems have evolved over decades from ozone-depleting R22 to chlorine-free hydrofluorocarbon (HFC) R410A. Note that some developing nations still use R22. R410A, unfortunately, has a high GWP of 2,088. It is one of the major greenhouse gases as it leaks from air conditioning systems. The US Environmental Protection Agency (EPA) estimated that residential and light commercial AC equipment contributed about 22% of total national HFC emissions, or 36.7 million metric tons of carbon dioxide equivalent [3]. Therefore, it is necessary to find low-GWP replacement candidates for R410A. The ideal candidate should exhibit similar or better performance compared to R410A, while not introducing extra product design constraints.

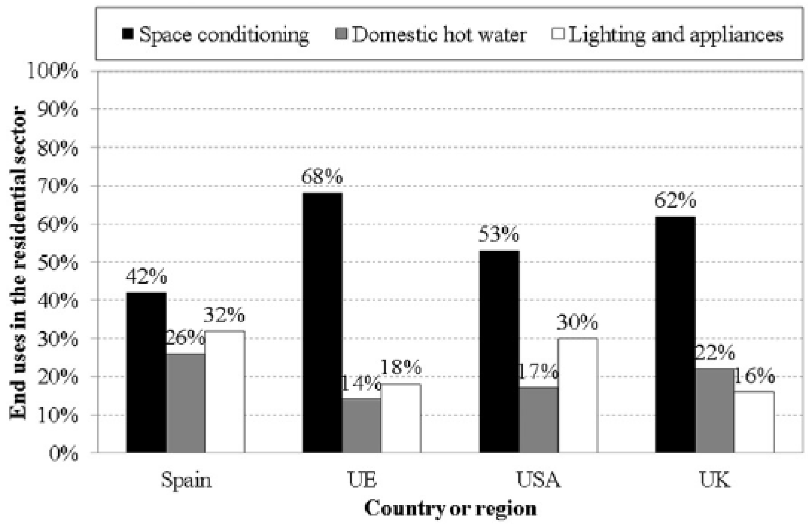

Figure 1-1: Energy consumption by end-uses in the residential sector [2]

Many industry and academic efforts have been carried out to search for candidates in residential and many other applications. A considerable number of reviews have been conducted for refrigerant research in recent years. The following are reflections of recent reviews. Dalkilic and Wongwises (2010) [4] reviewed mixture candidates for R12 and R22 based on theoretical analysis. The author found R290/R600a (40/60 by wt.%) and R290/R1270 (20/80 by wt.%) as the alternatives for R12 and R22, respectively. Benhadid-Dib and Benzaoui (2012) [5] provided a review of the basics of refrigerant phasing-down history, as well as regulation efforts. However, the review is limited to the very basics, with a few in-depth discussions on refrigerant candidates. Wang et al. (2012) [6] provided an overview of the AHRI Low-GWP Alternative Refrigerants Evaluation program. A series of alternative refrigerant candidates being evaluated in the program are introduced. The program covers a wide range of candidates for various applications. However, it focused on presenting objective results, rather than prioritizing refrigerants. We'll refer to many of the reports for specific refrigerants later in this report. Bolaji and Huan (2013) [7] analyzed potentials of various natural refrigerants in refrigeration and air-conditioning systems. The authors found natural refrigerants have good compatibility with materials and oils (for HCs) used in existing refrigeration systems. They concluded natural refrigerants are the most suitable long-term alternatives. McLinden et al. (2014) [8] explored the possibilities of low-GWP candidates from a set of about 1200 candidates. They selected 62 based on the targeted critical temperature range, 300 to 400 K. Thermodynamic analysis indicated no fluid was ideal. Sarbhu (2014) [9] presented a review of recent developments of possible substitutes for non-ecological refrigerants based on thermodynamic, physical, and environmental properties and TEWI (Total Equivalent Warming Impact) analysis. The author summarized refrigerant selection strategies for each utilization. In addition, they compared substitutes for R22 and found that R290, R600, and R152 have a minimal TEWI. Mota-Babiloni et al. (2015) [10] concluded in his recent review that there is yet to be a perfect low-GWP candidate for commercial refrigeration systems. His suggestions were that existing candidates had pros and cons: propane was for low-charge systems, low-GWP HFC/HFO was suitable for drop-in and retrofit, and CO_2 was applicable for transcritical systems.

Pham et al. (2016) [11] reviewed interim- and long-term candidates for R410A in unitary air conditioning and heat pump systems. The author concluded that R32 and HFO blends (such as R452B/R454B) are interim candidates offering balanced tradeoffs among flammability, efficiency, and cost. From the authors' perspective, paths towards long-term candidates (ultralow-GWP) still encounter many uncertainties. Domanski et al. (2017) [12] summarized the screening process and presented the performance of the "best" replacement fluids for small and medium-sized air-conditioning and refrigeration applications. The study shows that the low-GWP refrigerant options are minimal, particularly for fluids with volumetric capacities like those of R-410A or R-404A. Babiloni et al. (2017) [13] reviewed recent studies on low-GWP refrigerants because of the F-gas regulation. The author found most studies focused on mobile air conditioners with pure HFOs, due to earlier control by the F-gas regulation. Besides, the author noticed increasing trends in synthetic alternatives for domestic refrigerators and expected it to continue into other applications. Harby (2017) [14] conducted a review of recent studies on hydrocarbons as alternative refrigerants in refrigeration, air conditioning, heat pumps, and automobile air conditioning systems. He concluded hydrocarbons have advantages in terms of environmental impact, energy efficiency, refrigerant charge, and discharge temperatures. Future research directions were suggested on HFC/HC mixtures, low-charge large-capacity systems, and equipment reliabilities. In another review on natural refrigerants by Abas et al. (2018) [15], the author proposed a parametric quantification-based model to optimize the natural refrigerant selection process. It was concluded that CO2, NH3, and ethane performed better than other synthetic ones. Yet, more synthetic refrigerants were overlooked and should be investigated. Recently, Ciconkov (2018) [16] reviewed the history of refrigerant transitions and discussed the existing approach: first to low-GWP candidates, followed by a search for long-term solutions. He raised the question of economic and policy costs with this approach. He then suggested a new approach: phase-in of natural refrigerants directly.

Table 1-2 summarizes major literature reviews on alternative low-GWP refrigerant research since 2010. While there have been newly published reviews every year, there is a lack of a dedicated review covering recent progress in developing and searching R410A alternatives used in residential air conditioning systems. The majority of the reviews either cover all refrigerant research or focus on subcategories of refrigerants (e.g., natural refrigerants only). Besides, most reviews focused on academic research progress without much attention to advancements in regulations. Yet we have seen many regulation advancements, especially in recent years, to promote A2L low-GWP candidates for residential AC systems.

It is our objective to provide the most updated and comprehensive review of advancements in low-GWP alternatives for R410A in residential air conditioning applications from research and regulation perspectives. The review aims to provide the most updates by researchers, engineers, and enthusiasts concerning the status of the R410A phase-down. We anticipate the work will shed some light on future research and regulation directions in terms of interim and long-term. The review is organized as follows: First, we discuss research advancements of all alternative candidates for R410A. We categorize candidates based on ASHRAE standard 34 for refrigerant safety [17], namely A1, A2L, A3 and B groups. Second, we review regulation efforts based on geographical regions. Finally, we conclude the review by suggesting future R&D directions based on anticipated regulation moves.

Table 1-2: Summary of recent reviews on refrigerant research

Author	Focus	Highlights
Dalkilic and Wongwises (2010) [4]	Theoretical analysis R12, R22 and R134a replacement with mixtures from (R134a, R152a, R32, R290, R1270, R600 and R600a)	R290/R600a (40/60) best alternative for R12; R290/R1270 (20/80) best alternative for R22
Benhadid-Dib and Benzaoui (2012) [5]	Briefing on refrigerant phasing history up to 2015.	Cover the basics of most refrigerants, and overview of regulation efforts
Wang et al. (2012) [6]	Alternative refrigerants for air conditioners, heat pumps, chillers, water heaters, ice makers and refrigeration equipment	AHRI led, industrial-wide cooperative program lasts years, covers dozens of candidates
Bolaji and Huan et al. (2013) [7]	Natural refrigerants: R717, HCs, R718, R744	Reviewed potential application of major natural refrigerants, including oil compatibility
McLinden et al. (2014) [8]	62 candidates with critical temperatures within 300 to 400 K	Thermodynamic analysis indicates no ideal fluid
Sarbu (2014) [9]	Candidates for air conditioning, heat pump and commercial refrigeration applications	Describes refrigerants selection based on properties
Mota-Babiloni et al. (2015) [10]	Commercial refrigeration working fluids R404A, R507A and their substitutes: HC, Natural (CO2, NH3), HFC, HFO/HFC mixture	No universal solution: propane for low charge; low-GWP HFC or HFC/HFO mixture as drop-in or retrofit; CO2 for transcritical
Pham and Monnier (2016) [11]	R410A interim replacements (R32 and other A2L HFO blends) and longer term replacements (natural, HCs)	Dedicated overview for R410A, different paths towards long term ultra-low-GWP
Domanski et al. (2017) [12]	R410A and R404A candidates review	Comprehensive database search based on modeling and cycle analysis
Mota-Babiloni et al. (2017) [13]	Focused on recent low-GWP investigations because of F-gas Regulation for applications of all sizes	Unique perspective looking into policy driven advancements in low-GWP research and commercialization
Harby (2017) [14]	Hydrocarbon as replacements in refrigeration, AC and automobile AC	Comprehensive review of latest HC research efforts
Abas et al. (2018) [15]	Major natural and synthetic refrigerants	A parametric quantification-based model for natural refrigerant selection and optimization
Ciconkov (2018) [16]	Historic overview of refrigerants transitions up to now and future envisions, focusing on natural refrigerants and HFOs	Proposed direct phase-in of natural refrigerants instead of step-by-step phase down

1.2.2 Review of R&D progress in candidates

1.2.2.1 A1 candidates

<u>CO_2</u>

A1 candidates are rare for residential air conditioning systems. Carbon dioxide (CO_2) is one of the two major candidates. CO_2 (often referred as R744 in refrigeration industry) is made both naturally and artificially – artificially through the burning of fuel and other industrial processes. Approximately 0.04% of atmospheric air is CO_2. CO_2 is a natural refrigerant with an ODP of zero and a GWP of 1. Besides, it has additional properties, making it ideal for use in vapor compression refrigeration systems [18]:

- It is abundant in the environment at a low cost.
- It is non-flammable and non-toxic.
- As an inert gas, it is compatible with all common materials used in residential AC systems.
- Synthetic lubricants have been available for some time with good results.

That being said, CO_2 has other properties, making it hard to be used in AC systems. The foremost difference with other refrigerants is the low critical temperature (31°C of CO_2 versus 70°C of R410A). Since the heat sink (typically air) temperature for typical residential AC systems are usually above the critical temperature of CO_2, heat rejection of R744 does not undergo two-phase condensation, but happens as a dense gas progressively cooling at a constant pressure [19]. Therefore, CO_2 goes through the transcritical cycle. Another difference between CO_2 and other refrigerants is high operating pressure (72 bar of CO_2 versus 18 bar of R410A at 30°C). The high pressure penalizes the structural design of components of a refrigeration cycle. The penalization, however, can be counterbalanced with reduced component sizes since CO_2 has a higher volumetric latent heat and density.

Lorentzen (1994) revisited the idea of using CO_2 as a natural refrigerant. He presented a transcritical CO_2 cycle for an automotive air conditioning system. He then combined the cycle with commercial refrigeration and hot water production.

In a later review by Cavallini et al. (2007) [19]The author concluded that while CO2 does not compare favourably with traditional refrigerants in terms of energy efficiency, the proper system design can improve the situation significantly. However, the author's review found that CO_2 was mostly exploited in heat pump water heaters, mobile air conditioning systems, and small commercial refrigeration units.

Similarly, Groll and Kim (2007) [20] reviewed the latest research activities toward the transcritical CO_2 cycle, including residential air-conditioning systems. Most of the studies reviewed focused on energy efficiency comparisons between R22 and R410A systems. However, the author believed research had slowed down due to the limited induction of CO_2 systems in the market.

Tao et al. (2010) [21] performed experimental studies on a transcritical CO_2 residential air-conditioning with an internal heat exchanger. The author investigated the impacts of various parameters on the system COP and throttling loss, as well as exergy loss in all components, in a following publication [22]. They concluded that the gas cooler side air inlet parameters and system evaporating temperature are the main factors influencing the COP. The gas cooler was the most important component influencing throttling losses significantly and resulting in most exergy loss, about 30%. The author suggested that the design optimization on the gas cooler and expansion valve is crucial to the overall system performance.

Liu et al. (2012) [23] conducted modelling and experimental studies on a novel ejector expansion CO_2 air conditioning system. Test results show that with particular ejector geometries, the new cycle outperformed the conventional expansion-valve CO_2 cycle in COP and cooling capacity by about 31% and 32%, respectively. The author then suggested optimal ejector designs and operating conditions for maximum COP and cooling capacities.

Recently, Liu et al. (2018) [24] investigated using a CO_2 heat pump system as an environmentally benign solution for residential heating. Compared with other traditional heating methods, the CO_2 system has the highest primary energy ratio. However, it has the highest capital cost due to the dominant compressor cost. The same group conducted a follow-up analysis [25] for the CO_2 heat pump system in various climates. They concluded optimum discharge pressure and sub-cooling degree existed for maximum COP. The seasonal performance factor (SPF) could be improved by 32%, while the total cost was reduced by 7 to 15%.

Admittedly, CO_2 is more suitable in applications other than residential cooling systems, such as automobile air conditioning, commercial refrigeration, and water heating, where most studies have been focused on. Recent studies on CO_2 residential air conditioning systems have been limited to basic transcritical cycle, with limited significant innovations. Most studies focused on component and cycle optimization, aiming to reduce the system capital cost, one of the major barriers to promote CO_2 systems. That being said, not enough comparisons have been reported between CO_2 and R410A residential systems.

R466A

Another A1 candidate, R466A, is Honeywell's non-flammable, non-toxic, lower-GWP replacement for R410A. R466A is a blend of 49 wt.% R32, 11.5 wt.% R125 and 39.5 wt.% R13I1 (also known as CF_3I, a fire suppressant with a GWP of 0.4). It has a GWP of 730. Announced in 2018 by Honeywell, R466A is unveiled as Solstice N41 [26]. Note that R466A is not a "drop-in" replacement for R410A. Switching to R466A still requires minimal changes to equipment. One concern, though, is iodine, a major component of CF_3I, is a relatively expensive material with a limited supply of large sources [27].

Toshiba Carrier reported a similar performance between R466A and R410A in VRF system testing [28]. Recently, Midea and Sanhua partnered with Honeywell to commercialize the use of R466A [29] [30].

All that being said, more field studies and testing are still required to determine any possible adverse side effects. However, there have been a few published studies available regarding R466A. We expect to see more testing and evaluation studies soon when R466A is available to the public.

1.2.2.2 A2L candidates

A2L refrigerants are considered as a subcategory of A2 refrigerants from ASHRAE standard 34 [17]. They are mildly flammable. A2L refrigerants have the largest pool of candidates, compared to other categories.

R32

R32 is one of the most popular and extensively studied candidates. It is HFC (chemical formula CH_2F_2) with a GWP of 675, 32% of that of R410A. R32 has thermodynamic properties close to R410A, but the discharging temperature is. The commonly used compressor for R410A works well for the R32 in the operating envelope except in the high discharge temperature region, as shown in Figure 1-2. R32 was first successfully used in air conditioners by Daikin and was recently adopted by Daikin companies in North America [31]. However, there are still debates on how to use R32 and other A2L refrigerants safely, particularly in the North American market. The residential AC systems used in the North American market are mostly ducted systems with higher refrigerant charges than ductless systems used in Asia and European markets. Therefore, refrigerant leakage and flammability are a big concern.

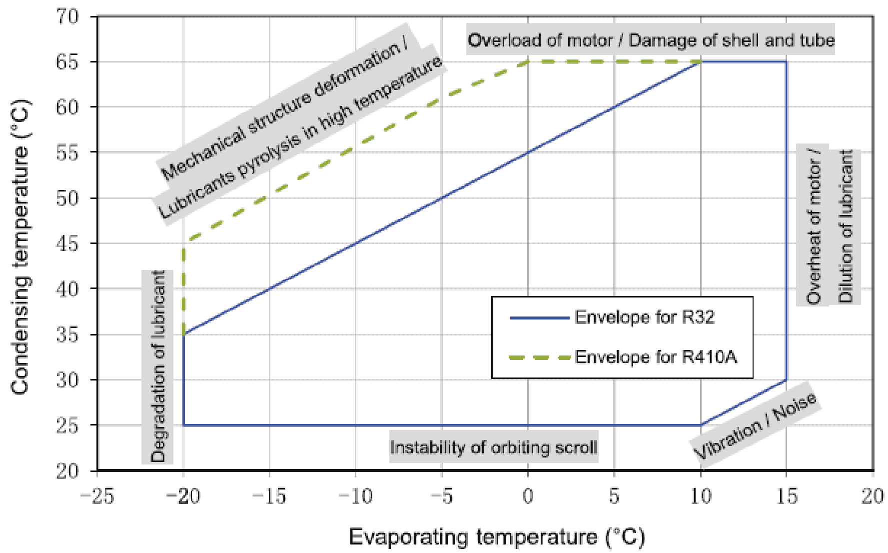

Figure 1-2: Comparison of R32 and R410A operating envelopes [32]

Many researchers have investigated and optimized the performance of R32 in residential air-conditioning applications. Chen and Yu (2008) [33] presented a new residential refrigeration cycle using R32/R134a. They found that while the mixture offers similar performance to R22 in the traditional cycle, it improves the COP by 8-9% in the new cycle. Pham and Rajendran (2012) [34] reviewed the status of replacing R410A with R32 through theoretical analysis, compiling test results and safety reviews, up to 2012. They concluded that R32 could serve as the initial candidate for the HFC phase-down plan. R32 also showed more advantages than HFO blends from test results. The natural refrigerants (like R290 and CO_2) did not show cost advantages as compared to R32. Barve et al. (2012) [35] performed an experimental comparison of R32 and R1234yf as drop-in replacements for an R410A residential heat pump unit. The author found both R32 and R1234yf have higher COP than R410A, except at very high temperatures. The author also concluded that system modifications were necessary for refrigerant charge management and TXV adjustments. Xu et al. (2013) [36] conducted an experimental comparison of R32 and R410A in a vapor-injected heat pump system. They found that R32 can increase the capacity and COP by 10% and 9%, respectively. Tian et al. (2015) [37] studies R32/R290 mixture (68%/32% by weight) as the drop-in replacement for R410A in household air conditioners. They found the mixture reduced the charge by up to 35% while increasing the capacities by 14 to 23.7%. Replacing the tube-fin heat exchanger with a microchannel heat exchanger further reduced the charge by 34%, diminishing flammability concerns. Zilio et al. (2015) [38] compared the performance of R32 and R410A in a packaged air-to-water reversible unit through modeling studies. Results indicated that the R32 system efficiency is comparable with the R410A system, with 6-7% higher capacities.

Mota-Babiloni et al. (2017) [39] presented a comprehensive review of R32 adoptions in residential air conditioning systems, particularly in Europe and the USA market. Highlights of their conclusions are: 1) R32 properties are well-defined, and current studies have focused on A2L mixture properties with R32; 2) R32 has a higher heat transfer coefficient and pressure drop than R410A, both in evaporation and condensation processes; 3) R32 presents similar system performance as R410A, yet most studies recommended system modifications to prevent high discharge temperatures, mainly for ducted systems.

We observed that the focus of adopting R32 has gradually shifted from performance to safety, in preparation for commercialization. Many industry associations and manufacturers have taken over major efforts in leading risk assessment studies and modifying safety standards. We'll cover this part in the "review of regulation process" section.

Mixtures

As stated above, R32-based A2L mixtures are a major branch of R410A alternative candidates. Researchers have proposed quite a few mixtures in the past decade, with many more under investigation. We considered the past few years as the "wild west" stage for all these candidates, as no superior has emerged. The following are representative studies on R32-based A2L mixtures.

Abdullah et al. (2014) [40] conducted drop-in tests and soft optimization of R32, D2Y60 and L41A. D2Y60 and L41A are two R32-based mixtures. D2Y60 has 40 wt.% of R32 and 60 wt.% R1234yf, with a GWP of 300. L41a has 73 wt.% R32, 12 wt.% R1234yf and 15 wt.% R1234ze. It has a GWP of 500. They found R32 slightly outperformed R410A, while D2Y60 performed worse than R410A and L41A was on par with R410A. The author also observed the suction line heat exchanger only enhanced the COP for D2Y60. Besides, system optimizations are necessary to bring the full potential of all three fluids/fluid mixtures. Bhanot et al. (2014) [42] found similar results through steady-state and transient simulations: R32 outperformed R410A while L41A was on par with R410A. D2Y60 performed the worst among all three. Lee et al. (2015) [41] applied D2Y60 in a multi-stage saturation cycle and found it increased COP by 46.9%, higher than CO_2 and propane.

Announced in 2015 by Ingersoll Rand, R452B (formerly DR55) is an HFO/HFC blend, with a GWP of 676. It has 67% R32, 26% R1234yf and 7% R125. It was claimed to have a 5% energy efficiency improvement over R410A. Chemours admitted that R452B offered the better combination of performance and flammability than R454B, formerly DR5. Kujak and Schultz (2016) [47] investigated the performance and safety of R452B later. The study focused on adjusting mixture compositions to comply with safety guidelines. The author concluded that GWPs of non-flammable blends are needlessly being elevated at the expense of flammability safety risks that are low or non-existent.

Devecioglu (2017) [48] evaluated R446A (mixture of R32/R600/R1234ze(E) in 29/3/68 wt.%), R447A (mixture of R32/R125/R1234ze(E) in 68/3.5/28.5 wt.%), R452B, and R454B (mixture of R32/R1234yf in 68.9/31.1 wt.%) as R410A candidates. The author concluded that R452B is a suitable R410A alternative for heating mode, while R446A is more suitable for cooling mode. Nonetheless, the study indicated that all alternative refrigerants have higher SEER values than R410A, while they are close to each other. Chen et al. (2018) [44] studied R452B and R447B as substitutions for R410A in a sub-cooler vapor injection cycle. R452B is comprised of R32/R1234yf/R125 (67/26/7 wt.%), with a GWP of 676. R447B is comprised of R32/R1234ze/R125 (68/24/8 wt.%), with a GWP of 714. They found that the capacities of the R452B and R447B systems are lower than the R410A system, ranging from 0.7% to 15%. The author concluded that R452B and R477B are more appropriate for high outlet water temperature conditions, while R452B is also suitable for low ambient temperatures. In 2021, Johnson Controls announced that R454B will serve as the primary low GWP refrigerant for all new residential and commercial ducted HVAC systems manufactured as of Jan 1, 2025.

Cheng et al. (2017) [43] investigated R32/R1234ze(E) mixtures numerically under different concentrations in an air-source heat pump. They showed that as the mass fraction of R1234ze(E) changes from 0 to 100%, the heat capacity decreases by 67.2% while the COP is increased by 70.3%.

Kong et al. (2018) [45] proposed a new refrigerant mixture (R152a/R32 in 75/25 wt.%), with a GWP of 279. Modeling results show that the mixture has a higher COP than R410A when the supply water temperature is above 40°C. The author concluded the candidate is reasonable for use in air source heat pump systems for space heating.

1.2.2.3 A3 candidates

A3 refrigerants have been aggressively pursued, particularly in China, as direct replacements for R22.

Wu et al. (2012) [49] conducted an experimental comparison of R161, R290, and R22 in a residential heat pump. They found that R161 reduced cooling and heating capacity by 7.6% and 6.8%, respectively, while increased cooling and heating COP by 6.1% and 4.7%, respectively. It reduced the charge by 43%. Results indicate that R161 outperformed R29 with a lower discharge temperature.

Han et al. (2013) [50] studied the heat transfer characteristics of R161 in microfin-tube experimentally. The author found out that the heat transfer coefficient of R161 was about 23% higher than R22. The ratio of pressure drops (R22/R161) ranged from 0.88 to 1.65. The author performed a follow-up study (2017) [51] on heat transfer characteristics of R161/oil mixture. A new correlation was developed to facilitate compact heat exchanger design with R161.

Given that A3 refrigerants are extremely flammable, most researchers consider current safety standards to be far from ready for A3 refrigerants. Except in China, A3 refrigerants have government support as they are natural, and most AC systems come with low refrigerant charges in China. That being said, organizations in other regions began looking at A3 refrigerants recently. AHRTI just released A3 refrigerants (R290) safety testing results in 2019 [52]. The tests focused on the leakage and ignition behavior of A3 refrigerants. Observations will be used as crucial proofs in setting up safety standards for A3 refrigerants.

1.2.2.4 B-class candidates

R717 (Ammonia) has been used in industrial applications since the 1930s and is generally acknowledged as being the most efficient refrigerant. It has a low boiling point and is favored because it is a highly energy-efficient refrigerant with a minimal environmental impact, with zero ODP and zero GWP.

R717 has a wide range of applications. It is particularly suited for working from approximately 0°C to -30°C and is widely used for food preservation. R717 has some chemical properties different from fluorocarbon refrigerants, as it is flammable, toxic, and corrosive. Therefore, the handling and use of R717 require adequate safety measures. Water present in R717 systems creates several problems, including internal system corrosion and the creation of sludge that can cause blockages in the systems.

R717 is classified in B2 (flammable and toxic) according to ANSI/ASHRAE Standard 34. The practical limit (the maximum concentration in an occupied room that does not create acute effects such that it is impossible to prompt evacuation of the occupants) for R717 is 0.00035 kg/m^3. However, due to its odor, ammonia will give a warning signal. That being said, researchers and manufacturers rarely considered R717 or other B-type refrigerants in residential air conditioning systems due to safety concerns.

1.2.2.5 Summary

In summary, there exist many refrigerant alternatives for R410A. All existing candidates require system modifications and/or regulation changes. Several US HVAC companies decided on R454B as a low GWP solution for residential and commercial ducted HVAC applications manufactured as of Jan 1, 2025.

Figure 1-3 compares different refrigerants under GWP versus refrigerant density (pressure). It is observed that the majority of the low-GWP candidates are A2L type with mild flammability. Recent developments focused on refrigerant mixture development. Most mixtures use non-flammable composition to suppress the flammability of R32. Some candidates sacrifice some degree of performance. That being said, most of the mixtures are still A2L refrigerants. Therefore, safety research on them is still necessary.

Other promising candidates are mostly natural refrigerants. CO_2 has been studied for many years, yet it requires a significant overhaul of the system design. Therefore, it is less attractive than the other A1 candidates, such as R466A, which requires much fewer system changes. Some of the A3 refrigerants are being considered and studied, especially in China. However, they are far away from commercialization due to safety concerns. B-type refrigerants (mostly R717) have not been actively discussed in residential sectors.

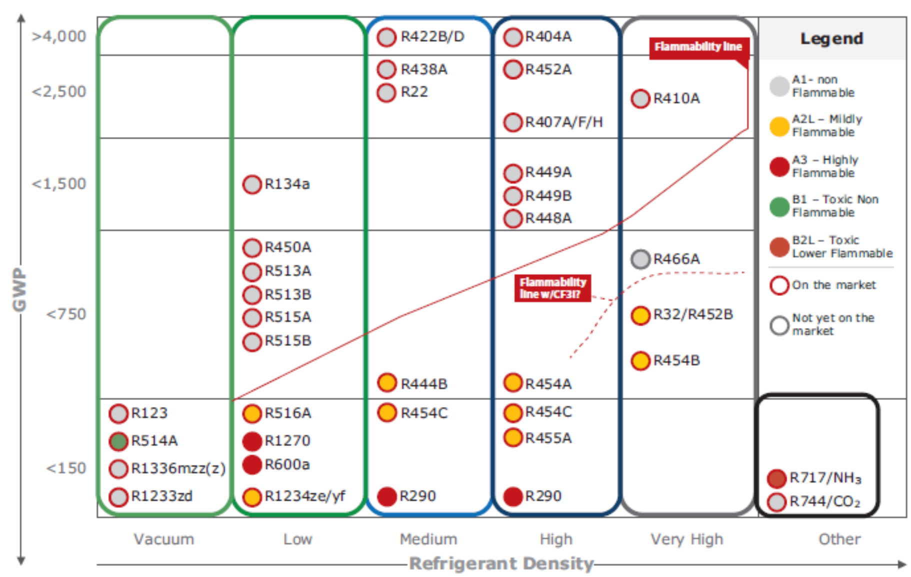

Figure 1-3: GWP versus Density (pressure) of the main refrigerant groups [54]

1.2.3 Review of regulation progress

Regulations have been key factors behind the transition to environmentally friendly refrigerants. The latest regulation efforts in introducing low-GWP refrigerants were initiated through the Montreal Protocol amendment, followed by the Kigali agreement in 2016. These two served as the basic framework based on which region-specific regulations/frameworks have been developed. The first section reviews the efforts in the USA (and North America regions), focused on R410A replacement in residential air-conditioning systems. In contrast, the second section examines those in other areas, notably Europe and East Asia.

1.2.3.1 Progress in the USA

According to the Montreal Protocol's North American HFC phase-down proposal, A5 parties need to reduce the HFC cap by 90% by 2035, while non-A5 parties need to reduce by 90% by 2050 [55].

EPA uses the SNAP program [56] to encourage the adoption of environmentally friendly refrigerants. It finalized two rulemakings in 2015 by accepting low-GWP refrigerants. Some flammable candidates are subject to use under restricted conditions, nonetheless. Notably, R32 was not approved as an alternative in residential central AC, mini-splits, and multi-splits.

The Environment & Climate Change Canada (ECCC) proposal [57] was HFC phase-down efforts from Canada. However, residential air-conditioning systems were not considered.

California proposed a climate reduction strategy in April 2016. The proposal aims to be in place if the Montreal Protocol is unsuccessful. According to the strategy, HFC emissions will be reduced by 40% by 2030. In particular, for the residential air conditioning systems, refrigerants with GWP higher than 750 will be prohibited, starting in 2021 [58].

Air-conditioning, Heating & Refrigeration Institute (AHRI) held a low-GWP Alternative Refrigerants Evaluation Program (AREP) [59]. The program was intended to test and present objective results for suitable alternatives rather than prioritize refrigerants. Phase 1 of the program was completed by the end of 2013, with nine candidates tested for R410A. Another six candidates for R410A were tested in phase 2 by the end of 2015. All the candidates promise low-GWP alternatives (GWP < 750), yet they are all classified as A2L under ASHRAE 34.

ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) has two standards addressing refrigerant classifications and usage restrictions: ASHRAE standard 34 and standard 15 [60], respectively. Type A refrigerants are considered non-toxic, with class 1 non-flammable, while class A2 and A3 are flammable. A2L is a subclass of A2 for mildly flammable refrigerants. However, ASHRAE 15 does not differentiate class 2 and 2L as of now. For residential systems, A2 or A2L refrigerants are only allowed in self-contained systems with less than 3 kg of refrigerant charge. A3 cannot be used with a few exceptions, such as portable units with less than 150 g of refrigerant charge or certain laboratories. Recently, the ASHRAE 15.2 subcommittee was formed to address the unique restrictions for A2L refrigerants.

It is a global trend to increase charge limits for flammable candidates, particularly for A2L refrigerants. In addition to ASHRAE 15, UL60335-2-40 [61] on heating and cooling equipment specifies restrictions on using flammable refrigerants in HVAC equipment. The standard is being reviewed and compared with its international counterparts (IEC60335-2-40), with final publications effective in 2023, for residential air conditioning systems.

Modifying the safety standards and codes to ease restrictions on the use of A2L and A3 low-GWP candidates is a critical step. While there have been tremendous efforts from ASHRAE standard committee 15.2, AHRI low-GWP AREP program, and UL, more building code revision needs to be addressed. Usually, the codes are on a 3-year cycle. The 2018 cycle for the ICC International Mechanical Code (IMC) is over, but it did not address A2L refrigerants. The 2018 cycle of the International Fire Code (IFC) is underway with two proposals addressing A2L refrigerants. The 2018 cycle of the IAPMO Uniform Mechanical Code (UMC) is underway, with several proposals addressing A2L refrigerants submitted. The UMC committee also agreed to reference ASHRAE standard 15 [55].

Recently, IEC60335-2-40 WG16 increased charge sizes for flammable refrigerants, while IEC 60335-2-89 refrigeration equipment was approved. UL 60335-2-40 increased charge sizes for flammable refrigerants as well, followed by ASHRAE standard 15-2019. CANENA WG12 is working on new charge sizes of flammable refrigerants for use in commercial refrigeration equipment.

To summarize, the code adoption process (as shown in Figure 1-4) of new refrigerants in the USA started with refrigerant safety classification in ASHRAE standard 34 and EPA SNAP program approval. The next step is to ensure the refrigerant usage complies with the safety standards (notably ASHRAE 15 and UL 60335-2-40). Following that is to be adopted by the relevant building codes, state, and local codes. 2021 is likely the earliest year that A2L and other low-GWP refrigerants will be addressed by major codes in the USA, with priorities lying in setting charge limits for A2L and A3 refrigerants.

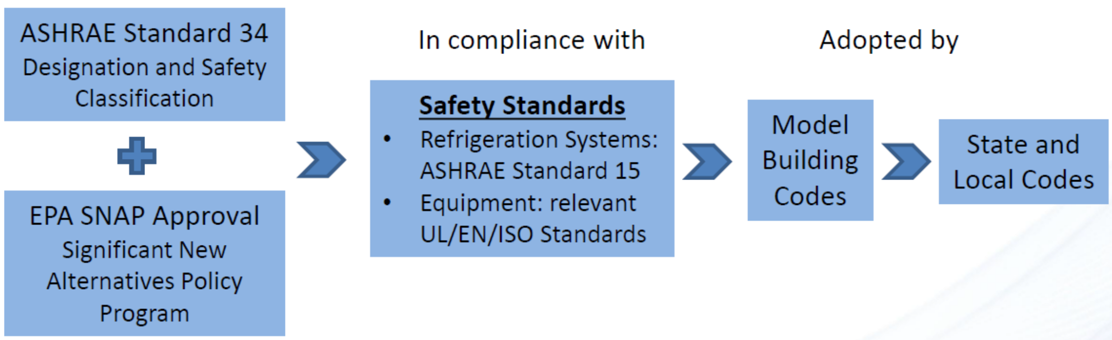

Figure 1-4: Code Adoption Process of New Refrigerants in USA [55]

1.2.3.2 Progress in other regions

The F-Gas regulations are leading regulation efforts in Europe [62]. The law aims to reduce high-GWP HFC refrigerants to 21% in 2030 as compared to the 2015 level. In particular, the GWP limits for portable room air-conditioning appliances and small split air-conditioning systems are 150 (starting in 2020) and 750 (starting in 2025), respectively. It is comparatively more aggressive than US phase-down goals.

In line with the GWP limits from the F-Gas bill, HCs (e.g., R290) are heavily considered for small self-contained air-conditioning systems with low refrigerant charge. R32 is regarded as the main alternative to R410A in single and multi-split systems.

It is worth noting that the F-Gas regulation has many other requirements affecting refrigerant usage besides GWP. Such requirements include refrigerant leak checking, leak prevention/repair, and proper refrigerant recovery. These requirements will be either new or stricter with flammable refrigerants.

Japan's HFC phase-down target is to reduce refrigerant usage by 80% by 2037, compared to the 2018 level. There are three significant regulations concerned with HFCs in Japan [63]. "Ozone Layer Protection Act" (2018 revised version) regulates HFC production and consumption. "Act on Rational Use and Proper Management of Fluorocarbons" (usually referred to as the F-gas Act) regulates the emission of HFCs. The last one, the "High-Pressure Gas Safety Act," covers the safety of flammable gas, including A2L refrigerants.

The F-gas Act aims to adopt a refrigerant GWP of 750 for residential air conditioning units by 2018 and 150 for mobile air conditioning units, which is similar to Europe's F-gas regulation. Japan Refrigeration and Air Conditioning Industry Association (JRAIA) is one of the leading bodies in the HFC phase-down in Japan. It led to A2L refrigerant risk assessment studies from 2011 to 2014, along with other regulation compliance studies. All the preparations contributed to the successful R32 residential model launch in Japan in 2012. As of 2017, nearly all residential air conditioners sold in Japan use A2L refrigerants (dominantly by R32). Hence, it is the leading market in adopting A2L refrigerants. JRAIA is investigating A3 refrigerants with risk assessment and safety standards, aiming to search for replacements for R32.

China has been aggressively looking into natural refrigerants recently [64]. While R22 is still allowed in developing countries like China, it must cut 35% for R22 usage by 2020 and phase it out for most uses by 2030 under the Montreal Protocol. While China has looked at low-GWP refrigerants such as R32 and other blends as replacements for R410A, it has inclined to natural refrigerants. R290 is the winner so far in room air conditioners. Many manufacturers recently showcased their hydrocarbon (R290) air conditioners and CO2 heat pumps, such as at China's 27th International Refrigeration Expo. Chinese companies have also actively participated in international standards, such as the International Electro-Technical Committee and Underwriting Laboratories. Manufacturers have invested considerable production lines for R290 room air conditioner units with over half a million capacities. Regulation bodies in China have progressively revised relevant domestic standards to pave the road for R290.

1.2.3.3 Summary

Recent advancements in agreeing to constrain global warming effects led to progressive refrigerant phase-down goals in different regions. Hence, we observed significant efforts by various organization entities. The efforts include new regulation initiatives and continuous updates on current standards, in particular for A2L and A3 refrigerants. That being said, efforts in different regions vary significantly due to different AC system designs and regulation processes. The North American market is notably different due to its unique ducted system design and legal culture. Japan and Europe are the leaders in A2L refrigerant adoption, mainly for R32. China is progressively leading the efforts in A3 refrigerant.

1.2.4 Conclusions

We have reviewed recent R&D and regulation advancements for R410A replacements in residential air conditioning systems. Below are highlights of our reviews:

There are very limited A1 candidates available as of now. CO_2 requires a system design overhaul and, therefore, has received limited consideration. R466A has been proposed recently as another promising candidate, yet full and credible evaluations are still needed.

Most of the available candidates are A2L refrigerants. R32 is the most investigated and proven. Many other candidates are mixtures, while most use R32 as their constituent. Research has been focused on two directions: developing new mixtures with low-GWP and low-flammability and safety research on A2L refrigerants. Several US HVAC companies decided on R454B as a low GWP solution for residential and commercial ducted HVAC applications manufactured on Jan 1, 2025.

A3 refrigerant is still only considered in systems with low charges due to safety concerns. It is in this nature that only China is progressively evaluating A3 refrigerants, where most residential systems have small charges. Its safety usage remains challenging, as reliable and cost-effective solutions are still missing.

Regulation efforts have been surging in the past decades. HFC phase-down goals and a large pool of A2L candidates contribute to the increasing activities in regulatory processes. In the US market, regulation efforts have focused on the safe usage of A2L refrigerants. Japan and Europe have been adopting A2L refrigerants successfully for some time. China leans more toward a direct path to A3 refrigerants.

Based on the summaries, we anticipate and recommend the following research directions:

- Comprehensive evaluations of R466A and developments of other potential A1 refrigerants: A1 candidates with low-GWP and similar performance are undoubtedly a perfect solution for R410A replacement. Any new developments meeting these criteria are worthy of research.
- Comprehensive safety research on A2L and A3 refrigerants: While R32 has been studied and safely applied in Japan and Europe for some time, similar refrigerants need to be reevaluated in the North American market as well due to different system designs and usages. A3 poses significant safety challenges and requires a reliable and cost-effective adoption solution. Such research is required.
- Evaluate the path beyond low-GWP refrigerants. Many existing candidates have a GWP in the range of 400 to 700. While this is a significant improvement compared to R410A (GWP of 2088), the ultimate solution would be zero or near-zero GWP refrigerant. A3 candidate and other natural refrigerants have such potential, yet systems must be redesigned significantly. Research on novel system redesigns or innovative new refrigerants is always needed.

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1.3 Review of Standards and Policies for Residential and Commercial Heat Pump Systems with Low-GWP Refrigerant

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1.3.1 Abstract

This report reviews the current standards and policies for residential and commercial heat pump (HP) systems with low-GWP refrigerants. Four standards and policies are reviewed: the Kigali Amendment to the Montreal Protocol, EU F-gas regulations, US EPA's Significant New Alternatives Policy (SNAP) Program, and China's National Green and High-efficiency Cooling Action Plan. The targets and requirements for low-GWP air conditioning are summarized for each standard and policy. This review aims to understand the policy drivers for refrigerant transition with a special focus on low-GWP HP applications and the GWP thresholds for the next step of HP component and system optimization.

1.3.2 Introduction

Under the Montreal Protocol on Substances that Depleting the Ozone Layer, HCFCs were scheduled to be completely phased out by 2030 in non-A5 Parties and A5 parties (UNEP, 2010). As a replacement for HCFC, the usage of HFC refrigerants grows rapidly. However, HFCs usually have high global warming potential (GWP). Improving the energy efficiency of air conditioners (heat pumps) and transitioning to low-GWP refrigerants are both important to reducing greenhouse gas (GHG) emissions.

This report reviews the current standards and policies for residential and commercial heat pump (HP) systems with low-GWP refrigerants. The objective of this review is to understand the policy drivers for refrigerant transition with a special focus on low-GWP HP applications and the GWP threshold requirements for the next step of HP component and system optimization.

1.3.3 Review of current standards and policies

1.3.3.1 The Kigali Amendment to the Montreal Protocol

The Montreal Protocol was established in 1987 to protect the Earth's ozone layer by phasing out ozone-depleting substances (ODS). It is a successful global agreement with financial mechanisms to support developing countries' transition. The ozone layer is expected to recover to 1980 levels by mid-century.

The Kigali amendment was adopted at the 28th Meeting of the Parties in Kigali, Rwanda, in 2016 to agree on a global schedule for phasing down HFC refrigerants for the high GWP. The goal is to reduce HFC production and emissions by more than 80% in the next 30 years. Table 1-3 shows the schedule of the Kigali amendment. It consists of three groups of Parties, each with a target phasedown date. The non-A5 countries have started following the schedule in 2019. It is noted that the market may move faster than the time defined in the schedule. This amendment is expected to avoid global warming by up to 0.5°C by 2100 (Xu et al., 2013), and it may be possible to double this contribution up to 1°C along with a 30% energy efficiency improvement of air conditioners (Shah et al., 2015).

Table 1-3: Kigali implementation schedule of HFC phase-down (EIA, 2016)

	Non-A5 (developed countries)	A5 (developing countries) Group 1	A5 (developing countries) Group 2
Baseline HFC component	2011-2013 (average consumption)	2020-2022 (average consumption)	2024-2026 (average consumption)
Baseline HCFC component	15% of baseline	65% of baseline	65% of baseline
Freeze	-	2024	2028
1st step	2019 - 10%	2029 - 10%	2032 - 10%
2nd step	2024 - 40%	2035 - 30%	2037 - 20%
3rd step	2029 - 70%	2040 - 50%	2042 - 30%
4th step	2034 - 80%	-	-
Plateau	2036 - 85%	2045 - 80%	2047 - 85%
Notes	Belarus, Russian Federation, Kazakhstan, Tajikistan, Uzbekistan, 25% HCFC component and 1st two steps are later: 5% in 2020, 35% in 2025	Article 5 countries not part of Group 2	GCC (Saudi Arabia, Kuwait, United Arab Emirates, Qatar, Bahrain, Oman), India, Iran, Iraq, Pakistan

1.3.3.2 EU F-gas regulations

Fluorinated gases (F-gases) include Hydrofluorocarbons (HFCs), Perfluorocarbons (PFCs), and Sulfur Hexafluoride (SF6). These gases are used in several industrial applications, including refrigeration and air conditioning, foam blowing, propellants, semiconductor manufacture, and electrical switchgear. Many f-gases have a high GWP value. The EU F-gas regulation aims to reduce emissions of these gases through stricter regulation.

The first EU F-gas regulation was issued in 2006, which focused on reducing emissions mostly by preventing leaks in systems and enforcing responsible end-of-life recovery and destruction of these gases. The revised regulation was effective on Jan 1st, 2015, and included further bans that impact the sale of certain air conditioning equipment using f-gases. The new F-gas regulation targets a 60% non-CO_2 greenhouse gas emission reduction between 2005 and 2030. According to the regulation Article 14, "From 1 January 2017, refrigeration, air conditioning, and heat pump equipment charged with hydrofluorocarbons shall not be placed on the market unless hydrofluorocarbons charged into the equipment are accounted for within the quota system.". Table 1-4 shows the AC-related F-gas product and equipment bans with target GWP thresholds (Linde, 2014).

Table 1-4: AC Related Additional F-Gas Product and Equipment Bans - Annex III

Ban Type and GWP Limits	Date	Commonly used gases and (GWP)	Possible alternatives* and (GWP)
14. Movable room air conditioning equipment (hermetically sealed equipment that is movable between rooms by the end-user) containing HFCs with GWP≥150	1.1.2020	R410A (2,088)	R290 (3) Future: HFO blends
15. Single split air conditioning system containing F-gases<3 kg, with GWP≥750	1.1.2015	R407C (1,774) R410A (2,088)	R32 (675) R290 (3) Future: HFO blends

1.3.3.3 EPA Significant New Alternatives Policy (SNAP) Program

Clean Air Act (Title VI) gives the US Environmental Protection Agency (EPA) authority to implement the Montreal Protocol and phase out ozone-depleting substances (ODS). Under Section 612, SNAP was established in 1994 to identify and evaluate substitutes for ODS. SNAP has reviewed over 400 substitutes. The objective of the SNAP includes the following:

- Identify and evaluate substitutes in end-uses that have historically used ODS;
- Study the overall risk to human health and the environment of existing and new substitutes;
- Publish lists and promote the use of acceptable substitutes;
- Provide the public with information

SNAP considers the following physical characteristics of the refrigerants:

- Ozone Depleting Potential (ODP)
- Global Warming Potential (GWP)
- Flammability
- Toxicity
- Occupational & Consumer Health/Safety
- Local Air Quality
- Ecosystem Effects

Refrigerants are listed by end-use. There are sixteen different end uses for Refrigeration and Air Conditioning, among which Residential and Light Commercial Air Conditioning and Heat Pumps are the essential categories. It includes room air conditioning such as window units, packaged terminal air conditioners (PTAC) and heat pumps (PTHP), and portable air conditioners; ducted (Central AC) or ductless systems (e.g., mini and multi splits); packaged rooftop units; water-source and ground-source heat pumps; and other products. SNAP considers the transition to low-GWP refrigerants. Table 1-5 summarizes the substitute refrigerants with GWP, SNAP listing date and status (EPA, 2015). Any new product using a refrigerant must have SNAP approval.

Table 1-5: Substitutes in Residential and Light Commercial Air Conditioning and Heat Pumps (EPA, 2015)

Substitute	Trade Name(s)	Retrofit /New	ODP	GWP	ASHRAE Designation (Safety Classification)	SNAP Listing Date	Listing Status
HFC-32		N	0	675	A2L	4/10/2015	Acceptable with Use Conditions: For use in self-contained room air conditioning; see rule for detailed conditions.
R-290 (Propane)		N	0	3	A3	4/10/2015	Acceptable with Use Conditions: For use in self-contained room air conditioning; see rule for detailed conditions.
R-441A		N	0	<5	A3	4/10/2015	Acceptable with Use Conditions: For use in self-contained room air conditioning; see rule for detailed conditions.
R-1270 (Propylene)		N	0	1.8	A3	12/1/2016	Unacceptable, as of January 3, 2017.
R-443A		N	0	2.5	A3	12/1/2016	Unacceptable, as of January 3, 2017.
R-404A	SUVA HP-62	R/N	0	3,920	A1	12/20/2002	Acceptable
THR-03		R/N	0	N/A	A1	12/6/1999	Acceptable with Use Conditions: For use in residential window unit air conditioning.
R-417A	ISCEON 59, NU-22	R/N	0	2,350	A1	12/6/1999; 6/16/2010	Acceptable
R-410B		N	0	2,230	A1	2/8/1996	Acceptable
R-410A	AZ-20, Suva 9100, Puron	N	0	2,090	A1	2/8/1996; 12/20/2002	Acceptable
R-407C	Suva 407C, Klea 407C	R/N	0	1,770	A1	2/8/1996; 12/20/2002; 8/21/2003	Acceptable
R-407A	Klea 60, Klea 407A	R/N	0	2,110	A1	1/2/2009	Acceptable
R-427A	Forane 427A	R	0	2,140	A1	1/2/2009	Acceptable
R-437A	KDD6, ISCEON MO49 Plus	R/N	0	1,810	A1	1/2/2009; 6/16/2010	Acceptable
R-458A	Bluon TdX 20	R	0	1,650	A1	7/21/2017	Acceptable
R-125/R-134a/R-600a (28.1/70.0/1.9)	NU-22 old composition	R/N	0	1,990	A1	6/16/2010	Acceptable

RS-44 (2003 formulation)		R/N	0	2,420	A1	6/16/2010	Acceptable
HFC-134a		R/N	0	1,430	A1	6/16/2010; 7/21/2017	Acceptable
R-125/R-290/R-134a /R-600a (55.0/1.0/42.5/1.5)	ICOR AT-22	R/N	0	2,530	A1	3/29/2006	Acceptable
R-422B	ICOR XAC1, NU-22B	R/N	0	2,530	A1	3/29/2006	Acceptable
R-422C	ICOR XLT1	R/N	0	3,390	A1	3/29/2006	Acceptable
R-434A	RS-45	R/N	0	3,250	A1	10/4/2007	Acceptable
R-438A	KDD5, ISCEON MO99	R/N	0	2,270	A1	10/4/2007; 6/16/2010	Acceptable
R-407F	Genetron Performax LT	R/N	0	1,820	A1	10/4/2011	Acceptable
R-417C	Hot Shot 2	R	0	1,820	A1	10/4/2011	Acceptable
R-421A	Choice R-421A	R/N	0	2,630	A1	9/28/2006	Acceptable
R-422D	ISCEON MO29	R/N	0	2,730	A1	9/28/2006	Acceptable
R-424A	RS-44	R/N	0	2,440	A1	9/28/2006	Acceptable
Ammonia Absorption		N	0	0	B2	9/5/1996	Acceptable
Evaporative Cooling		N	0	N/A	N/A	9/5/1996	Acceptable
R-507, R-507A	AZ-50	R/N	0	3,990	A1	9/5/1996; 12/20/2002	Acceptable
Desiccant Cooling		N	0	N/A	N/A	9/5/1996 6/16/2010	Acceptable

1.3.3.4 American Innovation and Manufacturing Act (EPA, 2020)

The American Innovation and Manufacturing (AIM) Act represents a significant legislative effort to combat climate change by phasing down high-global warming potential (GWP) refrigerants. Signed into law in December 2020, the AIM Act sets forth a comprehensive framework to reduce the production and consumption of these environmentally harmful substances. This chapter delves into the details of the AIM Act, its implementation, and its broader implications for the environment and the refrigeration and air conditioning industries.

Understanding High-GWP Refrigerants: High-GWP refrigerants are chemical compounds used primarily in refrigeration and air conditioning systems. These substances, including hydrofluorocarbons (HFCs), possess a high capacity to trap heat in the atmosphere, leading to a warming effect far more significant than carbon dioxide. As a result, their widespread use has become a significant concern for environmentalists and policymakers aiming to address global warming and climate change.

Legislative Background and Goals: The AIM Act emerged from a growing recognition of the need to address the environmental impact of high-GWP refrigerants. It aligns with international efforts, such as the Kigali Amendment to the Montreal Protocol, which seeks to phase down HFCs globally. The AIM Act outlines a clear timetable for reducing the production and consumption of these refrigerants in the United States, targeting an 85% reduction by 2036.

Key Provisions of the AIM Act: The AIM Act includes several key provisions designed to ensure the successful phase-down of high-GWP refrigerants:

- Regulatory Framework: The Environmental Protection Agency (EPA) is tasked with developing and enforcing regulations to achieve the phase-down targets.
- Industry Transition Support: The Act provides mechanisms to support the refrigeration and air conditioning industries in transitioning to low-GWP alternatives.
- Innovation and Research: It encourages the development of new technologies and refrigerants with minimal environmental impact.
- Enforcement and Compliance: Strict enforcement measures and penalties are established to ensure compliance with the phase-down schedule.

Implementation and Industry Response: The AIM Act has prompted significant changes within the refrigeration and air conditioning industries since its enactment. Companies are investing in research and development to create and adopt low-GWP refrigerants. The EPA has been actively engaged in stakeholder consultations to develop effective regulations and support the industry in this transition.

Environmental and Economic Implications: The phase-down of high-GWP refrigerants under the AIM Act is expected to yield substantial environmental benefits. By significantly reducing the emissions of potent greenhouse gases, the Act will help mitigate climate change and contribute to global efforts to limit temperature rise. Additionally, the transition to low-GWP refrigerants is anticipated to drive innovation and create economic opportunities within the green technology sector.

Challenges and Future Outlook: While the AIM Act sets a clear path for reducing high-GWP refrigerants, several challenges remain. These include ensuring the availability and affordability of low-GWP alternatives, addressing potential technical and safety issues, and maintaining industry compliance. Ongoing collaboration between government agencies, industry stakeholders, and environmental organizations will be crucial to overcoming these challenges.

The AIM Act marks a pivotal step in the United States' efforts to address climate change by targeting high-GWP refrigerants. By establishing a structured phase-down approach, supporting industry innovation, and fostering international cooperation, the Act underscores the nation's commitment to environmental sustainability. As implementation progresses, the AIM Act serves as a model for balancing environmental protection with economic growth, highlighting the transformative potential of proactive climate legislation.

1.3.3.5 China National Green and High-efficiency Cooling Action Plan

China released its Green and High-Efficiency Cooling Action Plan in June 2019. It is a national cooling plan based on the commitments in the Sino-French bilateral agreement in March. According to the agreement, China and France will “work together to promote the ratification and implementation of the Kigali amendment to the Montreal Protocol on the phasedown of HFCs and to promote the improvement of energy efficiency standards in the cooling sector.”

The China Cooling Action Plan sets targets for improving cooling-product energy efficiency by

2022 and 2030. The plan also describes key cooling-related priorities for China, including the development and reform of the environment and resources:

- strengthening energy efficiency standards;
- expanding the supply of green and high-efficiency cooling products, including through increased R&D on low-GWP and high-efficiency refrigerants;
- promoting green and high-efficiency cooling product consumption, including through government and enterprise green procurement;
- advancing energy-saving transformations, including through demonstration projects involving retrofits of central air-conditioning systems, energy efficiency upgrades to data-center cooling systems, cooling-system retrofits for zones and parks, and upgrades of cold-chain logistics; and
- deepening international cooperation, including the HFC phase-out pursuant to the Montreal Protocol and the promotion of green and high-efficiency cooling for all, in both domestic and export markets, through mechanisms such as the Belt and Road Green Cooling Initiative.

The China Cooling Action Plan also calls for strengthened compliance accountability in the cooling energy efficiency area, including enforcement spot checks and release of compliance information through the national credit information public disclosure platforms.

Especially the action plan promotes the deployment of low-GWP refrigerants with the following actions:

- Accelerate the amendment of product and safety standards.
- Energy efficiency top-runner and GWP value of refrigerant will be added to the energy efficiency labels of the primary cooling products.
- Guide manufacturers to quickly convert to low-GWP AC production lines, accelerate the phase-out of HCFCs, and limit HFCs usage.

1.3.4 Conclusions

This report reviews four standards and policies for heat pump systems with low-GWP refrigerants, including the Kigali Amendment to the Montreal Protocol, EU F-gas regulations, US EPA's Significant New Alternatives Policy (SNAP) Program and China's National Green and High-efficiency Cooling Action Plan. The objectives and air conditioner-related GWP requirements are summarized for the next step, HP component and system optimization.

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1.4 Very Low-GWP Refrigerant R-516A for Heat Pump Systems

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1.4.1 Abstract

Very low-GWP refrigerant R-516A has shown promising results as an R-134a replacement in medium-pressure chiller applications. In this work, R-516A has been further evaluated for its usage in air-to-water source heat pumps. Modeling results based on an R-134a oil-free compressor showed that R-516A could provide performance close to R-134a without any changes to the system, both in heating and cooling modes.

1.4.2 Introduction

R-134a is being used across many different applications ranging from medium-pressure chillers, refrigeration, mobile air-conditioning, and heat pumps. However, due to its relatively high GWP of 1300 (AR5), the HVAC&R industry is looking for alternative refrigerants to replace R-134a, with the automobile industry taking the lead. R-516A has a very low GWP of 131 (AR5) and showed promising results with matching performance as an R-134a replacement for medium-pressure chillers (Abbas et al., 2016; Schultz and Perez-Blanco, 2018). In this report, the authors will focus on the potential of using R-516A in air-to-water source heat pumps.

1.4.3 Refrigerant properties

1.4.3.1 Compatibility and stability

R-134a is used with polyol ester (POE) type lubricants. The miscibility of an earlier version of R-516A, ARM-42, was tested with typical POE oils over a wide range of concentrations and temperatures that cover the operating ranges typically encountered in commercial refrigeration and air-conditioning. The results showed that the operating range of ARM-42 was comparable to that of R-134a.

The stability of ARM-42 in the presence of materials that it would likely encounter in practical use was evaluated in sealed tube tests according to the ASHRAE Standard 97-2007. At the evaluated test conditions, ARM-42 and ARM-42/POE blends in the presence of steel, copper, and aluminum showed thermal stability similar to that of R-134a.

1.4.3.2 Flammability

R-516A's A2L safety classification indicates that it is a lower toxicity and lower flammability refrigerant. The flammability properties of R-516A are shown in Table 1-6. The burning velocity was measured using the worst case of formulation for flammability (WCF) as defined by ASHRAE 34.

Table 1-6: Flammability characteristics of R-516A

LFL (%v/v) @ 23°C	5
UFL (%v/v) @ 23°C	14
Burning velocity (cm/s) @ 23°C	4.8
Heat of combustion (MJ/kg)	11.2

1.4.3.3 Thermodynamic properties

The fluid and thermodynamic properties of R-516A are presented in Table 1-7 and compared to R-134a.

Table 1-7: Fluid and Thermodynamic properties of R-134a and R-516A

Refrigerant	R-134a	R-516A
GWP_{100} (AR5)	1300	131
Flammability Class	1	2L
Bubble point (°C) at 1atm*	-26.3	-29.6
Dew point (°C) at 1atm*	-26.3	-29.6
Critical Temperature (°C)*	101.1	96.8
Critical Pressure (MPa)*	4.0	3.6
Liquid Density at 25°C (kg/m^3)*	1206.7	1069.2
Vapor Density at 25°C (kg/m^3)*	32.4	34.5

* R-134a properties were obtained from REFPROP (Lemmon et al, 2018). R-516A properties were calculated by proprietary measurements and models.

The vapor pressure of R-516A is very close to that of R-134a for a wide range of temperatures with only minor differences observed at temperatures above 60°C, Figure 1-5.

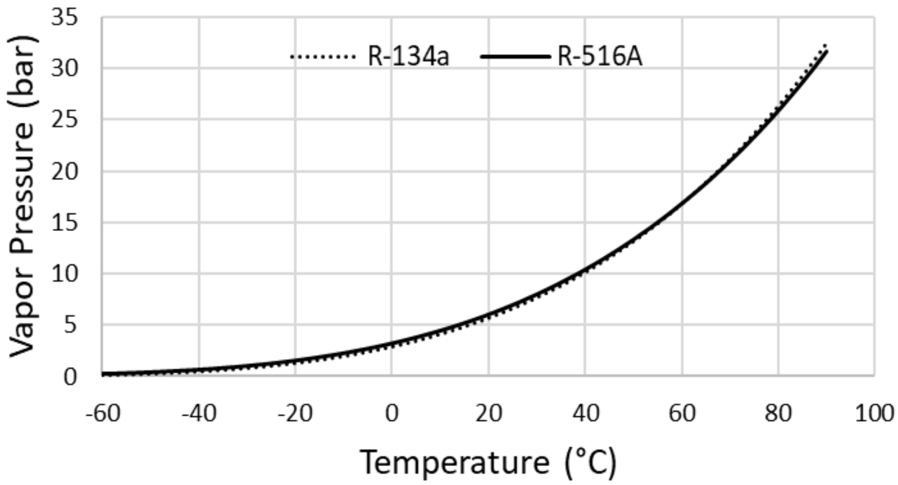

Figure 1-5: Vapor pressure of R-134a and R-516A

1.4.4 Results

In order to be utilized in a low ambient heat pump, the evaporating temperature must be below -25°C, which makes R-516A a suitable refrigerant for this application. R-516A was compared to R-134a based on an oil-free R-134a compressor model with the same electricity power input inside the aero map.

The performance of the refrigerants was calculated at the following conditions:

Cooling (ARI)

Evaporation Temperature = 7.2 °C | Evaporator Superheat = 11.1 K

Condensation Temperature = 54.4 °C | Condenser Subcooling = 8.3 K

No economizer

Heating

Evaporation Temperature = -7 °C | Evaporator Superheat = 1 K

Condensation Temperature = 50 °C | Condenser Subcooling = 3 K

Temperature approach of economizer = 0 K with flash tank

When the compressor performance of R-516A was compared to that of R-134a with the same motor design, both capacity and EER for cooling and COP for heating of R-516A were found to be slightly lower than R-134a as shown in Figure 1-6 and Figure 1-7. In terms of compressor operation speed, R-516A showed nearly identical speed with R-134a.

The capacity and efficiency performance of R-516A in the heating mode was closer to R-134a, which becomes critical with the choice to optimize product designs to either cooling or heating, depending on the primary application driver.

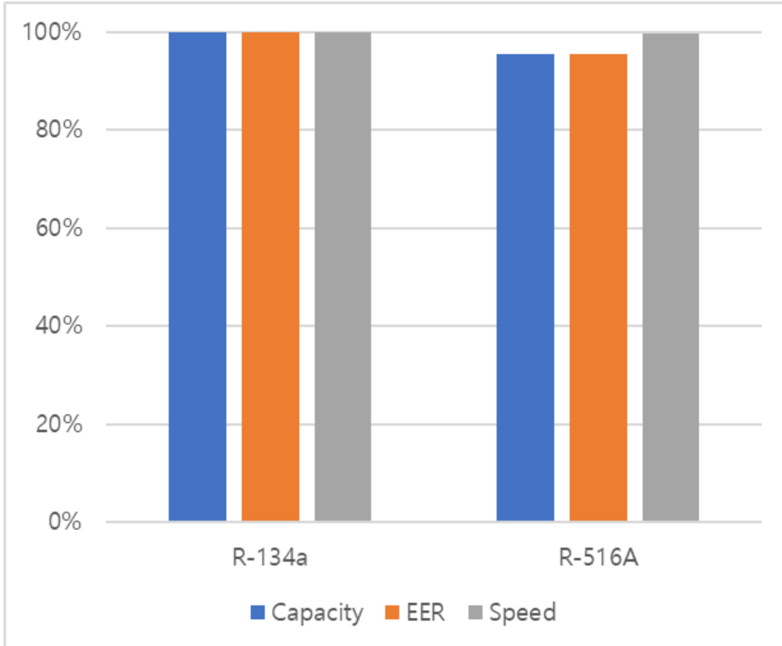

Figure 1-6: Cooling mode compressor performance prediction of R-134a and R-516A in a R-134a optimized compressor.

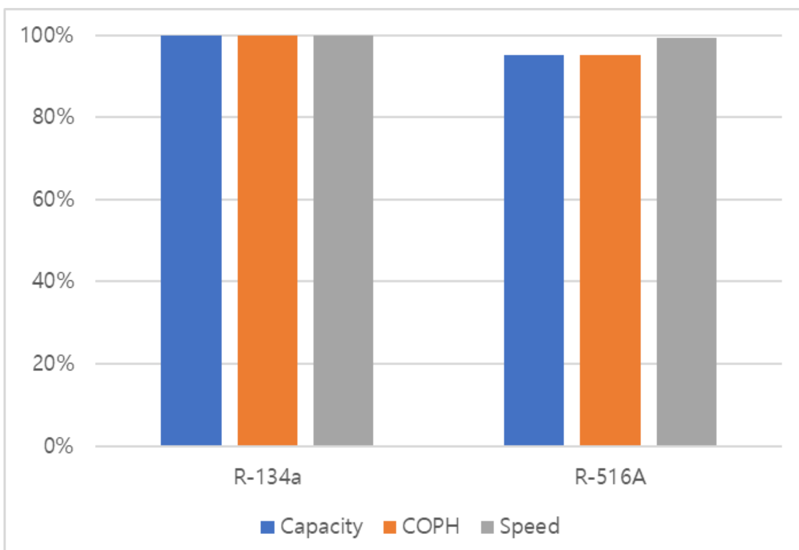

Figure 1-7: Heating mode compressor performance prediction of R-134a and R-516A in a R-134a optimized compressor.

In terms of discharge temperature, R-516A showed approximately 6 K lower discharge temperature at the cooling condition and 8 K lower at specific heating conditions than those of R-134a. These results make R-516A more favorable in high-lift conditions.

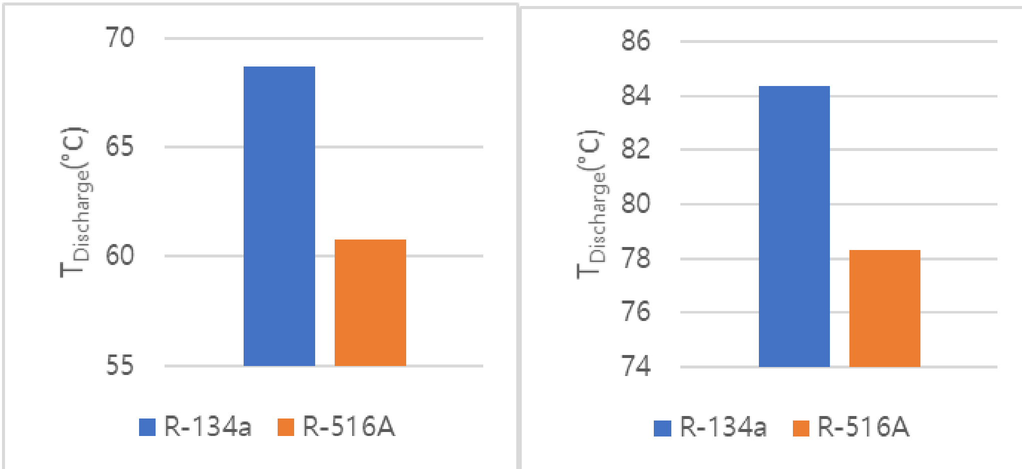

Figure 1-8: R-134a and R-516A cooling (left) and heating mode (right) discharge temperatures in an R-134a optimized compressor.

1.4.5 Conclusions

R-516A showed comparable performance to R-134a both in cooling and heating mode when evaluated in an R-134a optimized oil-free compressor while providing a 90% reduction in direct emission. The heating performance of R-516A was closer to R-134a than in cooling mode. Similar stability, compatibility, and oil miscibility of R-516A to R-134a will also allow easy adoption of R-516A as a replacement fluid.

1.4.6 Nomenclature

COP	Coefficient of Performance
EER	Energy Efficiency Ratio
GWP	Global Warming Potential

1.4.7 Acknowledgement

The authors would like to acknowledge Leping Zhang (Danfoss China) and Laurent Abbas (Arkema France) for their contribution to this work.

1.4.8 Reference

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1.5 Comprehensive Investigations on Life Cycle Climate Performance of Unitary Air-Conditioning Systems

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1.5.1 Introduction

Heating, Ventilation, and Air Conditioning (HVAC) systems represent up to 49% of total household energy usage (Song et al., 2020). Therefore, HVAC systems' environmental impact is under people's concern due to recent severe climate change and global warming issues. There has been increasing discussions and studies in the industry to migrate to lower GWP refrigerants (Motta, 2021) (*28-37_Flammable_Refrigerant_Standards_Update_Roundtable, 2021*) (Gao et al., 2021). A holistic evaluation of the HVAC system's environmental impact during its life cycle requires the translation of greenhouse gas (GHG) emissions from direct refrigerant leakage, indirect fuel consumption, and embodied equipment emissions (Andersen et al., 2018). The Institute of International Refrigeration (IIR) developed the Life Cycle Climate Performance (LCCP) evaluation, which adopted the rigorous approach to identifying and quantifying the direct and indirect environmental impact over a stated life cycle (Choi et al., 2017). The LCCP has been used to evaluate the LCCPs of different HVAC systems in the past ten years, as shown in Table 1-8. Horie et al. (2010) assessed the LCCP of the residential heat pump in Japan. Zhang et al. (2011) developed an LCCP tool for a residential heat pump for four U.S. cities. Li (2015a) evaluated the LCCP of various Packaged Air Conditioners (PAC) involving micro-channel heat exchangers for typical U.S. cities. Troch et al. (2016) and Lee et al. (2016) conducted an LCCP evaluation for the same heat pump system in five U.S. cities. Choi et al. (2017) developed an LCCP model and evaluated it for South Korean weather conditions. Wu and Jiang (2017) developed an LCCP calculation software to analyze different climate zones in China. Kim et al. (2018) applied a Neural Network algorithm to predict the LCCP value using three different U.S. weather conditions. In most of the past LCCP works, the environmental impact of the system was not evaluated in different countries but rather evaluated in one country. Also, half of the literature only concentrated on R-410A. Almost none of them discussed the recently announced refrigerants like R-466A and R-452B. Besides weather conditions, other factors can affect the LCCP evaluation. First, the grid emission factors are different in different countries. The range could be from 0.1 to 1.0 kg CO_{2e} per kWh (Transparency, 2018). This difference can bring obvious gaps in indirect carbon emission calculation. Second, the Embodied Carbon-dioxide Coefficients (ECCs) of the materials are different. They can cause discrepancies when calculating the carbon emission in the system's manufacturing phase. Previous studies did not consider all these differences at the same time. To fill the literature gaps, we evaluated the LCCP of a unitary air-conditioner (UAC) in different countries. UAC is a kind of HVAC system that combines heating, cooling, and fan sections in one or a few assemblies for simplified application and installation (Qiu, 2018). In addition to the regional difference, we also compared the system using different low GWP refrigerants, including R-290, R-32, R-452B, and R-466A, and used R-410A as the baseline. This study only discussed the fixed-speed compressor system. Finally, the effect of Urban Heat Island (UHI) on the LCCP was investigated using some open weather datasets.

Table 1-8: Recent LCCP evaluation research

Author (year)	System	Refrigerant	Country
Horie et al. (2013)	1.3 kW HP	R-410A, R-32, R-1234yf	Japan
Zhang et al. (2014)	11 kW HP	R-410A, R134a, R-1234yf	US
Li (2015)	13, 14 kW AC	R-410A, R-22	US
Troch et al. (2016)	11 kW HP	R-410A	US
Lee et al. (2016)	11 kW HP	R-410A, R-32, R-290, DR5, L41, D2Y60	US
Choi et al. (2017)	11 kW VI HP	R-410A, R-32, R-290	Korea
Wu and Jiang (2018)	-	R-410A	China
Kim et al. (2018)	12.4 kW VI HP	R-410A	US

1.5.2 Methodologies

1.5.2.1 LCCP Calculation Process

Troch et al. (2018) wrote IIR's guideline summarizing the calculation process of LCCP. The symbols in Troch's work were adopted in this paper. LCCP consists of direct and indirect emissions and is typically calculated in kg CO_{2e} unit, as shown in eq. 1:

$$LCCP = Direct\ emissions + Indirect\ emissions \tag{1}$$

Direct emissions are the refrigerant emissions during the usage phase in the equipment's lifetime and end-of-life (EOL) phase. Direct emissions can be calculated by Eq. 2:

$$Direct\ emissions = C \times (L \times ALR + EOL) \times (GWP + Adp.GWP) \tag{2}$$

Where C means a refrigerant charge (kg); L means average life of the equipment (yr); ALR means annual leakage rate (percentage of refrigerant charge); EOL means End of Life refrigerant leakage (percentage of refrigerant charge), GWP means Global Warming Potential (kg CO_{2e}/kg), Adp.GWP means GWP of Atmospheric Degradation Product of the Refrigerant (kg CO_{2e}/kg).

Indirect emissions include emissions from the power plants by consuming electric power for the equipment operation, manufacturing of materials, manufacturing of refrigerant, and disposal of the unit, as shown in eq. 3:

$$Indirect\ emissions = L \times AEC \times EM + \sum(m \times MM) + \sum(mr \times RM)$$
$$+C \times (1 + L \times ALR) \times RFM + C \times (1 - EOL) \times RFD \qquad (3)$$

Where AEC means Annual Energy Consumption (kWh); E.M. means CO_2 produced/kWh (kg CO_{2e}/kWh), which is the Grid Emission Factor (GEF) if electricity is the only energy source; m means a mass of unit (kg); MM means CO_2 Produced/Material (kg CO_{2e}/kg), which is also known as ECC; mr means the mass of recycling material (kg); R.M. means CO_2 produced/ recycled material (kg CO_{2e}/kg); RFM means refrigerant manufacturing emission (kg CO_{2e}/kg); RFD means refrigerant disposal emissions (kg CO_{2e}/kg); L, C, ALR, and EOL have the same meaning as the ones in eq. 2.

Direct emissions are more straightforward to calculate than indirect emissions. L, C, EOL, and ALR can be obtained from the manufacturers. GWP and Adp.GWP were widely reported in the literature (Bobbo et al., 2018). As for indirect emissions, AEC showed large differences in different climate countries for the same system (Choi et al., 2017). AEC was reported to be the most significant part of the LCCP calculation (G. Li, 2015b). AEC was usually estimated from simulations using EnergyPlus or other software as examples (Hong et al., 2016; G. Li, 2015b). AHRI standard 210/240 (2017) also provides a method to estimate the AEC of UAC (Alabdulkarem et al., 2014). We discussed these two methods in section 2.3. E.M. and MM vary in different countries. Different countries showed more than 100 times of differences for E.M. (Ryan et al., 2016) and 50% differences for MM (Ibn-Mohammed et al., 2013). However, limited studies showed a specific dataset for these values. We summarized the values in section 2.4. Other parameters like RFM and RFD can be obtained from the manufacturers.

1.5.2.2 Refrigerants Selection

Long-term usage of halogenated refrigerants in refrigeration and air conditioning systems has caused severe environmental damages. With the phasing down of high-GWP refrigerants, the replacement of currently used refrigerants requires safe, energy-efficient, and environmentally friendly characteristics. Nevertheless, no perfect alternative refrigerant exists, satisfying all these requirements (Venkatarathnam and Murthy, 2012). Many target parameters are involved, including flammability, GWP, compressor efficiency, compressor cost, heat transfer, and pressure drop. A trade-off map among these can be drawn, as shown in Figure 1-9 (Gilmour & McNally, 2010). Therefore, how to choose an appropriate refrigerant is significant. We need some metrics to combine these criteria. LCCP is a reasonable way to evaluate the performance, including system efficiency and environmental impact at the same time. Table 1-9 summarizes the refrigerants evaluated in the LCCP calculation. Till now, the manufacturing emissions for R-466A have not been reported. In the results part, we discuss R-466A manufacturing emissions' effects on the LCCP with different assumption values.

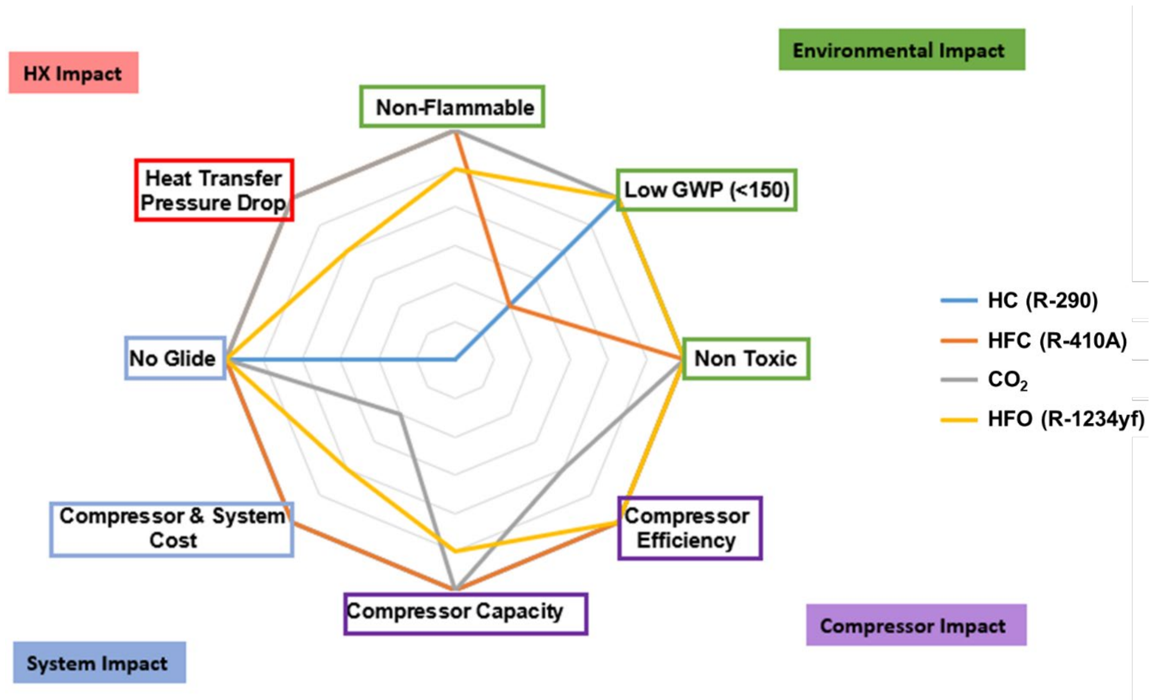

Figure 1-9: An example of the target parameter trade-offs involved in choosing the ideal refrigerant

Table 1-9: Target Refrigerants

Refrigerant	Safety Category	GWP	Drawback	Announced Year	Manu-factoring Emissions (kg CO_{2e}/kg)	Reference
R-410A	A1 (baseline)	2,088	High GWP	Honeywell, 1991	10.7	(Goto et al., 2001; Wang et al., 2009)
R-466A (N41)	A1	730	High cost	Honeywell, 2018	-	(Devecioğlu & Oruç, 2020; *Honeywell Announces R410A Breakthrough*, 2018; *Honeywell's N41 – a Blast from the Past*, 2018)
R-32	A2	675	Flammability	-	7.2	(Mota-Babiloni et al., 2017; Pham & Rajendran, 2012; Xu et al., 2013)
R-452B (DR-55)	A2	676	Flammability	Ingersoll Rand, 2015	8.9	(Kedzierski & Kang, 2016; S. A. Kujak et al., 2014)
R-290	A3	3	Flammability	-	0.05	(*AHRTI-9007-02_Final_Report.Pdf*, n.d.; Wu et al., 2012)

1.5.2.3 System Annual Energy Consumption

The AEC consists of the cooling power consumption and heating power consumption of the target system throughout the year. In a real-life application, field tests and energy surveys can help determine the HVAC system's annual energy consumption. However, the field test is not always available. To estimate power consumption, we need to know the cooling and heating loads and the system performance at a given ambient temperature. In this study, to compare the LCCP in different countries for different refrigerants, we used the simulation method to estimate the annual energy consumption.

1.5.2.3.1 System Performance

We developed models of a 10.5 kW and 115 kg-weight system (Alabdulkarem et al., 2014, 2015) using an in-house component-based steady-state vapor compression cycle modeling tool, VapCyc (Winkler et al., 2008). The models were validated by experiments using R-410A and R-32. The validation results of cooling experiments are shown in Figure 1-10. All the results agree with the experimental test data within 5% deviations. We used this model to predict the system performance in different ambient environments. In the experiments, an accumulator was used before the compressor to ensure a saturated vapor suction condition. In the model, an assumption of 2.1 K superheat at suction was assumed for these tests. An assumption of 2.8 K subcooling was used to predict the charge level. A constant isentropic and volumetric efficiency compressor model was used. The volumetric efficiency, isentropic efficiency, and mechanical efficiency were assumed to be 0.95, 0.75, and 0.95, respectively. The assumptions were based on our previous experiments (Alabdulkarem et al., 2015). Table 1-10 shows the compressor's Revolutions Per Minute (RPM) and displacement volume for different refrigerants and the predicted charge level. The charge level was consistent with the density of each refrigerant. The RPM and displacement volume were set to optimize the system performance with the capacity constraints. We also designed the system for the 12.3 kW system since for cities in hot climate regions like Miami and Phoenix, the 10.5 kW system could not suit the load requirement for a similar size room in other areas. Our study focused on office buildings with relatively higher occupants density and equipment loads than residential buildings. Thus, for cold countries in winter like Switzerland and Sweden, the systems could also be designed for the cooling season.

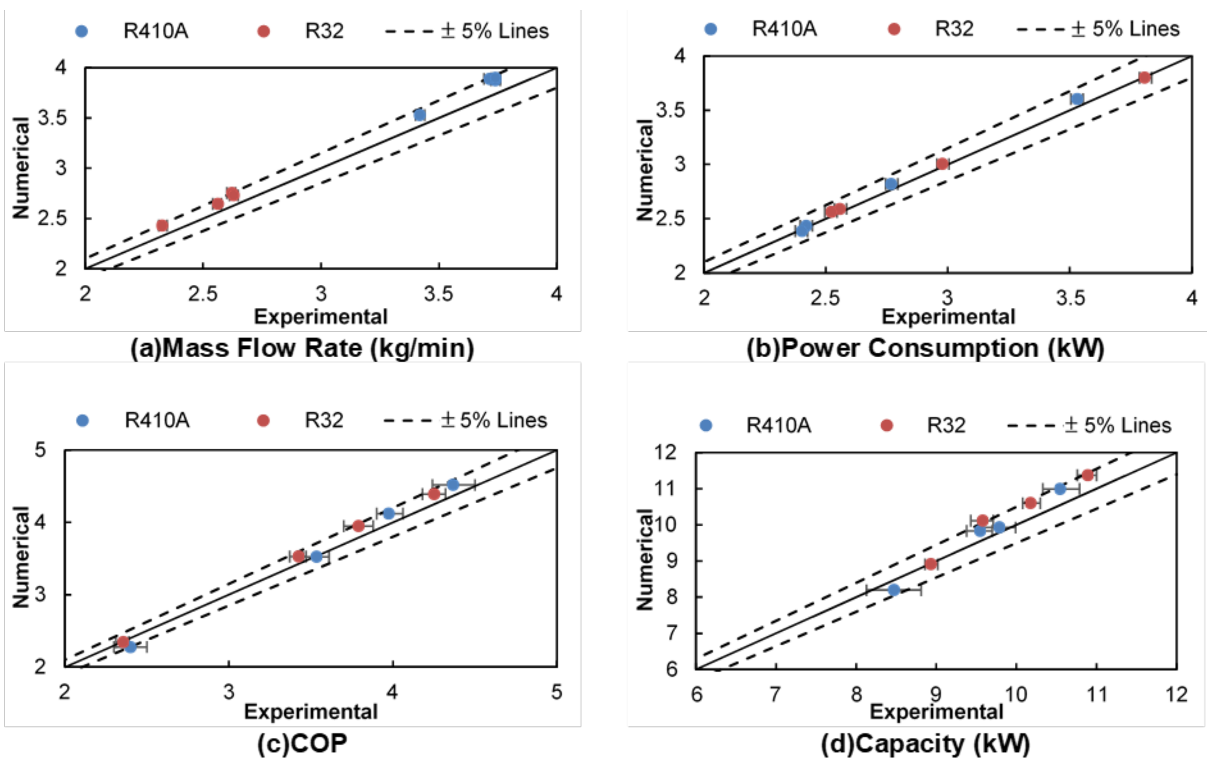

Figure 1-10: Cooling experiments validation results (Alabdulkarem et al., 2014)

Table 1-10: Design compressor displacement volume and predicted charge level

Capacity	Refrigerant	Compressor RPM	Compressor Displacement Volume (cm^3)	Charge (kg)
10.5 kW	R-410A	4,700	34	4
	R-290	7,300	43	1.2
	R-32	4,000	34	3.5
	R-452B	4,700	34	3.8
	R-466A	4,800	34	4.2
12.3 kW	R-410A	4,700	47	4.1
	R-290	7,300	61	1.8
	R-32	4,000	47	3.6
	R-452B	4,700	47	3.9
	R-466A	4,800	34	4.3

Figure 1-11 shows the simulation results of 10.5 kW capacity systems' COP for five refrigerants with different ambient temperatures in the range of 25 °C to 45 °C. R-290 has the best performance, while R-410A has the worst performance. R-32, R-466A, and R-452B have similar performances. From our modeling results, R-32 has a better performance than R-466A, while R-466A has a better performance than R-452B. When the ambient temperature increases, the performance of the three refrigerants gets even closer.

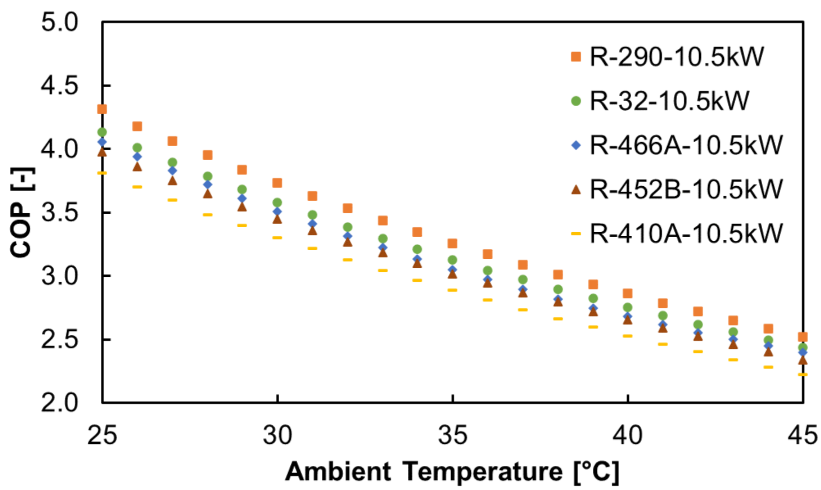

Figure 1-11: Cooling COP comparisons for different refrigerants

Some researchers studied different refrigerants' performances with different ambient temperatures. Kenneth and Steve reported that R-32 had a lower COP than R-452B when the ambient temperature was lower than 47 °C and higher COP when the ambient temperature was higher than 47 °C (Kujak, 2019). Binbin et al. (2020) studied tens of alternatives for R-410A and concluded that the COPs for R-290, R-32, R-466A, and R-452B were 5%, 1%, 1%, and 1%, respectively higher than the R-410A. Our results are consistent with the literature.

Since our study only considered a cooling-season-based design (high-density occupants and equipment), the system performances would be very close (within 1% differences) for different refrigerants at the same ambient temperature. The reason was that the heating loads were from 3 kW to 6 kW for different cities, which were less than half of the design capacity.

1.5.2.3.2 Load Prediction

AHRI standard 210/240 (Standard, 2017) provides an approach to estimate the load. This approach is called the temperature bin method. However, this approach applies to a fixed-speed system. If the system had a variable speed compressor, the compressor frequency's control logic would also affect the result. With the development of data-driven methods, some researchers used machine-learning-based models for load forecasting (Madonna and Bazzocchi, 2013). The data-driven approach requires a large amount of test power data. When the weather data is available, we could use a physics-based method to simulate the target building's load or a room by following the ASHRAE standard (*ANSI/ASHRAE Standard 34-2019*, n.d.). In this study, we chose the physics-based method to estimate the load since this method controls the variables, which are the regions and refrigerants. We considered a 10 m × 10 m room facing south in the Northern Hemisphere. Two windows were installed facing south and north. The ceiling and floor were assumed to be adiabatic. Other parameters can be found in Table 1-11. We also made the following assumptions to eliminate other factors impact on LCCP calculation: the optical depth parameters for the location were assumed to be constants through the year; windows had no shading. We used the model introduced by Wijeysundera (2015) to estimate the cooling and heating load of the target room.

Table 1-11: Parameters for simulation

Item	Value
Height	3 [m]
Window to Wall Ratio	0.6 [-]
Ground Reflectivity	0.25 [-]
Solar Absorptivity of Wall	0.8 [-]
Wall	Brick and a layer of insulation board
U-value of Wall	0.58 [$Wm^{-2}K^{-1}$]
Window	Double-glazed
Occupant	75 W for sensible heat, 55 W for latent heat
Occupant per unit floor area	0.1 [m^{-2}]
Equipment per unit floor area	13.5 [m^{-2}]
Light per unit floor area	4.5 [Wm^{-2}]
Working Hours	9:00-19:00

1.5.2.4 Grid Electricity Emission Factor and Material-Embodied Carbon Coefficients

The emissions from energy consumption are a dominant factor in the LCCP calculation. Different countries and regions have different power plant emission factors due to the resource portion difference (Choi et al., 2017). Carbon Footprint (2019) summarizes the country-specific electricity grid carbon emission factor in June 2019. The data for Asian countries is from G20 Green Report 2018 (Transparency, 2018), for European countries is from the Association of Issuing Bodies (*European Residual Mix | AIB*, n.d.), and for the U.S. is from the U.S. Environment Protect Agency database (US EPA, 2015). The second column of Table 1-12 shows the GEFs used in this study.

Table 1-12: GEF, Material usage, and ECCs

Weight: 115 [kg]		GEF [kg CO_2e/kWh]	ECCs [kg CO_2e/kg]			
Material (% usage)			Aluminum (12%)	Copper (19%)	Plastic (23%)	Steel (46%)
Average around world		0.623	13.1	2.71	3.31	3.02
EU	UK	0.277	6.58	2.71	-	1.80-2.89
	Sweden (SE)	0.012				
	Switzerland (CH)	0.014				
NA	US FL	0.467	5.65	3.00	2.80	1.80
	US AZ	0.425				
	US GA	0.457				
AS	JP	0.492	10.60	-	-	1.64
	KR	0.517	11.90	-	-	-
	CN	0.623	14.60	-	-	3.50-4.50

The carbon emission during the system's manufacturing phase is another factor that affects the LCCP calculation. Some previous studies used the same emissions values for the material in every country. For example, Choi et al. (2017) used the IIR's LCCP guideline (Life Cycle Climate Performance Working Group, 2015) to estimate the LCCP in Korea. However, IIR's LCCP guideline only provides the recommended values in the U.S. Some researchers, especially those working on the Life Cycle Assessment of buildings, have developed a database for different materials' Embodied Carbon Coefficients (ECC) in different countries (De Wolf et al., 2016). For this study, we used the Inventory of Carbon and Energy database developed by Hammond et al. (2011). For plastic and steel, we used the general values for these two materials. Some ECCs were not found in the literature for some countries. The average value around the work was used as a substitute in this study. As we can find in Table 1-12, the ECC for aluminum in the U.S. is around one-third of China's value. Thus, ECCs could be a crucial factor in the LCCP calculation for different countries.

1.5.2.5 Weather Station Data and On-site Weather Data

Most building simulation studies utilize data collected from weather stations. The most commonly used database includes the EnergyPlus built-in weather data, NOAA weather data, and TMY3 weather data. The first two datasets are the Actual Multi-Year (AMY) dataset, while the last one is a Typical Meteorological Year (TMY) dataset. Some researchers studied the difference between the AMY dataset and the TMY dataset (Kamel & Sheikh, 2020). They concluded that the dry-bulb ambient temperature had a significant impact on the simulation results. Most of the weather station data were collected around the airports. Some studies pointed out the temperature gaps between a city and an airport. Such a temperature gap in an urban area or metropolitan area due to human activities is called a UHI Effect (Kotharkar et al., 2018). This effect's leading cause is modifying land surfaces (Solecki et al., 2005) and waste heat generated (Y. Li and Zhao, 2012). Santamouris et al. (Santamouris et al., 2017) studied the UHI effect from 220 projects and found that 31% of the analyzed projects resulted in a peak temperature drop below 1 °C, 62% below 2 °C, 82 % below 3 °C, and 90 % below 4 °C. Munck et al. (2013) found that the increase in temperature was 0.5 °C in the situation with current heat releases, 1 °C with recent releases converted to only sensible heat, and 2 °C for the future doubling of air conditioning waste heat released to air in Paris. This temperature gap could bring some differences in LCCP calculation. Thus, we would compare the LCCP results using weather station data and weather data corrected by Santamouris's statistics (Santamouris et al., 2017).

We also measured the local ambient temperature. Thermocouples were installed next to an air conditioner outdoor unit in a campus building at UMD, College Park, US. The thermocouples were exposed in the air facing north and had no shadings. The ambient temperature tested was compared with the temperature data from Airport, College Park, US. The distance between the two places is 1.8 km. Figure 1-13 shows the comparisons between the two data. Figure 1-12 (a) shows the daily temperature measured on January 15, 2019. Figure 1-12 (b) shows the daily temperature tested on July 15, 2019. The blue line is the temperature tested in the campus reading through LabView, which is marked as UMCP. The red line is the temperature tested in the airport from the NOAA database. We could find that in winter, the UMCP campus temperature was 10-20 °C higher than the airport's temperature. Since the campus building sensors had no shading, solar radiation would have a significant effect on them. As a comparison, the weather stations' temperature sensors were usually stored in a shaded structure, which had less impact on the radiation. In the field test, the built-in sensors of the outdoor units are usually exposed to the air directly. Thus, the UMCP campus case should be closer to the field test case. This temperature gap could also be caused by human activity and other A.C. outdoor unit outlet waste heat. However, during the summer, the UMCP campus temperature had a higher peak but lower valley than the airport's temperature. Figure 1-13 shows a histogram of the two temperatures in the year 2019. 118-hour data points in the campus testing dataset and 43-hour data points in the airport dataset were not validated due to the power outage or broken database. We excluded these data points when we draw Figure 6. Thus, 8,599 data points exist in this figure. We used these two datasets separately to calculate the LCCP and discussed the differences. The finding is that the gap between the onsite data and the weather station data is much larger than what previous researchers assumed.

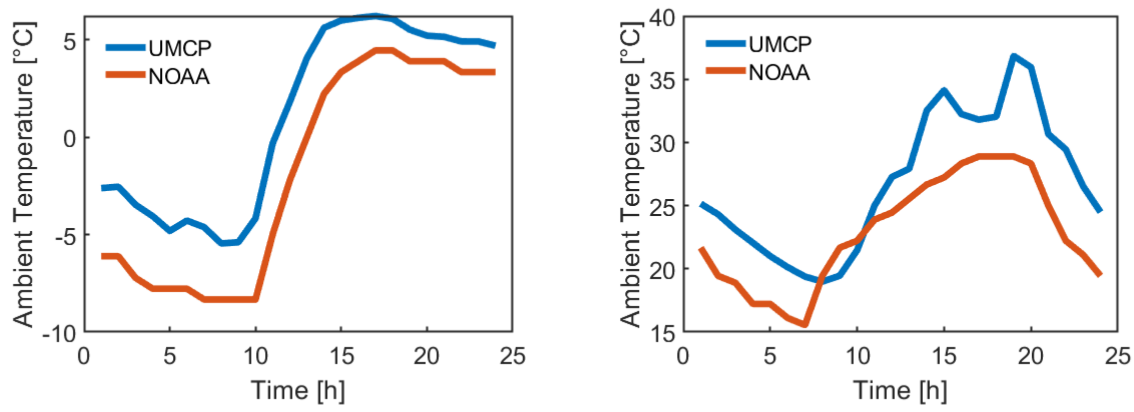

Figure 1-12: Comparison of Ambient Temperatures

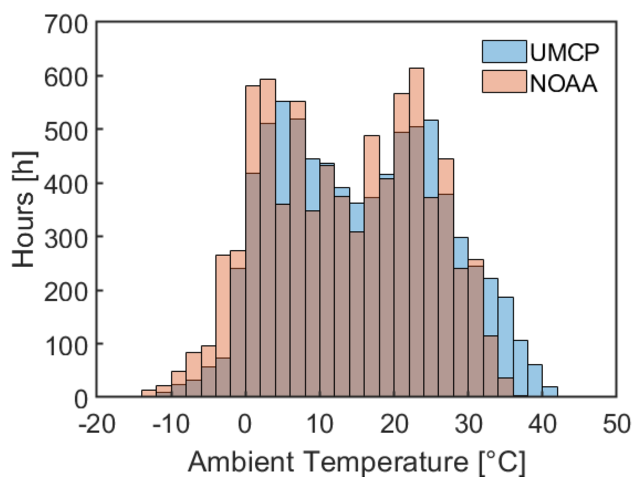

Figure 1-13: Histogram of the year 2019 ambient air temperature in College Park, MD, U.S.

1.5.3 Results

1.5.3.1 Different Countries and Regions

Figure 1-14 shows the LCCP results in different areas for R-410A as an example. "CollegePark1" and "CollegePark2" show the calculation results using UMCP campus weather data and College Park airport weather data, respectively. We discuss this in detail in section 3.3. From the figure, we can find that the LCCP results for Basel and Kallax are very small. The reason is that the GEFs of Sweden and Switzerland are very small. Only for the two countries, the annual leakage is the primary factor of the LCCP. For all other countries, annual energy consumption is the main factor affecting the LCCP. Li (2015b) concluded that the SEER rating had a far more significant impact on lowering CO_2e. Nevertheless, based on our study, this conclusion is only valid in the countries with a high GEF.

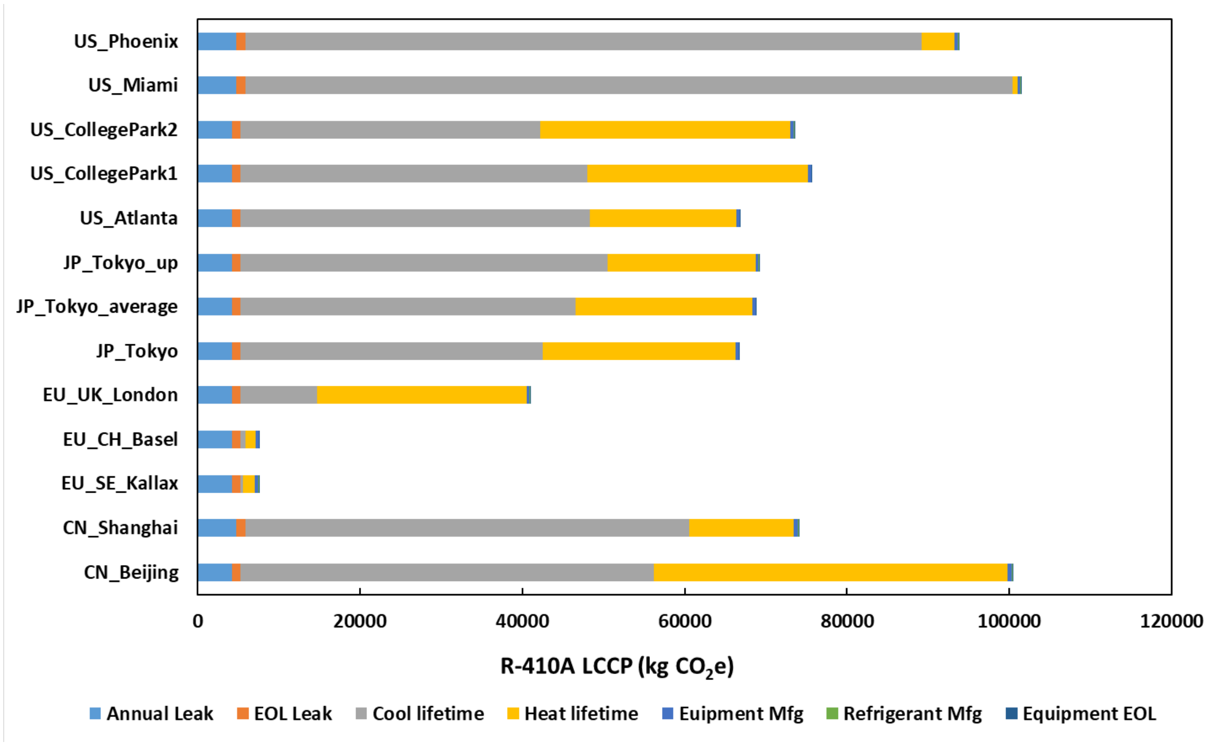

Figure 1-14: LCCP results for R-410A at various cities

1.5.3.2 Different Refrigerants

Figure 1-15 shows the LCCP in four different cities for four different refrigerants. We can see that R-290, R-32, and R-452B are good alternatives for R-410A with lower LCCP. For Kallax, the annual leakage is the major contributor to emissions since its GEF is low. Thus, the LCCP could be decreased by 60% for this city if R-290 substituted R-410A. As for the previous studies, Choi et al. (2017) compared the LCCP of R-290, R-410A, and R-32 for five different cities in Korea. They found that the LCCP of R-410A was 9% higher than that of R-32 and 21% higher than that of R-290 in Seoul. Lee et al. (2016) calculated the LCCPs for R-410A, R-32, R-290, and R-452B, and the results were 126, 119, 111, and 120 MT of CO_2e, respectively. The LCCP order of different refrigerants was consistent with ours.

As for R-466A, the emission from the refrigerant manufacturing process had not been reported until now. Thus, we made three assumptions for the values and studied whether different emissions in the refrigerant manufacture phase would bring some differences in the LCCP calculation. According to the IIR guideline (Life Cycle Climate Performance Working Group, 2015), the emissions from the refrigerant manufacturing process for HFC refrigerants range from 5 to 20 kg of CO_2e per kg. We assumed the value to be 5, 10, and 20 kg of CO_2e per kg for assumptions 1, 2, and 3, respectively. The LCCP calculation results are shown in Table 1-13. The three columns for each assumption are the emissions during the refrigerant manufacturing process, the total LCCP result, and the percentage of the emissions from the refrigerant manufacturing process in the total LCCP. We can find that even for low emission cities like Basel and Kallax, the emissions from the refrigerant manufacturing process are only 3% of the total LCCP. Thus, we concluded that the refrigerant manufacture phase's effect is insignificant in the LCCP calculation. Furthermore, when we compare the LCCP calculation results for R-466A (assumption 1) with other refrigerants' results in Figure 1-16, we can find that the LCCP of R-466A is 1.6% higher than that of R-452B but 8% lower than that of R-410A for College Park, US as an example. Thus, R-466A is also a good substitute for R-410A from the LCCP perspective.

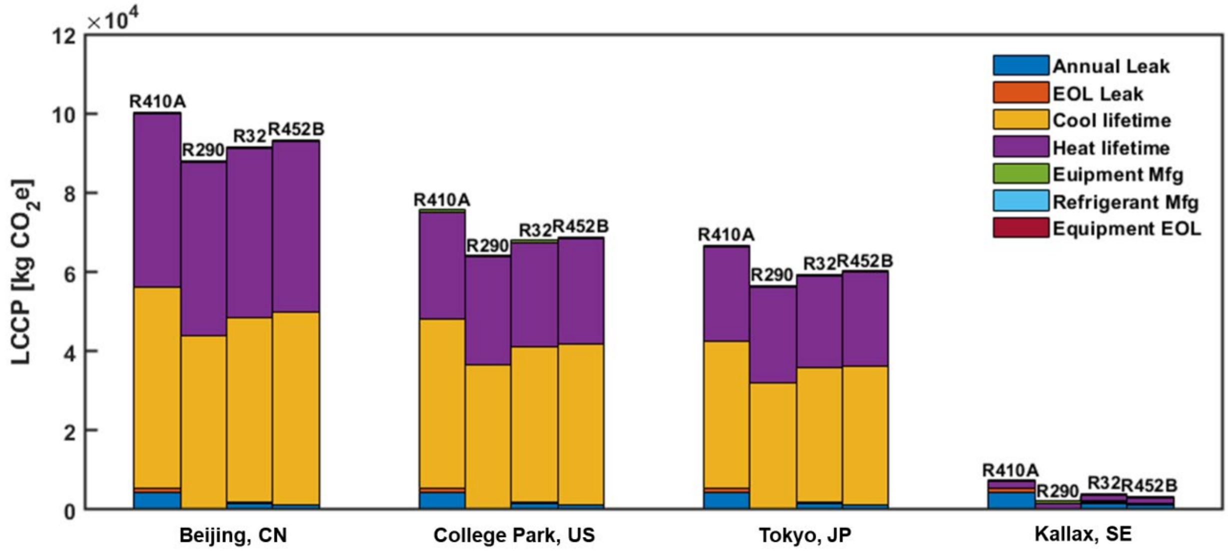

Figure 1-15: Different refrigerant LCCP results for four selected cities

Table 1-13: R-466A Refrigerant Manufacturing Process Emission Effect

(kg CO_2e)	Assumption 1			Assumption 2			Assumption 3		
City	RFM	LCCP	Percentage	RFM	LCCP	Percentage	RFM	LCCP	Percentage
Beijing, CN	31	94164	0.03%	62	94195	0.07%	124	94257	0.13%
Shanghai, CN	31	69482	0.04%	62	69513	0.09%	124	69575	0.18%
Tokyo, JP	31	61524	0.05%	62	61555	0.10%	124	61617	0.20%
Kallax, SE	31	4246	0.73%	62	4277	1.45%	124	4339	2.85%
Basel, CH	31	4314	0.72%	62	4345	1.42%	124	4407	2.81%
London, UK	31	61524	0.05%	62	61555	0.10%	124	61617	0.20%
Atlanta, US	31	61518	0.05%	62	61549	0.10%	124	61610	0.20%
College Park, US	31	70087	0.04%	62	70118	0.09%	124	70180	0.18%
Miami, US	34	98865	0.03%	69	98899	0.07%	138	98968	0.14%
Phoenix, US	34	91276	0.04%	69	91311	0.08%	138	91379	0.15%

1.5.3.3 Weather Data Source and LCCP

We compared the LCCP results using weather data from the UMCP campus field tests and the local airport weather station. Figure 1-16 shows the results. From Figure 1-16, the UMCP campus ambient temperature was always higher than that of the College Park airport. This brings a higher emission in the summer but a lower emission in the winter. Figure 1-16 shows that the decrease in heating is smaller than the increase in cooling. This brings a total increase in the final LCCP result. Compared with the airport data, the LCCP results using the campus data are up 8.1%, 2.4%, 2.8%, and 0.6% for R-452B, R-32, R-410A, and R-290. This result means that using local airport weather data can result in up to an 8% decrease in LCCP If onsite ambient data is not available, a correction on the ambient temperature is recommended.

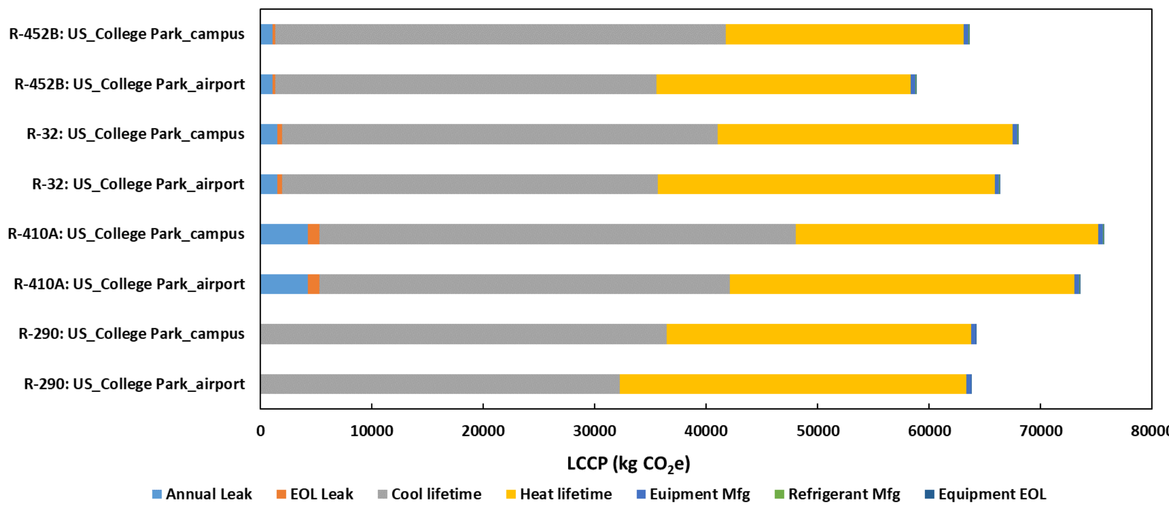

Figure 1-16: Comparison of LCCPs based on weather data from UMCP campus and College Park Airport

1.5.4 Conclusions

A comprehensive LCCP assessment was conducted for a 10.5 kW capacity unitary air conditioner with five refrigerants using various influencing parameters in 11 cities. The conclusions from the study are as follows:

1) The system efficiency has a 10 to 100 times greater impact on the HVAC system's emissions than refrigerant leakage only in higher GEF countries. For lower GEF countries like Sweden and Switzerland, annual leakage is the major factor.
2) The refrigerant manufacturing process, which takes up to 3% of LCCP emissions, is a minor factor compared with emissions from annual energy consumption and annual leakage. While no data was reported on the emissions from the R-466A manufacturing phase, the LCCP can still be estimated by assuming equivalent values to the typical HFC value.
3) R-290, R-32, R-452B, and R-466A are all excellent alternatives for R-410A. The LCCPs of R-32, R-452B, and R-466A are close to each other. The LCCP of R-410A is the highest, while the LCCP of R-290 is the lowest. In the low-GEF countries, the LCCP can be decreased by 60% by substituting R-410A with R-290.
4) The ambient temperature weather data from the UMCP campus field test and College Park airport weather station are different up to 5 °C, possibly due to the UHI effect. This effect can cause up to an 8% difference in LCCP calculation. Thus, researchers are suggested to carefully consider the ambient temperature when conducting LCCP calculations for high-population-density regions. Some correction factors could be needed if the weather station database and local ambient temperature show measurable differences.

1.5.5 Acknowledgment

We gratefully acknowledge the support of the Center for Environmental Energy Engineering (CEEE) at the University of Maryland and Air Solution R&D Laboratory at L.G. Electronics Inc.

1.5.6 Reference

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1.6 Energetics/ORNL Heat Pump Annex 54 Research Project: Milestone 1 Report

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Energetics

1.6.1 Introduction

This report summarizes the activities related to establishing a detailed database of alternative lower Global Warming Potential (GWP) refrigerants for different HVAC&R applications, including thermodynamic and transport properties.

The goal is to identify the most suitable candidates for the replacement of traditional refrigerants on the basis of thermodynamic performance and environmental benefits. It also considers the extent of potential system adjustments required to accommodate the new alternative refrigerants. It should be noted that this analysis is based on thermodynamics only, and a more complex analysis involving targeted equipment will be conducted following this study.

1.6.2 Methods & Assumptions

In this report, we evaluated common HVAC&R applications, including chillers, heat pumps, residential and mobile air conditioning, high-temperature heat pumps, and Low and medium. And high refrigeration applications. Air-to-air, air-to-water, water-to-air, and water-to-water configurations were evaluated when relevant. Table 1-14 provides a summary of the refrigerants and alternatives included in the study.

Table 1-14. Traditional refrigerants and the evaluated alternatives

Baseline Refrigerant	Alternatives			
R-123	R-1233zd(E) R-1224yd(Z)			
R-404A	R-448A R-449A	R-452A R-454A R-454C	R-457A R-457B R-465A	R-449C R-455A R-507A
R-134A	R-516A R-515B R-513A	R-450A R-444A R-1234ze(E)	R-456A R-1234yf	
R-410A	R-454B R-32 R-466A	R-452B R-463A R-468C		
R-245fa	R-1336mzz(Z) R-1233zd(E) R-1234ze(E)			

The thermodynamic properties of the refrigerants were obtained via NIST REFPROP v10. The majority of the refrigerants are included in the REFPROP package, while others were obtained by contacting manufacturers directly.

Assumptions related to the temperatures for the condenser, evaporator, subcooling, superheating, and suction superheating are provided in Table 1-15.

Table 1-15. Assumptions for thermodynamic analysis

Baseline Refrigerant	Application	Evaporator fluid	Condenser fluid	°C				
				Tcond	Tevap	Subcooling	Superheat	Suction Superheat
R410A	A/C	Air	Air	45	7	5	5	5
		Water	Air	45	4	5	5	5
		Air	Water	30	7	5	5	5
		Water	Water	40	4	5	5	5
	Heat Pump	Air	Air	55	5	5	5	5
		Water	Air	55	20	5	5	5
		Air	Water	50	5	5	5	5
		Water	Water	50	20	5	5	5
R123	Chiller	Water	Water	40	7	5	5	5
R134a	Chiller	Water	Water	40	7	5	5	5
		Water	Air	45	7	5	5	5
	Mobile A/C	Air	Air	55	4	5	4	5
	HT Refrig.	Air	Air	55	0	10	5	30
	MT Refrig.	Air	Air	55	-10	10	5	40
	LT Refrig.	Air	Air	55	-25	10	5	55
R245fa	High T HP	Air	Air	120	20	5	10	5
R404A	HT Refrig.	Air	Air	55	0	5	5	13.3
	MT Refrig.	Air	Air	55	-10	5	5	23.3
	LT Refrig.	Air	Air	55	-25	5	5	38.3

1.6.3 Thermodynamic Analysis Results

Analyses were broken into the categories indicated in Table 1-15. By refrigerant, they were as follows:

- R-123: Chillers
- R-134a: Chillers, varying mediums (W2A, W2W)
- R-134a: Mobile Air Conditioning
- R-134a: Refrigeration, varying source temperatures (low, medium, high)
- R-245fa: High temperature heat pumps
- R-404A: Refrigeration, varying source temperatures (low, medium, high)
- R-410A: Heat Pumps, varying mediums (A2A, A2W, W2A, W2W)
- R-410A: Building Air Conditioning, varying mediums (A2A, A2W, W2A, W2W)

For each of the listed categories, the following plots were generated:

- Pressure vs. specific enthalpy (P-h) and refrigeration cycle plots

- Relative coefficient of performance (COP vs Baseline COP) and volumetric capacity (Q_v vs baseline Q_v) graphs
- GWP vs. boiling point plots, including indication of ASHRAE designation - *for the refrigerant, agnostic of application or medium*
- Pressure vs. inverse temperature (P vs. -1/T) - *for the refrigerant, agnostic of application*
- Volumetric cooling capacity (calculated using the suction density condition) vs. evaporating temperature – *for the refrigerant, agnostic of application or medium*

1.6.3.1 R-123: Chillers

R-123 was evaluated for chillers along with R-1224yd(Z) and R-1233zd(E). The GWP of R-123 is 77, while the GWPs of the alternatives are both 1. The transition does not sacrifice safety, as both alternatives are classified as A1 (Figure 1-17).

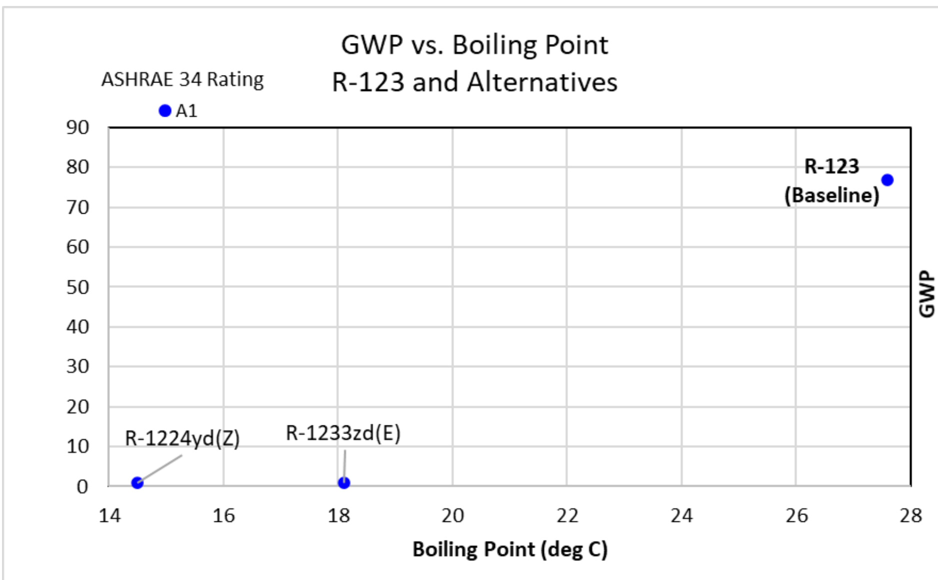

Figure 1-17. GWP vs. Boiling Point for R-123 and evaluated alternatives, including ASHRAE 34 classification

The P vs. -1/T graph (Figure 1-18) demonstrates that the alternatives have higher pressures than R-123 across the evaluated temperature range. Their performances are more similar to each other than they are to R-123.

Figure 1-18. Pressure vs. -1/T for R-123 and alternatives

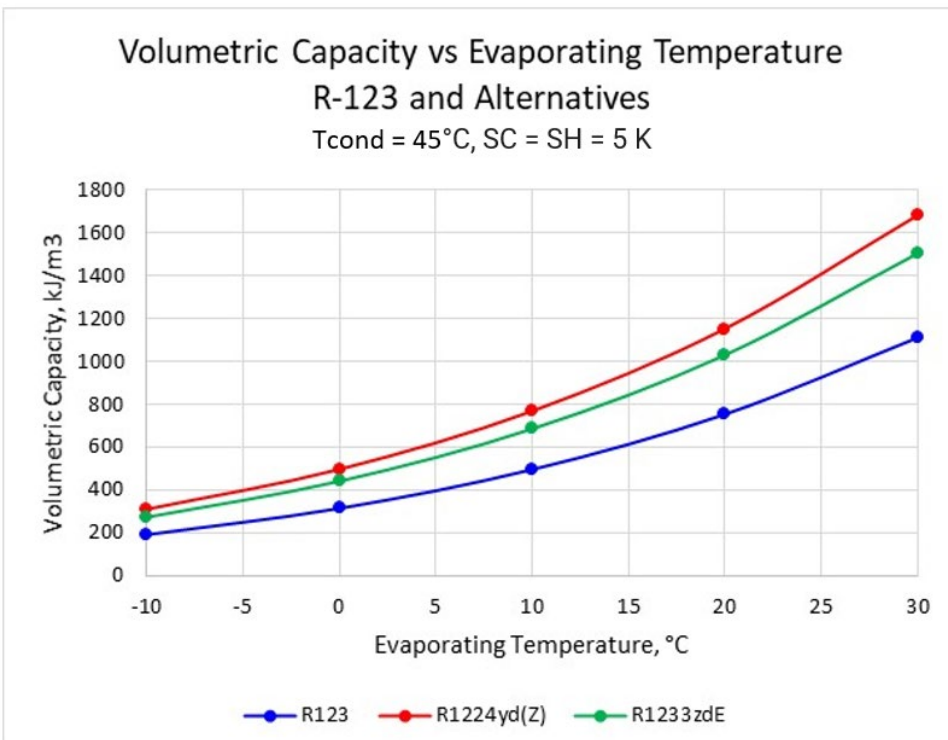

Figure 1-19. Volumetric capacity (calculated using the suction density condition) vs. evaporating temperature for R-123 and alternatives. Note that the condensing temperature differs from that given in the assumption summary (Table 1-15). The general trend can still be captured regardless of the difference in system conditions.

For R-123, water-to-water chillers were examined on a thermodynamic basis. The cycle diagrams (Figure 1-20) align with the P vs. -1/T diagram in that they show that the operating pressures of the evaporator and condenser for the alternatives will need to be higher than those required for R-123. This may result in using smaller compressor. Between the higher operating temperature – resulting in thicker tubes – and smaller compressors, the chiller initial cost may be similar to the baseline refrigerant.

The refrigerant cycles also demonstrate differences in capacity, but it is important to the consider the densities of these substances in order to examine capacity on a volumetric basis. The volumetric capacity can be observed in Figure 1-19, as well as the comparison in Figure 1-21.

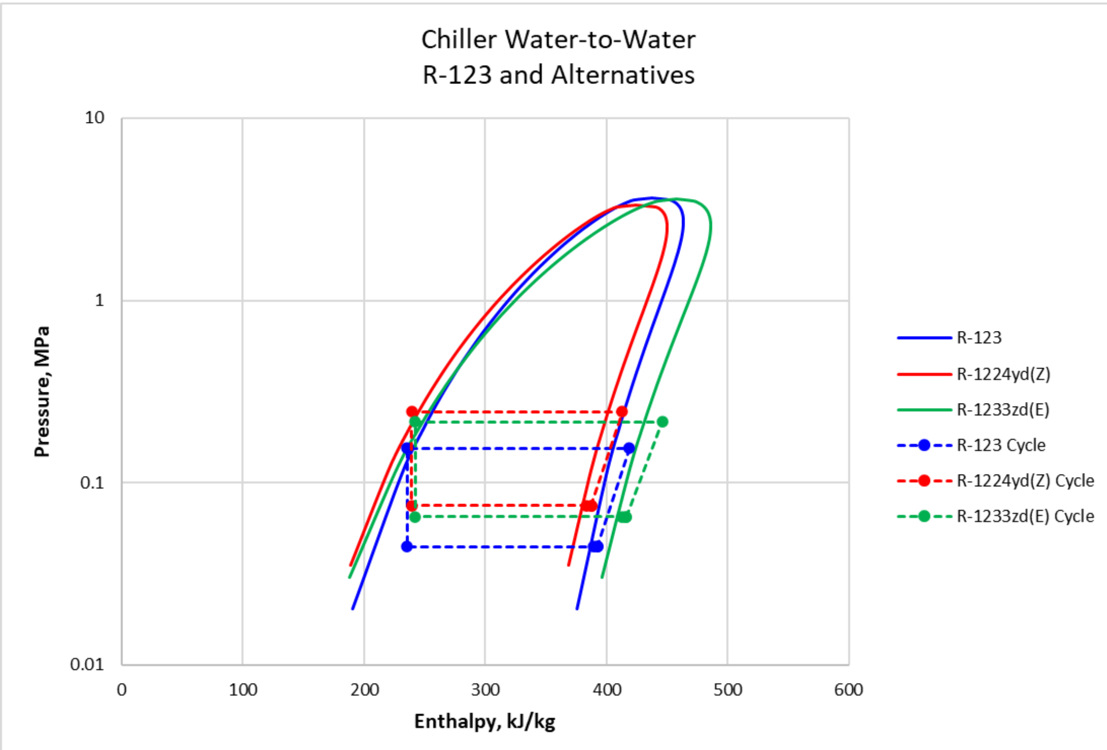

Figure 1-20. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-123 and alternatives for water-to-water chillers

From a design standpoint, these two figures demonstrate that the compressors for R-1224yd(Z) and R-1233zd(E) could be downsized. While volumetric capacity is improved for both alternatives, COP remains very similar. Overall, R-1224yd(Z) and R-1233zd(E) are great candidates in terms of performance and potential for equipment savings, as well as significant reduction in GWP.

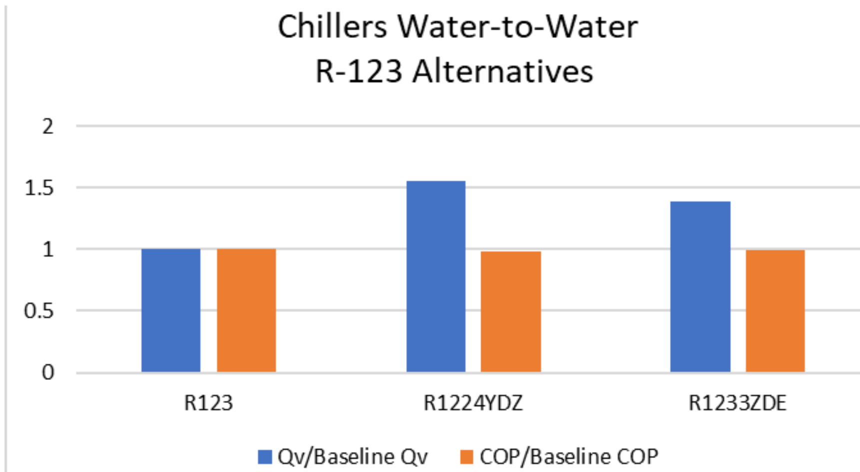

Figure 1-21. Relative volumetric capacity (Q_v) and coefficient of performance (COP) of R-123 alternatives compared to the baseline for water-to-water chillers

1.6.3.2 R-134a: General Comparison

All R-134a alternatives result in a significant reduction in GWP, as shown in Figure 1-22. R-456A, the highest GWP refrigerant of all the alternatives, has a GWP less than half that of R-134a. The A1 refrigerants, R-450A, R-456A, R-513A, and R-515B, all have GWPs under 700. The under-200 blends are all classified as A2L (mildly flammable). R-515B and R-1234ze(E) may also be used as low-pressure alternatives due to their relatively high boiling points.

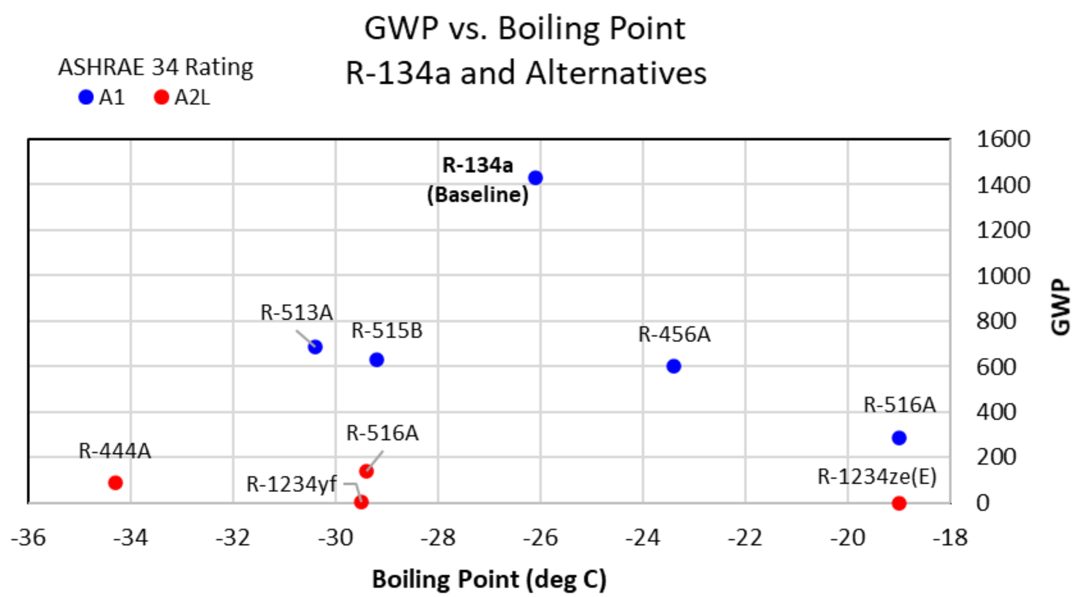

Figure 1-22. GWP vs. Boiling Point for R-134a and evaluated alternatives, including ASHRAE 34 classification

Based on the pressure vs. -1/T plot (Figure 1-23), there is wide variation in the pressures of the alternative refrigerants. Closer analysis can be carried out through review of the cycle diagrams in the following section.

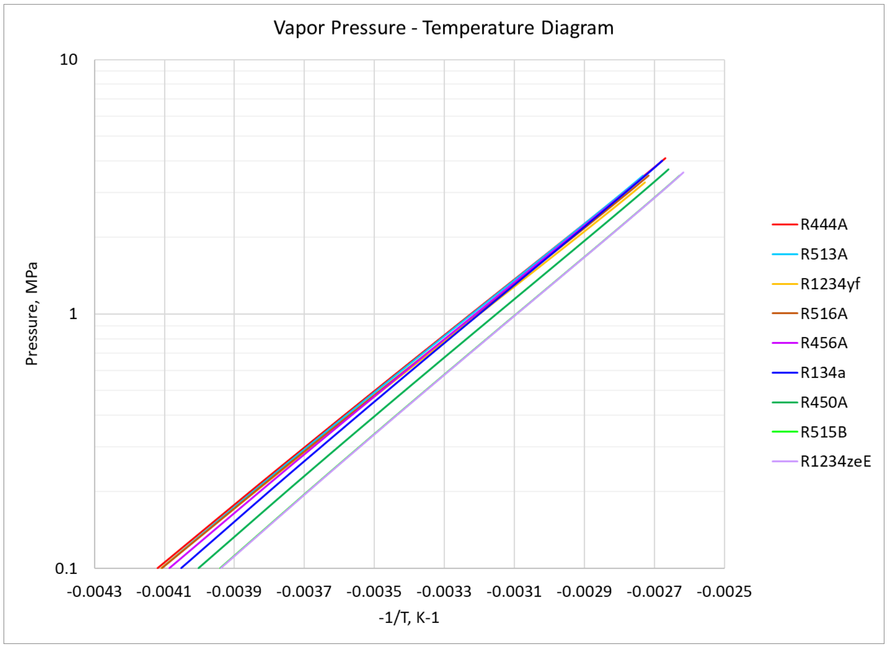

Figure 1-23. Pressure vs. -1/T for R-134a and alternatives. Note that the line for R-515B is nearly covered by that of R-1234ze(E). The legend is ordered based on the order in which the lines cross the horizontal axis, from left to right.

Generally, volumetric capacities match within 6% that of R-134a for R-456A, R-513A, R-516A, and R-1234yf, up to an evaporating temperature of 20°C. Clear reductions from the baseline can be observed in R-444A, R-450A, R-515B, and R-1234ze(E), especially as evaporating temperature increases (Figure 1-24). These trends can be more closely examined based on medium type in the next section.

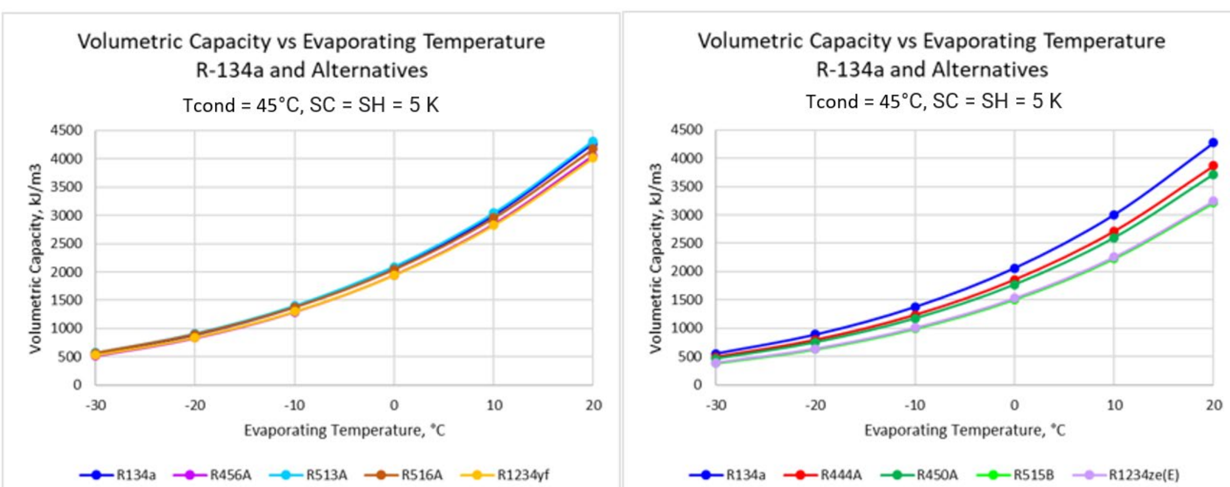

Figure 1-24. Volumetric capacity (calculated using the suction density condition) vs. evaporating temperature for R-134a and alternatives. Note that this analysis was conducted for only one of the condensing temperatures given in the assumption summary (Table 1-15). The general trend can be captured at any of the condensing temperatures.

1.6.3.3 R-134a: Chillers and Mobile Air Conditioning

R-134a can be used for air-to-water and water-to-water chillers, as well as mobile air conditioning. Based on the P-h cycle diagram for air-to-water chillers (Figure 1-25), several of the cycles for alternatives are very similar to that of R-134a, including the required operating pressures for the equipment and the enthalpy change across the cycle. The key differences are that R-450A, R-515B, and R-1234ze(E). They resulted in lower pressures for both the evaporator and condenser, which can also be observed in Figure 1-23. These lower operating pressures could potentially enable the use of less costly equipment. Note that the cycles for R-1234ze(E) and R-515B are nearly identical, due to R-515B's composition of 91.1%wt R-1234ze(E).

In the case of water-to-water chillers, the P-h cycle diagram is only subtly different from that of air-to-water chillers. There is slightly greater variation in the evaporator operating pressures of the alternatives, and the condenser operating pressures are lower by around 10% (Figure 1-26). The condenser pressures become higher for the mobile air conditioning application, while evaporator pressures slightly drop compared to air-to-water chillers (Figure 1-27).

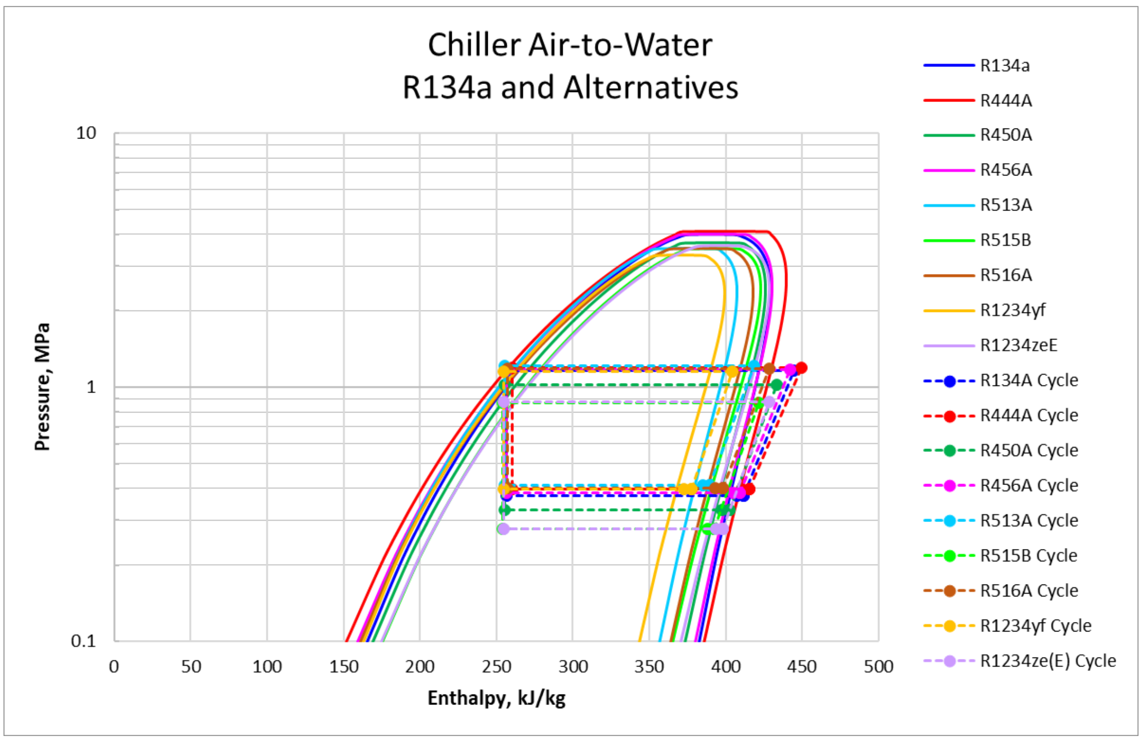

Figure 1-25. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for air-to-water chillers

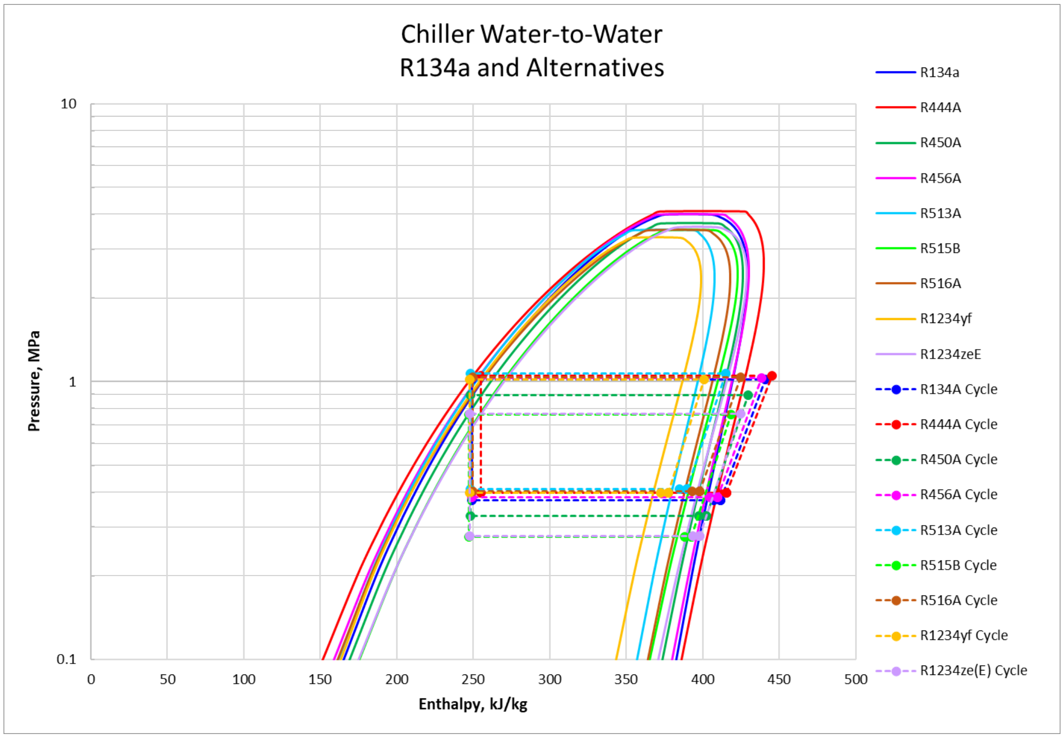

Figure 1-26. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for water-to-water chillers

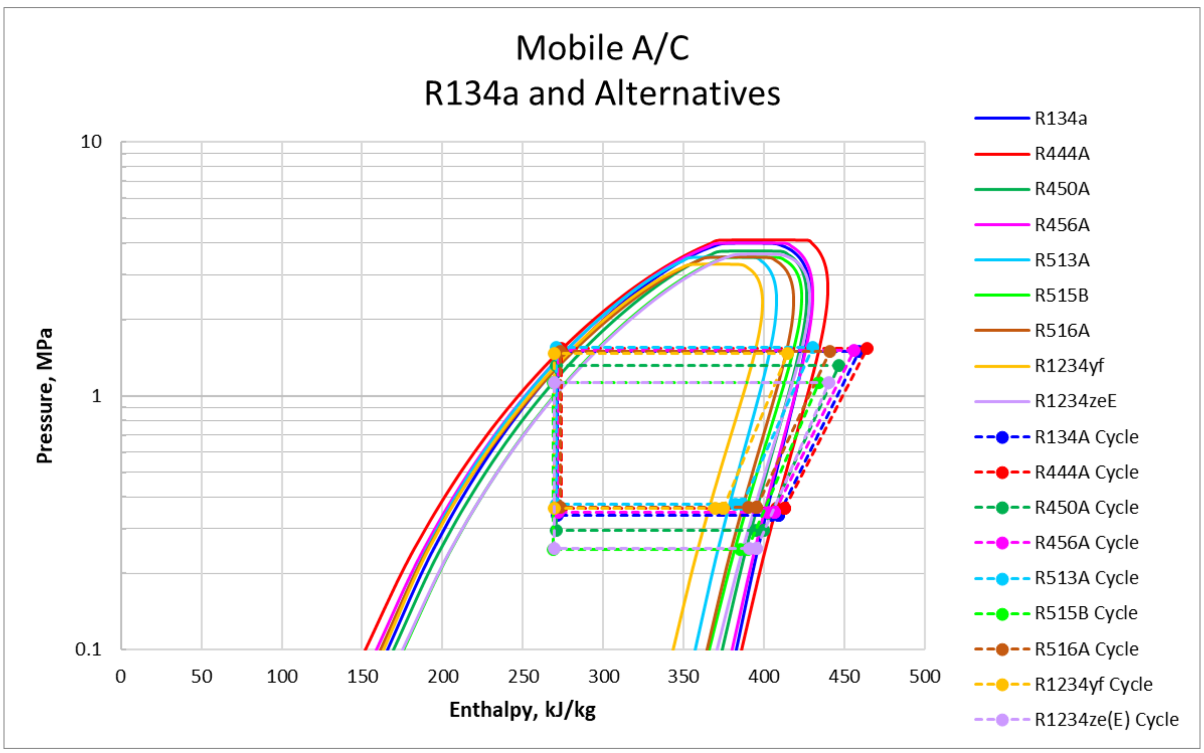

Figure 1-27. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for mobile air conditioning

Comparing COP and volumetric capacity of alternative refrigerants to R-134a, the results are, again, very similar for the air-to-water and water-to-water configurations of chillers. R-444A, R-456A, R-513A, and R-516A all have relative volumetric capacities within 2% of the baseline for both configurations (Figure 1-28 and Figure 1-29). Some alternatives perform well in terms of COP but poorly when it comes to volumetric capacity. For instance, R-1234ze(E) does not have a COP reduction compared to R-134a, but its volumetric capacity is only 75% of R-134a's for both configurations. R-450A and R-515B also do not have COP reductions, but their volumetric capacities are 87% and 75% of the baseline, respectively. Due to their relatively low volumetric capacities, R-1234ze(E), R-450A, and R-515B may require larger compressors. To minimize reductions in both COP and volumetric capacity, R-456A would be the best alternative for air-to-water and water-to-water chillers. However, it is important to realize that refrigerant selection may take into account many other factors, including GWP, ASHRAE 34 classification, and other physical properties.

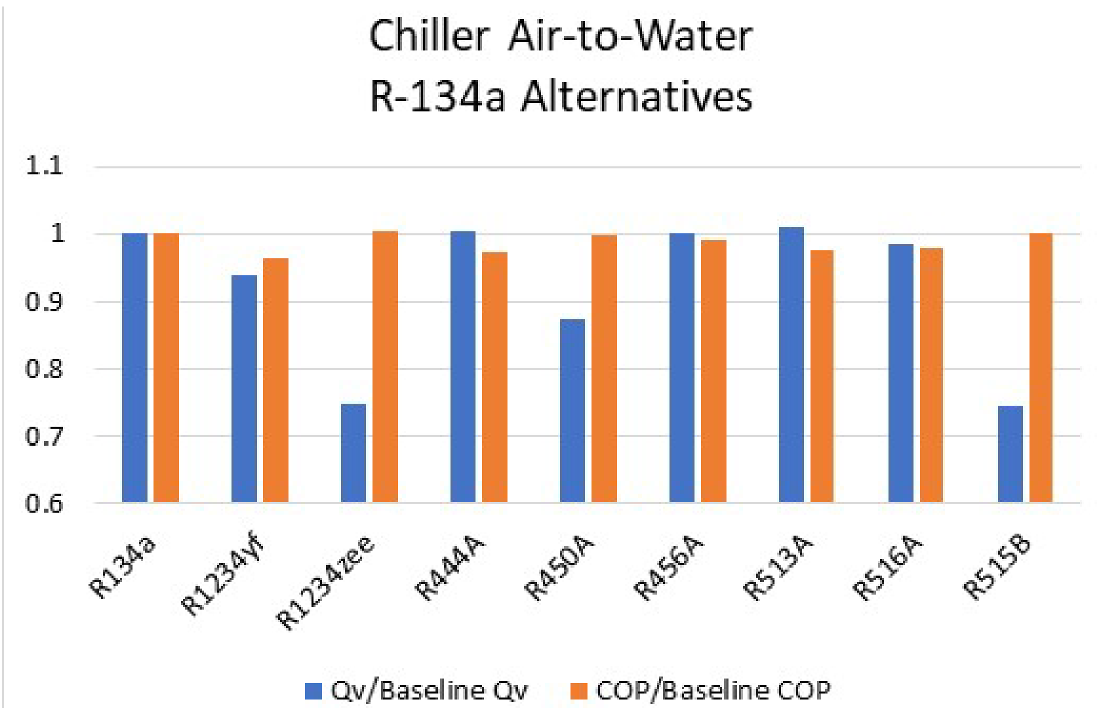

Figure 1-28. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for air-to-water chillers

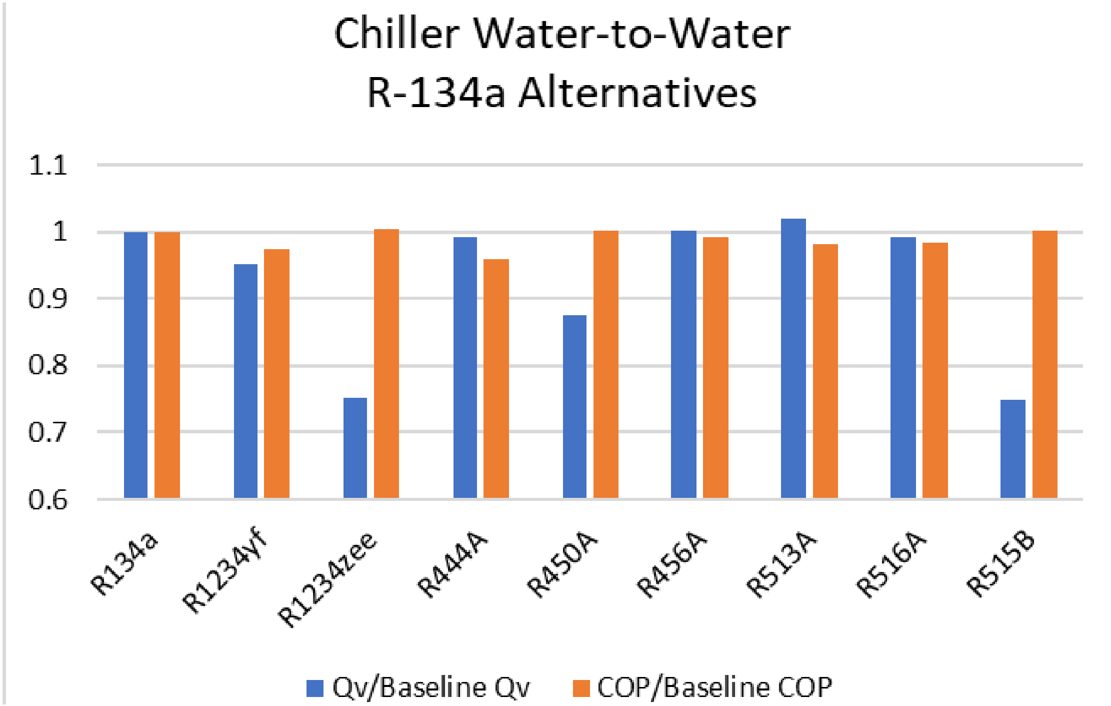

Figure 1-29. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for water-to-water chillers

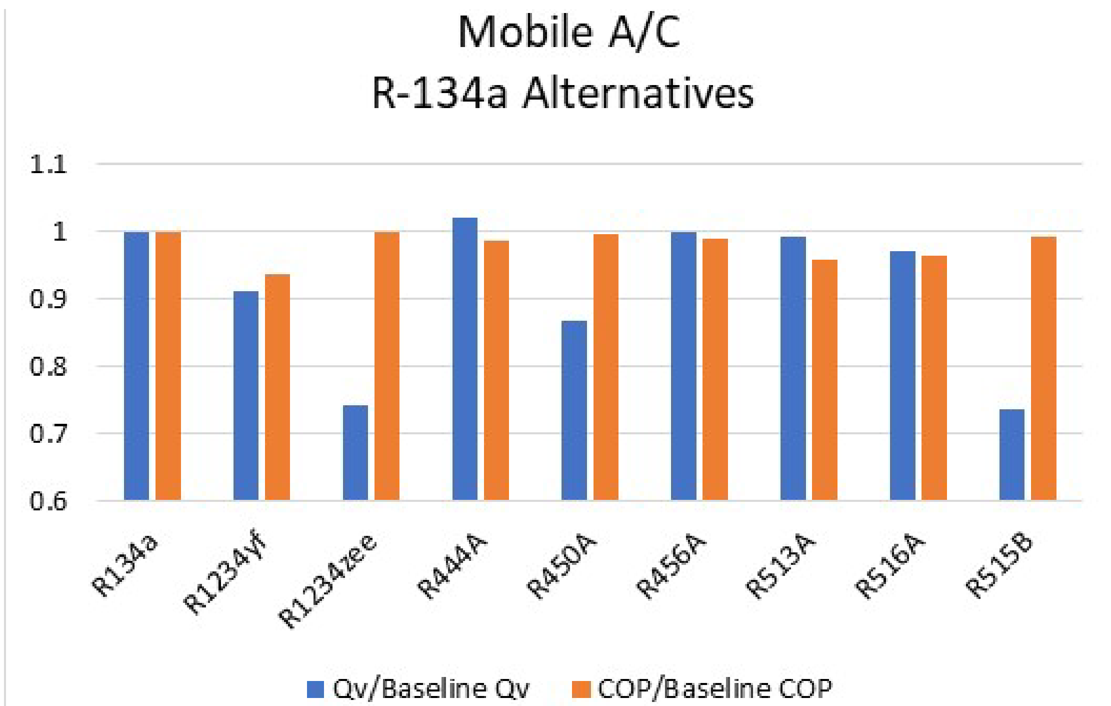

Figure 1-30. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for mobile air conditioning

The COP and volumetric capacity (Q_v) comparisons for mobile air conditioning are roughly the same as they are for chillers. The only notable differences are that R-1234yf sees a small reduction in relative COP and Q_v compared to chillers and that R-444A slightly outperforms R-456A for mobile A/C (Figure 1-30).

1.6.3.4 R-134a: Refrigeration

R-134a is also used for low, medium, and high-temperature refrigeration. For low-temperature refrigeration, R-134a and its alternatives have a wide gap between the operating pressures of evaporators and condensers (Figure 1-31). The gap becomes smaller as pressure increases (Figure 1-32 and Figure 1-33). In all cases, the R-1234yf cycle has the smallest enthalpy change, and R-444A has the largest. The R-456A cycle is the most similar to that of R-134a across all temperature ranges. From low to high temperature, the enthalpy increases of the cycles goes down by about 50 kJ/kg.

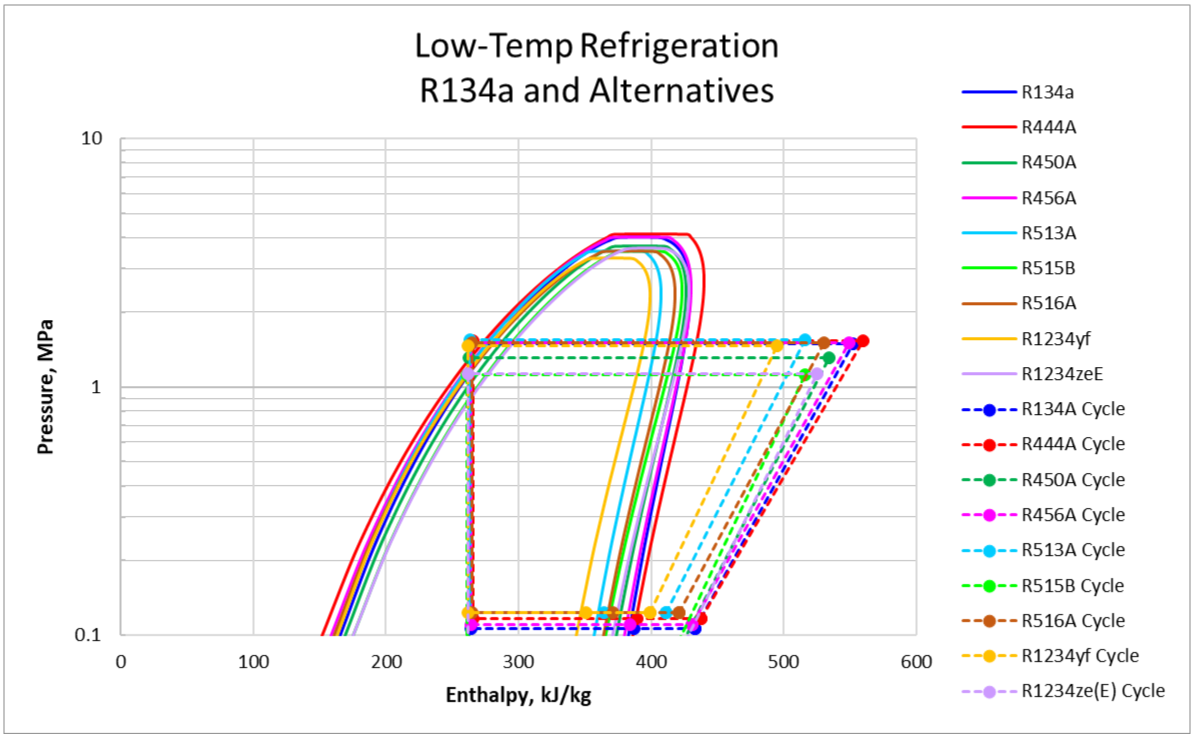

Figure 1-31. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for low-temperature refrigeration

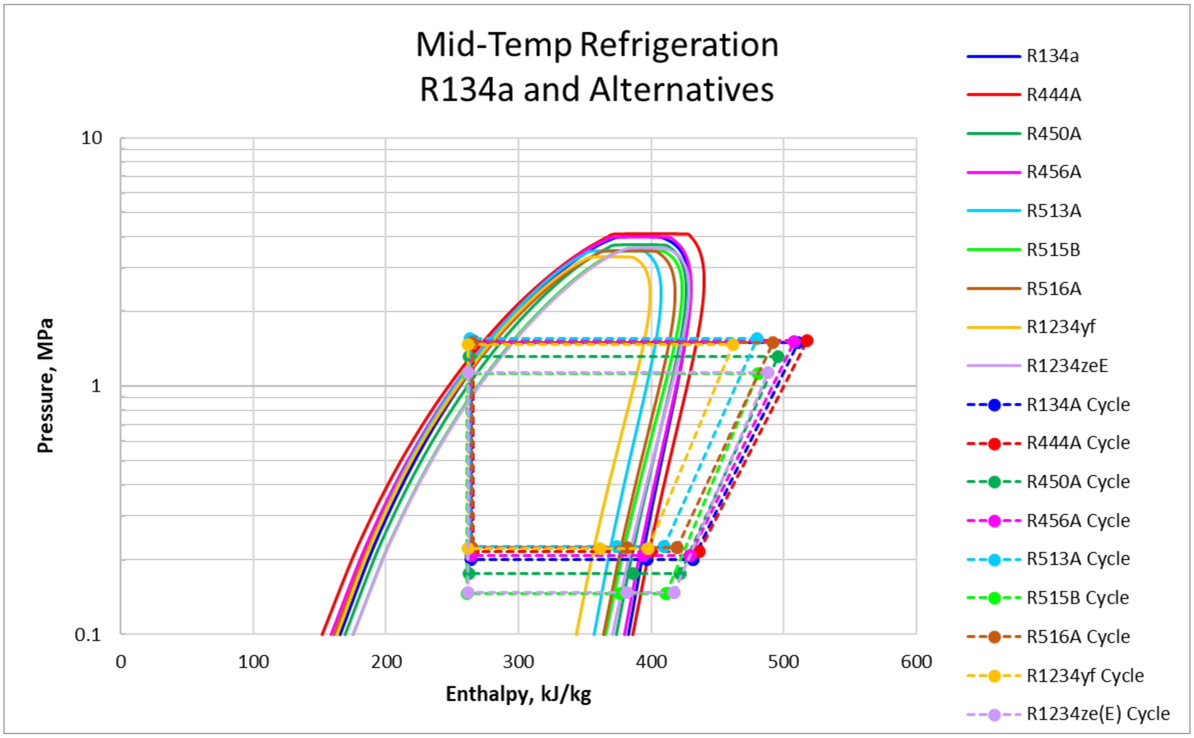

Figure 1-32. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for mid-temperature refrigeration

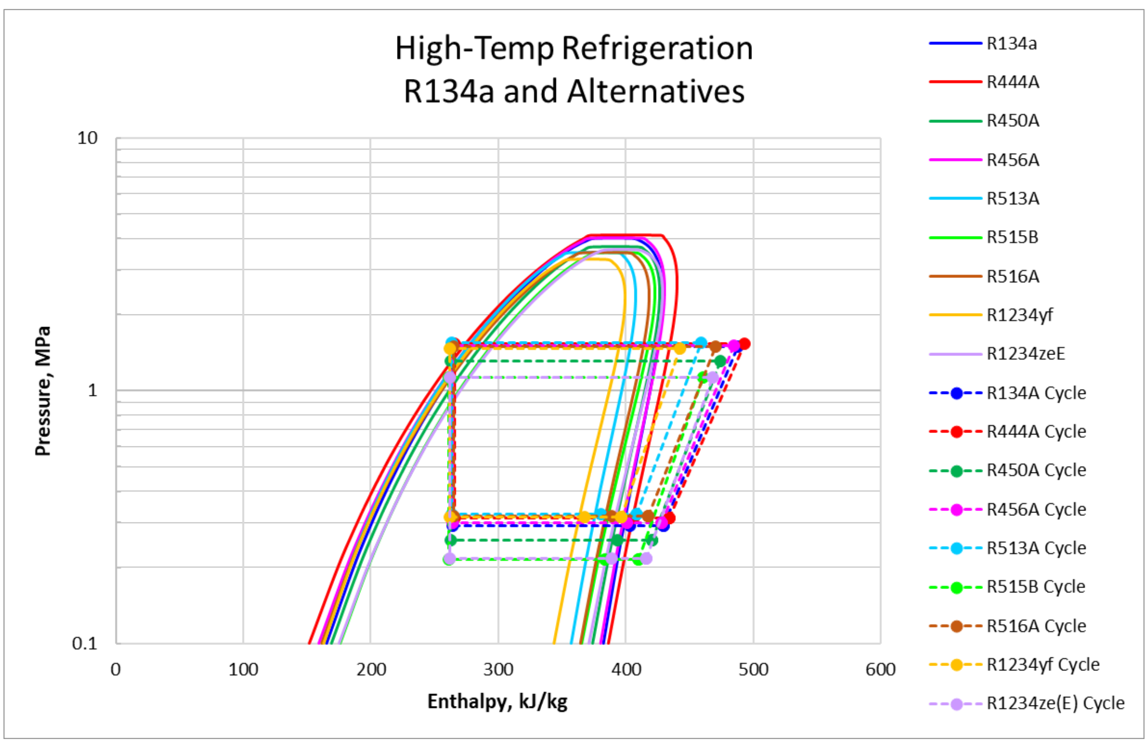

Figure 1-33. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for high-temperature refrigeration

The relative COPs of R-134a alternatives for refrigeration are generally lower for the low-temp application than the mid- and high-temp applications, although COPs are overall comparable to the baseline for all temperature ranges (Figure 1-34 to Figure 1-36). For volumetric capacity (Q_v), there is more variation across the alternatives. While several refrigerants have sufficient values for Q_v, R-1234ze(E) and R-515B are both around 70% of the baseline for low-temperature applications and around 74% of the baseline for high-temperature applications. R-450A also performs rather poorly in volumetric capacity, with a value at 85%, 86%, and 87% of baseline for low-, mid-, and high-temperature applications, respectively. The compressors for refrigerants with low relative Q_v would need to be up-sized. Overall, these relative performances are similar to those of chillers and mobile A/C for R-134a. Based on this data, to achieve a COP at least 98% of that of R-134a for refrigeration, R-1234ze(E), R-444A, R-450A, and R-456A are potential candidates for all temperatures, and R-515B is a good candidate for medium and high temperatures.

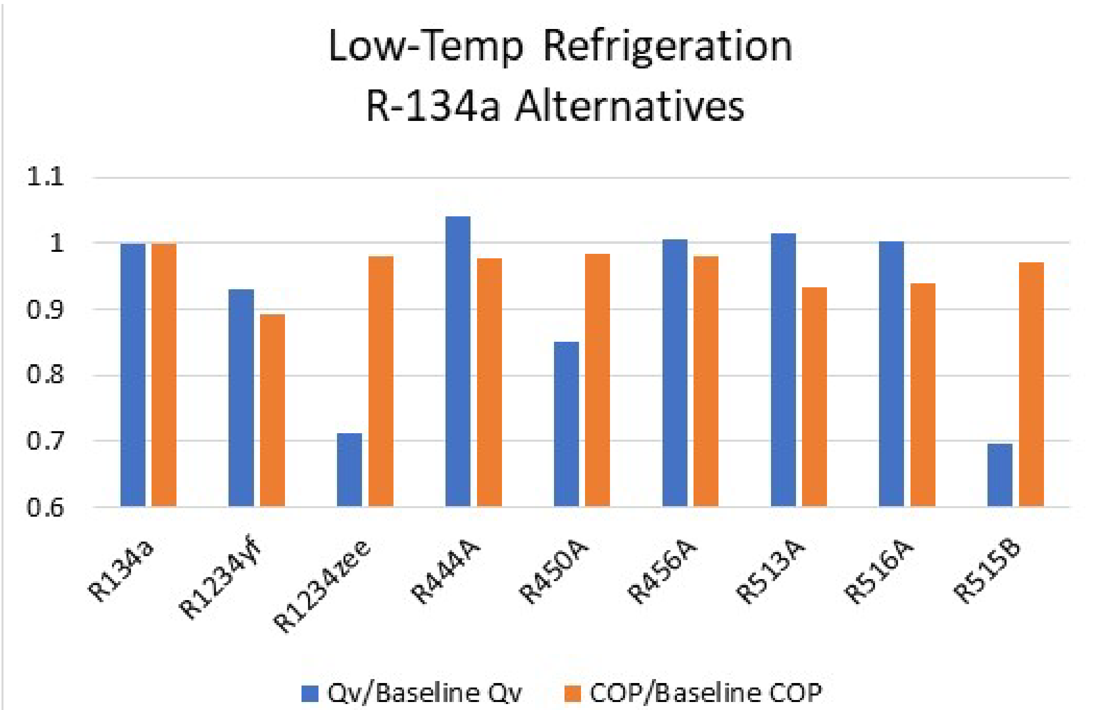

Figure 1-34. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for low-temperature refrigeration

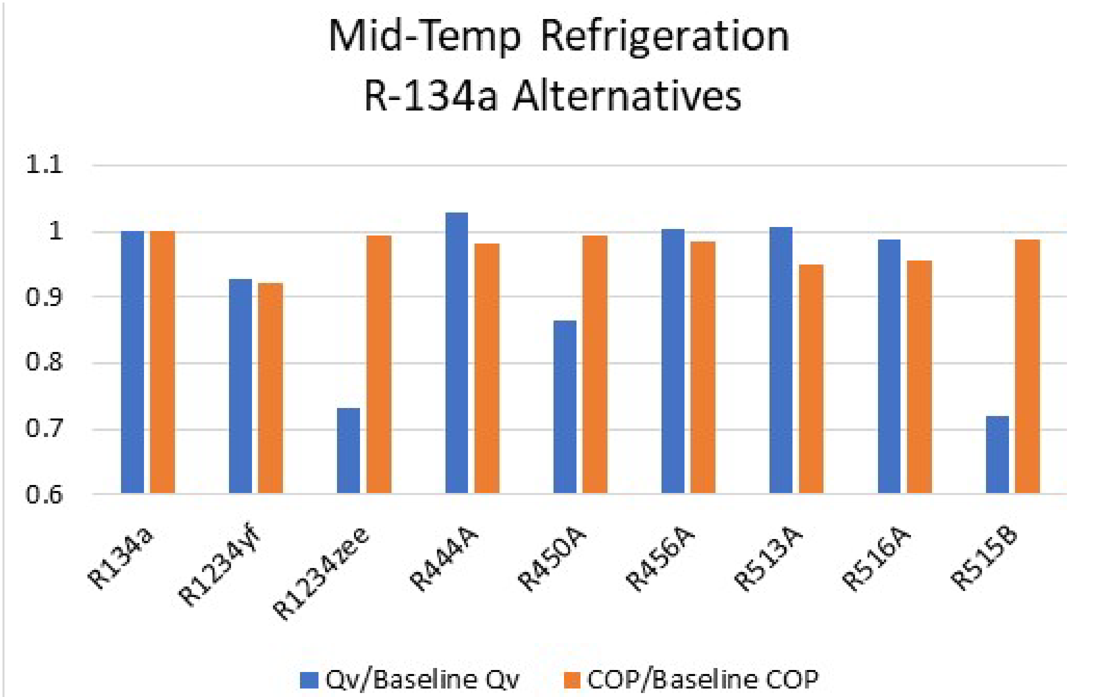

Figure 1-35. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for mid-temperature refrigeration

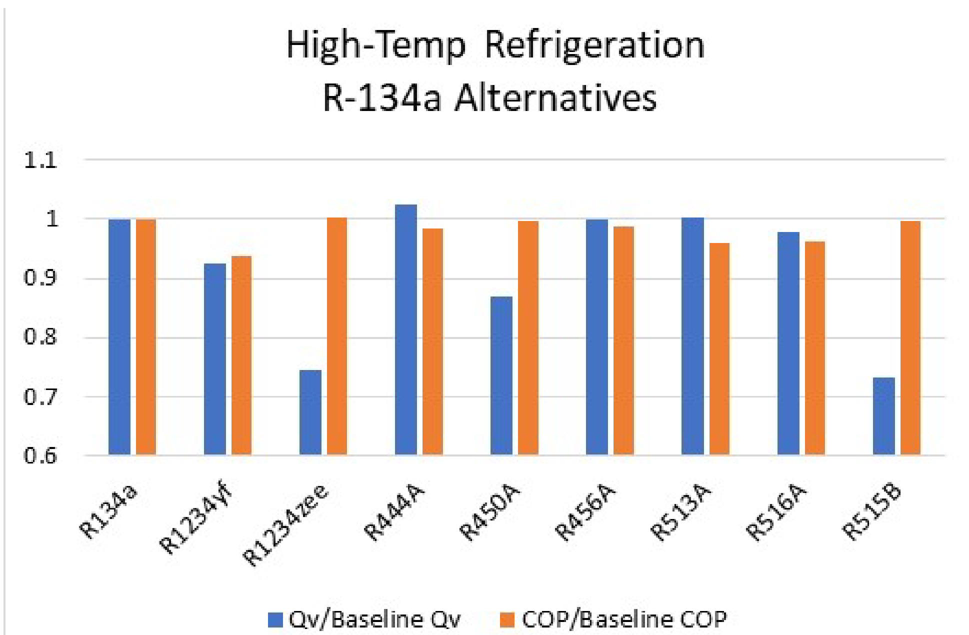

Figure 1-36. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for high-temperature refrigeration

1.6.3.5 R-245fa: Heat Pumps

The alternative mixtures for R-245fa for use in high-temperature heat pumps yield significant reductions in GWP, as well as improvements in safety (Figure 1-37). The GWP of R-245fa is 1030, while R-1336mzz(Z) has a GWP of 2 and R-1233zd(E) and R-1234ze(Z) both have GWPs of 1. Both R-1233zd(E) and R-1336mzz(Z) are designated as non-toxic (class A) compared with R-245fa being classified as toxic (class B). None of the alternatives are exact matches in terms of pressure, but all can be used in high-temperature heat pumps.

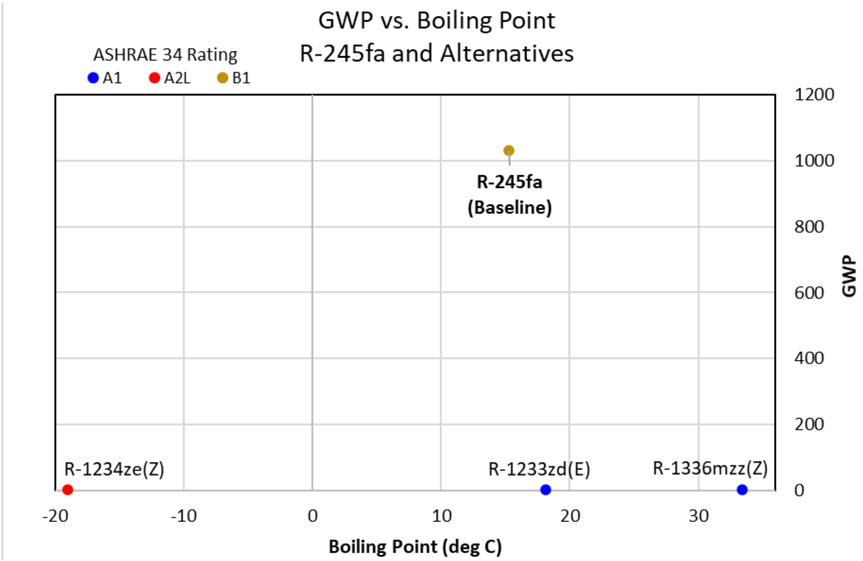

Figure 1-37. GWP vs. Boiling Point for R-245fa and evaluated alternatives, including ASHRAE 34 classification

The P vs. -1/T graph (Figure 1-38) indicates a similar performance of R-1234ze(Z) compared to the baseline at high temperatures. It also shows that R-1336mzz(Z) is the most dissimilar to R-245fa. With its relatively low pressures, it will likely require a much larger compressor than what is required by R-245fa.

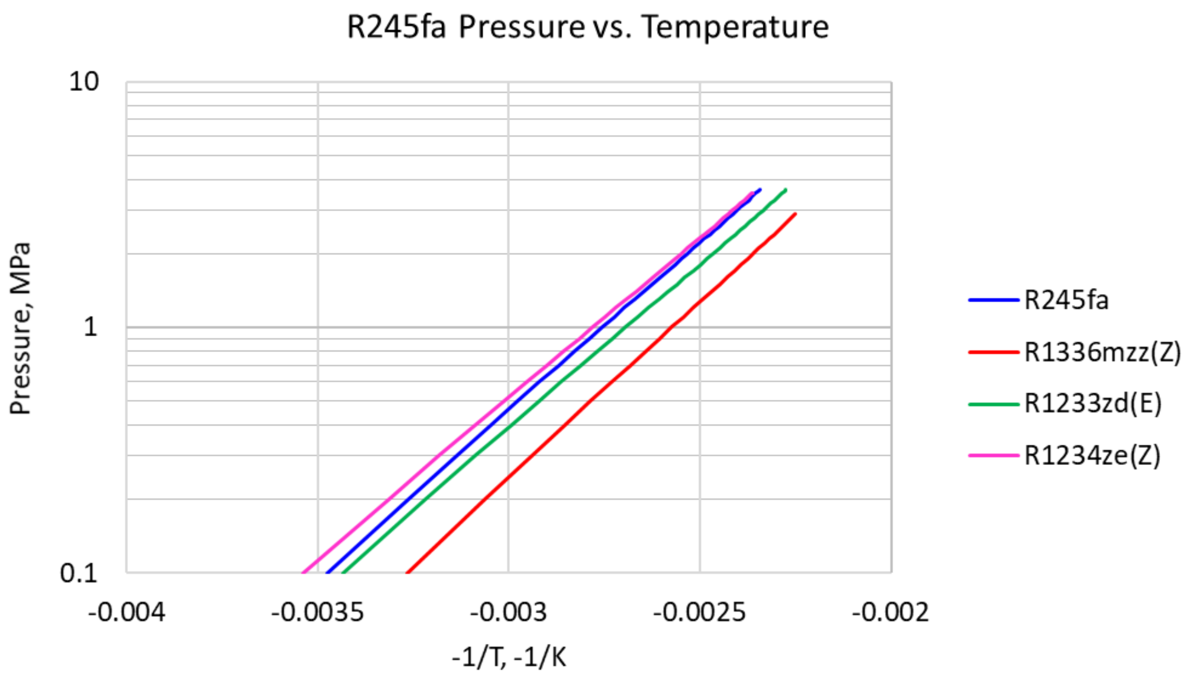

Figure 1-38. Pressure vs. -1/T for R-245fa and alternatives

R-1336mzz(Z) in a high-temperature heat pump would require an evaporator that runs at a lower operating pressure than R-245fa, due to its relatively high boiling point (Figure 1-37 and Figure 1-38). R-1233zd(E) would need the same, and both of these refrigerants would also result in lower operating pressures for the condenser. For R-1234ze(Z), the condenser operating pressure would be relatively the same, and the evaporator pressure would be slightly higher compared with R-245fa.

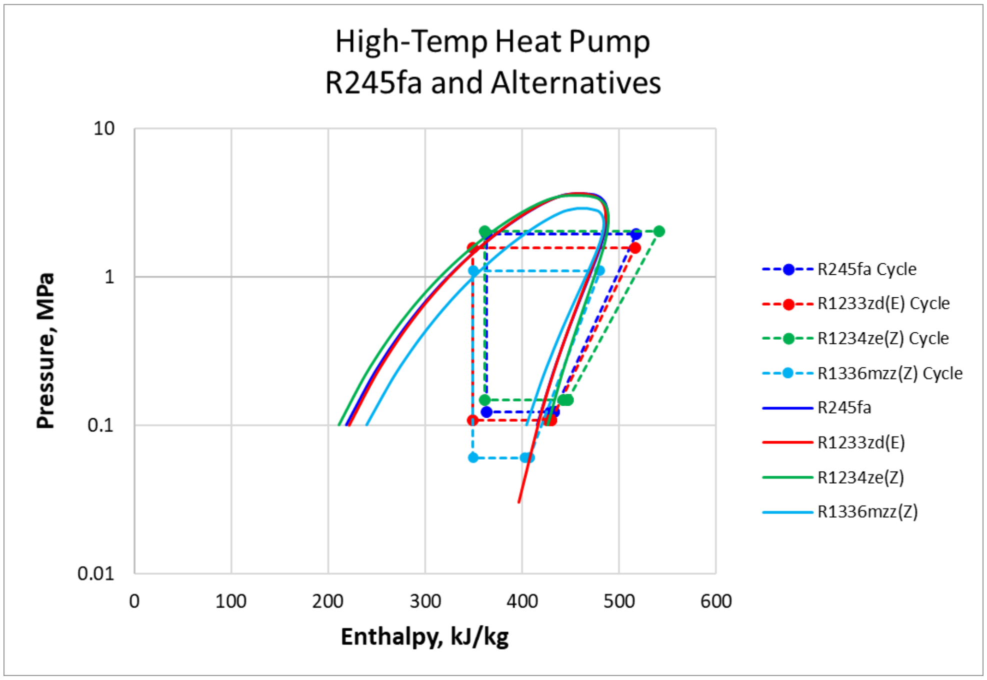

Figure 1-39. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-245fa and alternatives for high-temperature heat pumps

R-1233zd(E) and R-1234ze(Z) have COPs exceeding that of the baseline by about 6% and 5%, respectively (Figure 1-40). The R-1336mzz(Z) COP is about 99% of the baseline, which is a good performance indicator, but its volumetric capacity is significantly reduced to 50% of the baseline. This would require a major up-sizing of the compressor, as also indicated by the P vs. -1/T graph. R-1233zd(E) may need a small increase in compressor size, depending on the original size used, because it has a volumetric capacity (Q_v) at 93% of the baseline. The best-performing alternative in terms of Q_V and COP is R-1234ze(Z). Its Q_V is 22% greater than the baseline, so the compressor size can be reduced, and its COP is higher by about 5%. One tradeoff between this and the other alternatives is its lower safety rating.

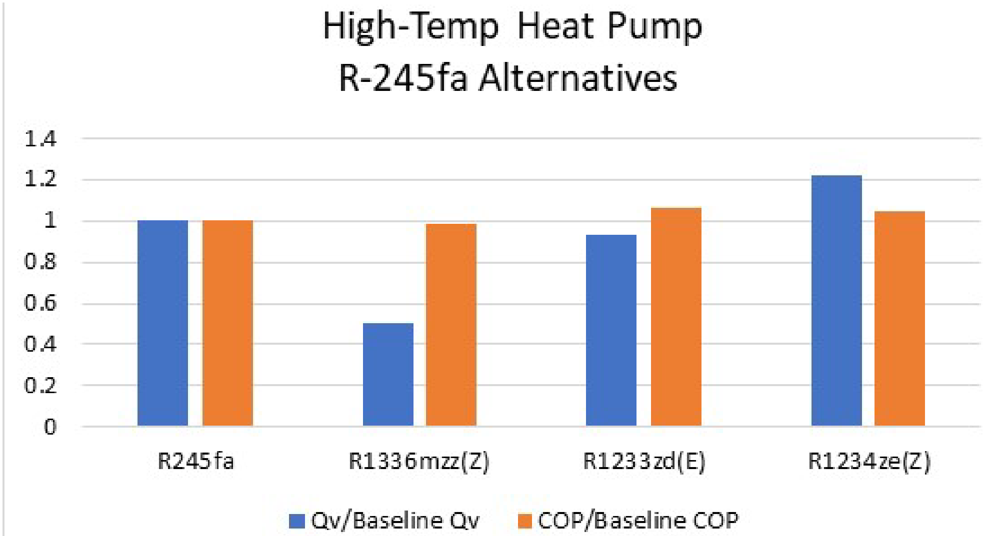

Figure 1-40. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-245fa alternatives compared to the baseline for high-temperature heat pumps

1.6.4 R-404A: Refrigeration

R-404A was evaluated against several alternatives for refrigeration. One of the alternatives, R-507A, has a higher GWP than the baseline, while the rest have lower GWPs (Figure 1-41). Although the GWPs of R-452A, R-449A, and R-448A show great improvement compared to R-404A, they are still relatively high considering long-term goals. R-465A, R-455A, R-454A, R-457B, R-454C, and R-457A are all aligned with long-term goals, but at the risk of higher flammability. R-454C has a similar boiling point as R-448A and R-449A, but there is some variance in their cycles, as shown by the cycle diagrams below (Figure 1-44-Figure 1-46).

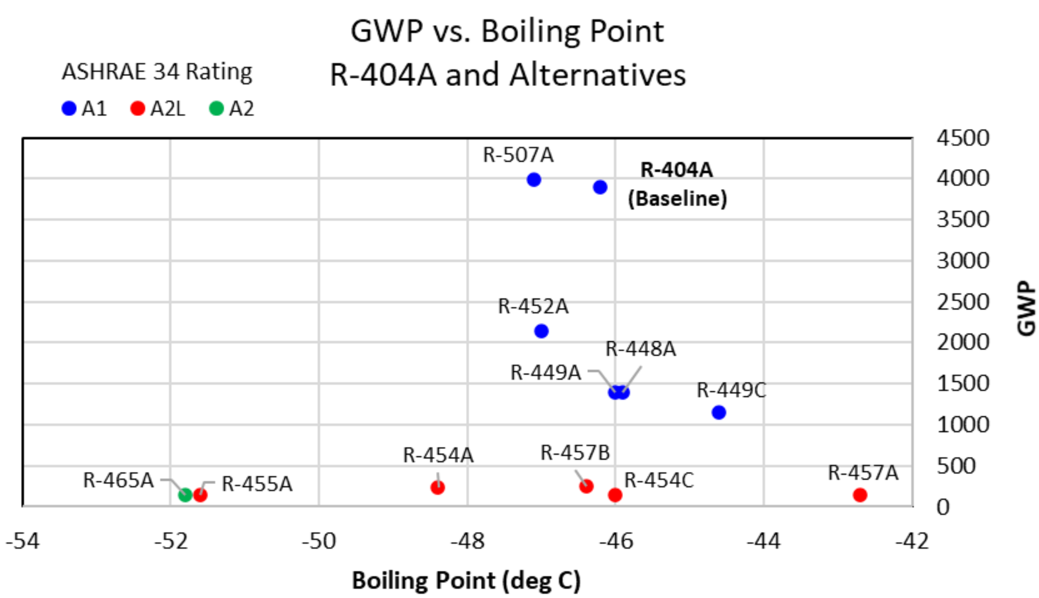

Figure 1-41. GWP vs. Boiling Point for R-404A and evaluated alternatives, including ASHRAE 34 classification

Based on the P vs -1/T plot (Figure 1-44), R-457A is the most dissimilar from R-404A thermodynamically.

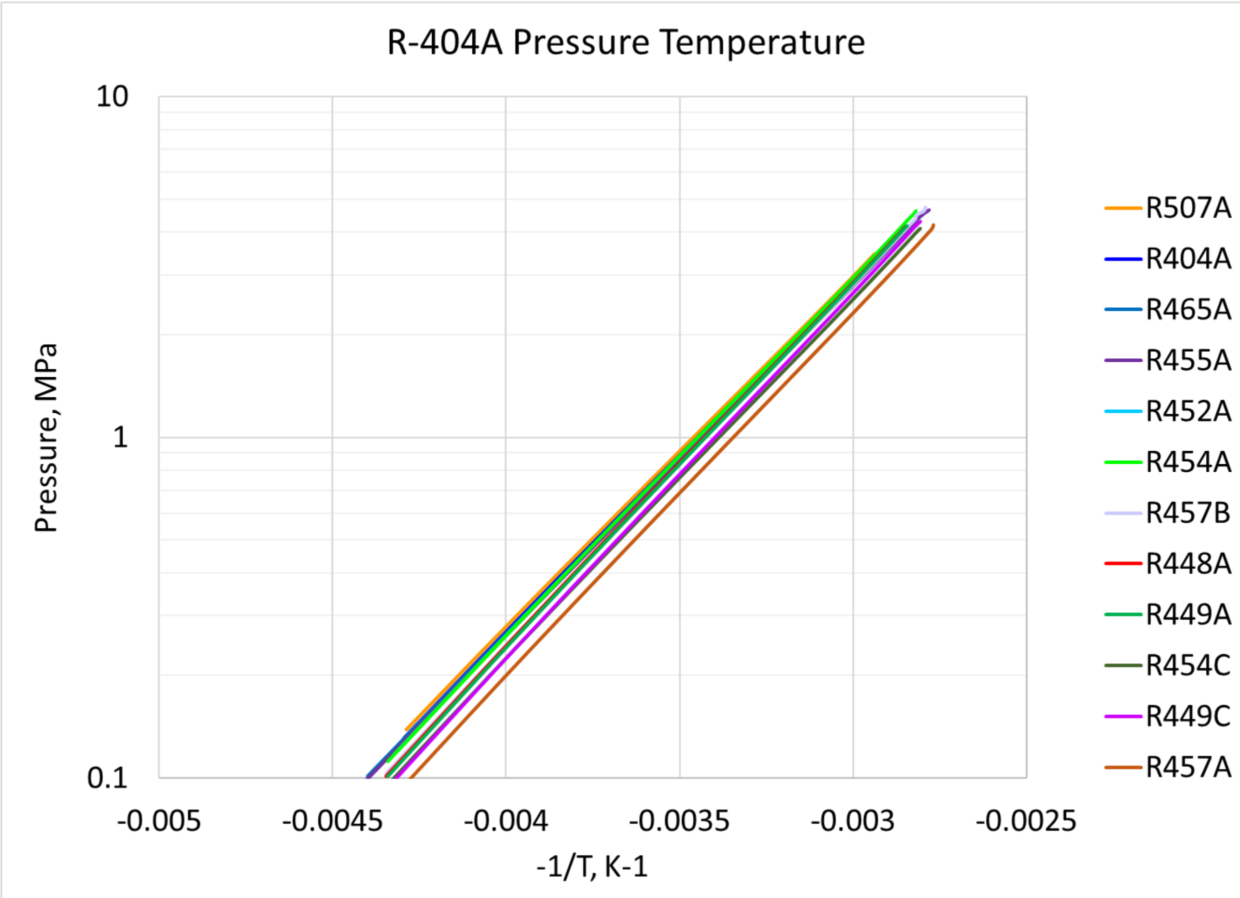

Figure 1-42. Pressure vs. -1/T for R-404A and alternatives. The legend is ordered based on the order in which the lines cross the horizontal axis, from left to right.

Above evaporating temperatures above 0°C, the volumetric capacities of R-449C, R-465A, R-454C, and R-457A lag behind the baseline, as shown in Figure 1-43. Figure 1-47 to Figure 1-49 also demonstrate this trend of decreasing performance in volumetric capacity relative to the baseline when transitioning from low-temperature applications to medium- and high-temperature applications.

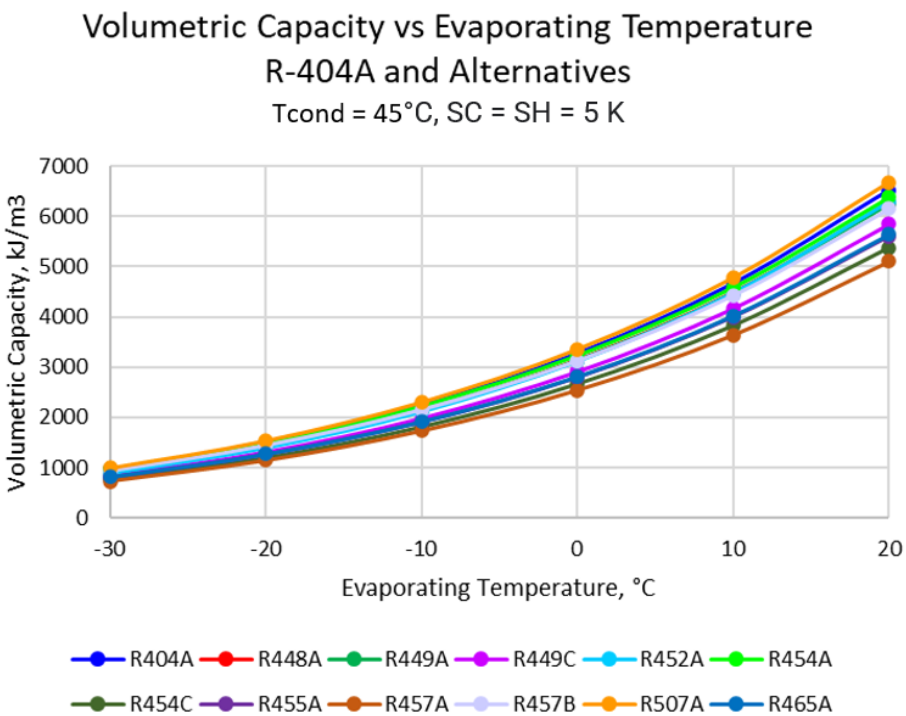

Figure 1-43. Volumetric capacity (calculated using the suction density condition) vs. evaporating temperature for R-404A and alternatives. Note that the condensing temperature differs from that given in the assumption summary (Table 1-15). The general trend can still be captured regardless of the difference in system conditions.

For refrigeration, several R-404A alternative refrigerants have wider cycles than R-404A, including R-448A, R-449A, and R-449C. The cycle diagrams demonstrate that the majority of evaporators and compressors would operate at lower pressures than the baseline (Figure 1-44-Figure 1-46).

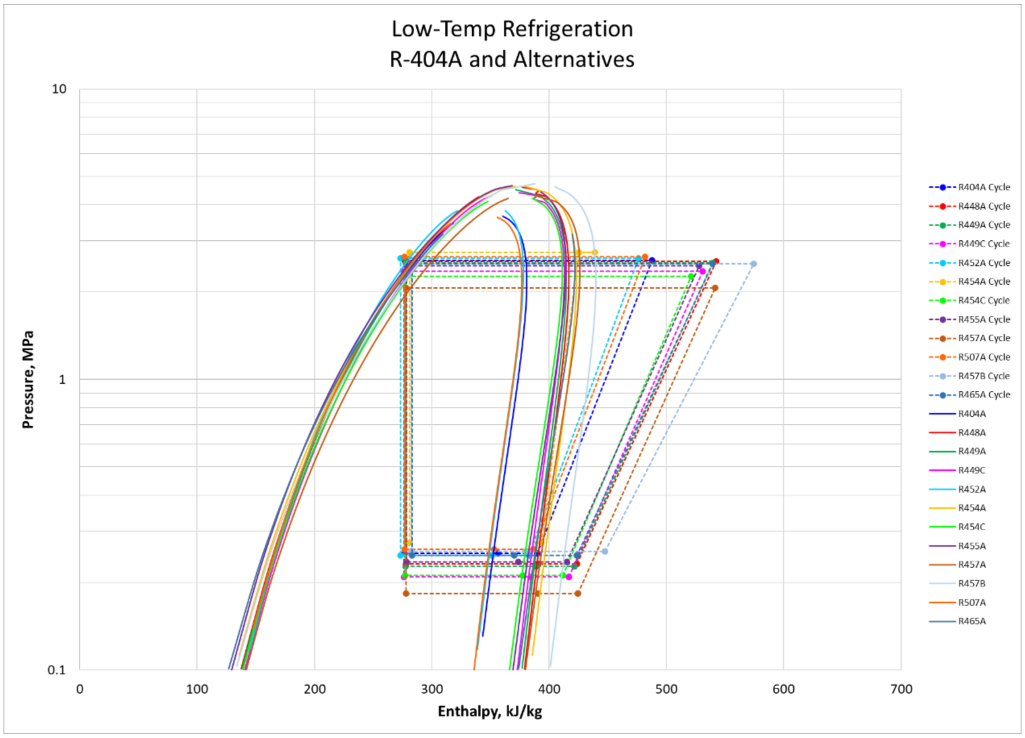

Figure 1-44. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-404A and alternatives for low-temperature refrigeration

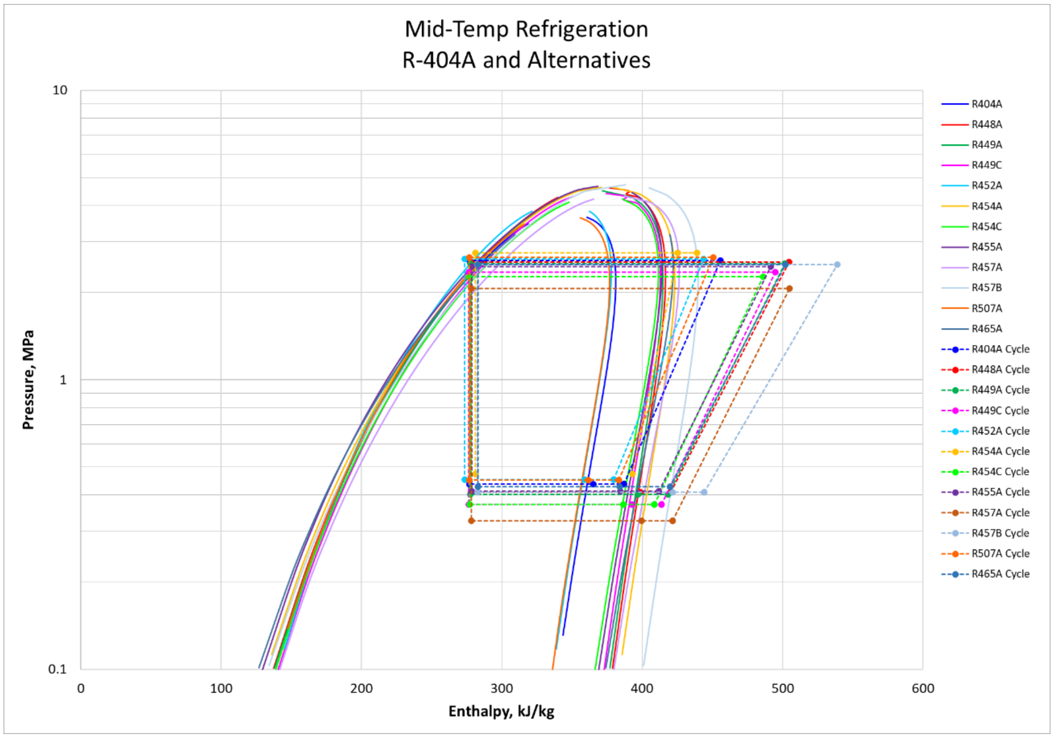

Figure 1-45. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-404A and alternatives for mid-temperature refrigeration

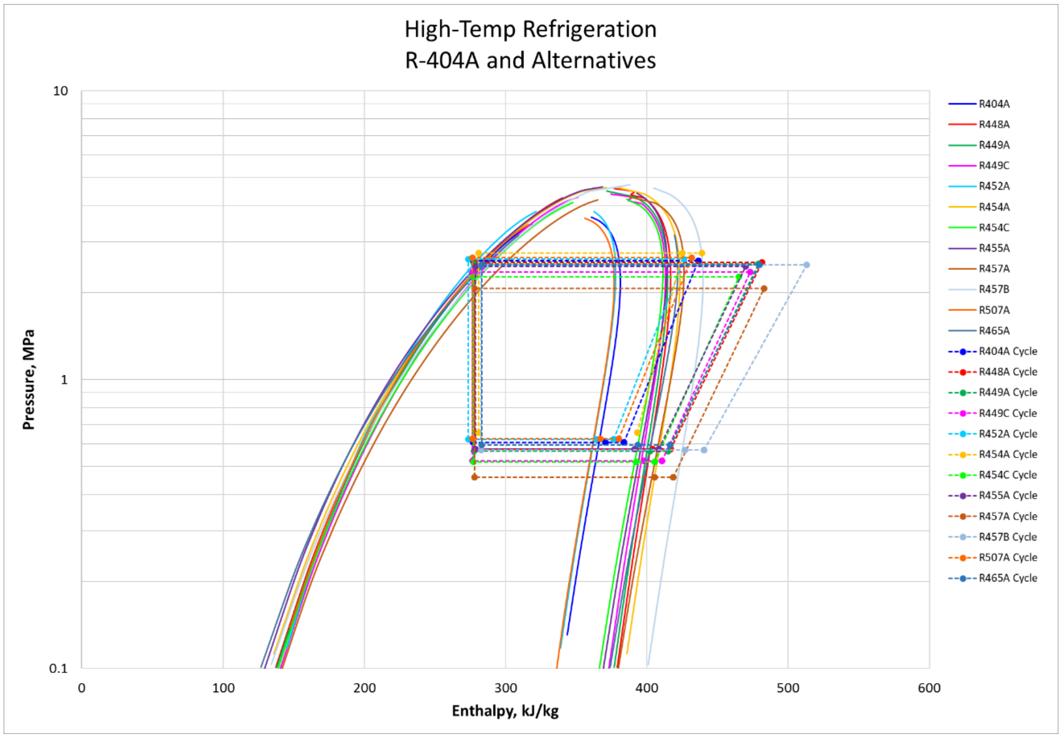

Figure 1-46. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-R404A and alternatives for high-temperature refrigeration

For all alternatives except R-465, R-507A, and R-457B, the COP is higher than the baseline across all temperature ranges (Figure 1-47-Figure 1-49). R-457B has a COP and volumetric capacity significantly higher than the baseline for mid- and high-temperature applications, but it performs under the baseline at low temperatures. Similarly, increasing from low-temp to mid- or high-temp refrigeration pulls performance above the baseline for R-452A. The overall lowest-performing alternative on the basis of relative COP is R-465A, which is at about 92% of the baseline for low-temperature applications, 96% for mid-temperature applications, and 99% for high-temperature applications. Generally, all the alternatives perform well in terms of COP for all three temperature ranges. Apart from R-456A, the only other instance of a COP below 95% is R-457B for low-temperature refrigeration (Figure 1-47-Figure 1-49).

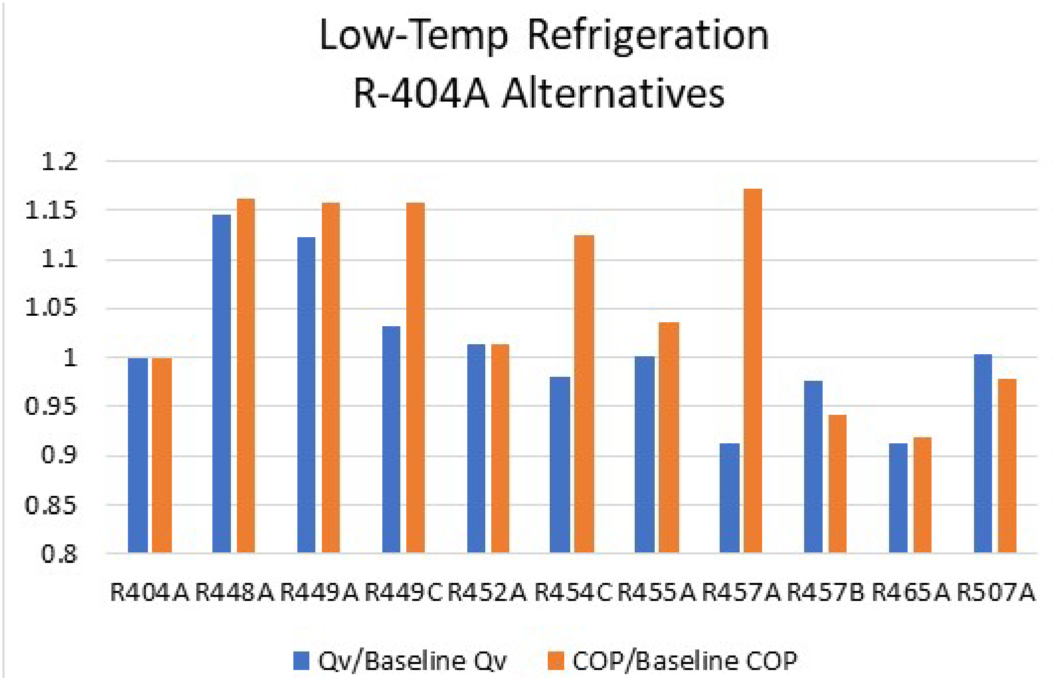

Figure 1-47. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-404A alternatives compared to the baseline for low-temperature refrigeration

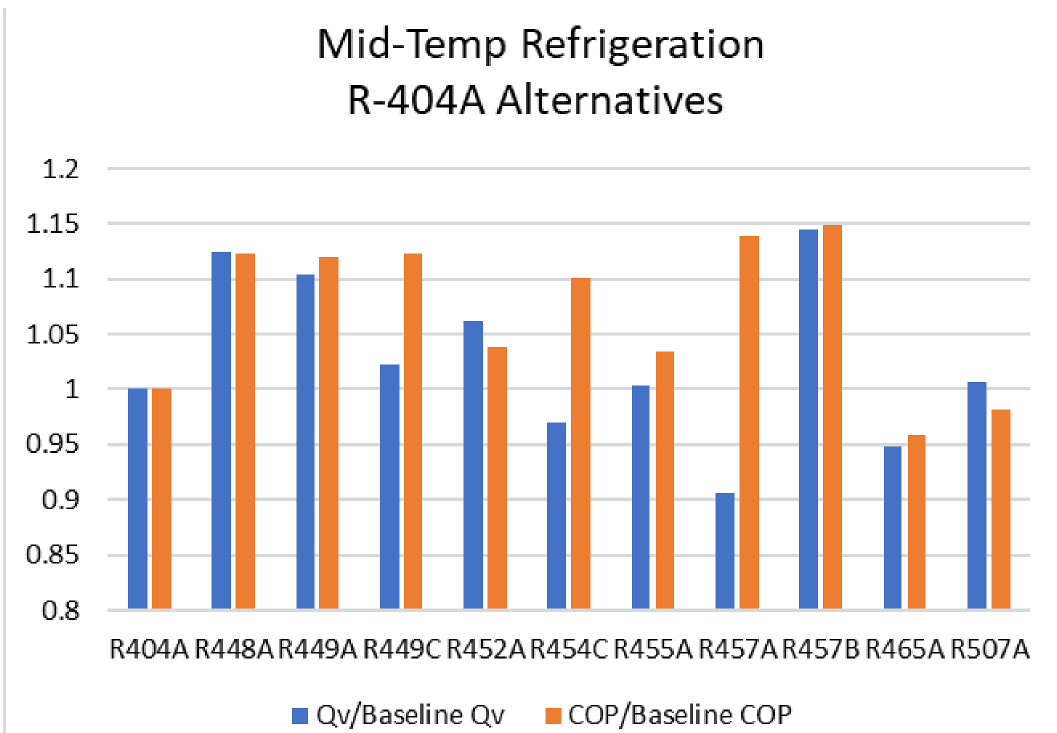

Figure 1-48. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-404A alternatives compared to the baseline for mid-temperature refrigeration

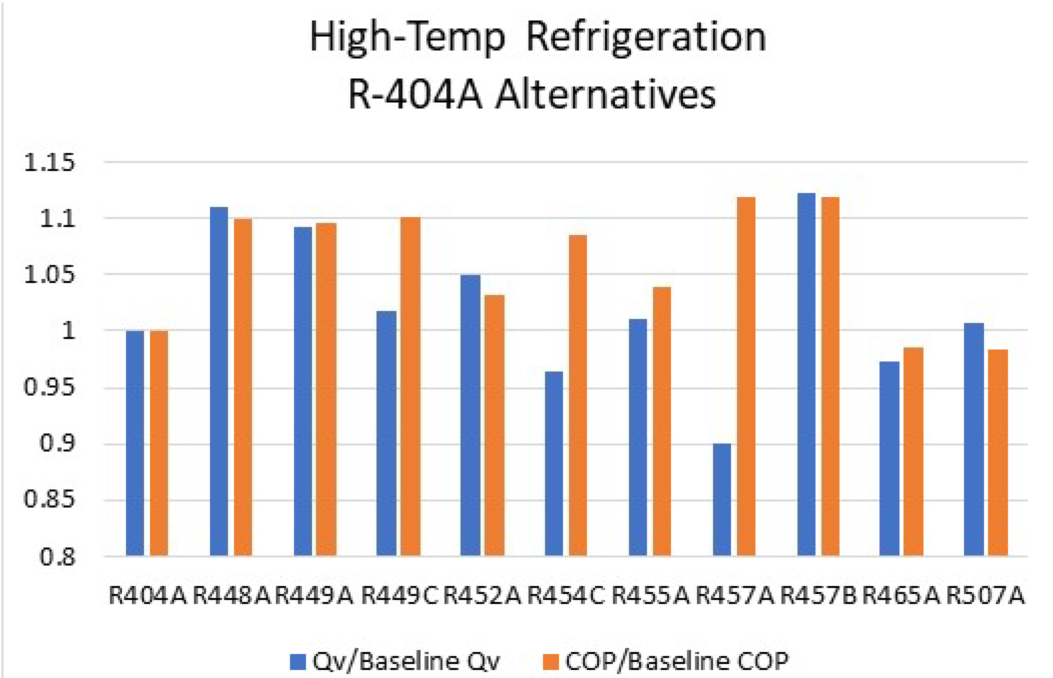

Figure 1-49. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-404A alternatives compared to the baseline for high-temperature refrigeration

1.6.4.1 R-410A: General Comparison

There is an opportunity for significant reduction in GWP relative to R-410A, particularly with the A2L-designated alternatives. R-466A and the A2L refrigerants all have GWPs under 750. The boiling points of the A2L alternatives are the most similar to R-410A, while the A1 alternatives have significantly lower boiling points. Overall, R-463A has the greatest difference in boiling point, with a decrease of about 7°C compared to R-410A (Figure 1-50). However, operating pressures of system components remain relatively the same, as shown by the cycle diagrams in Figure 1-53-Figure 1-56.

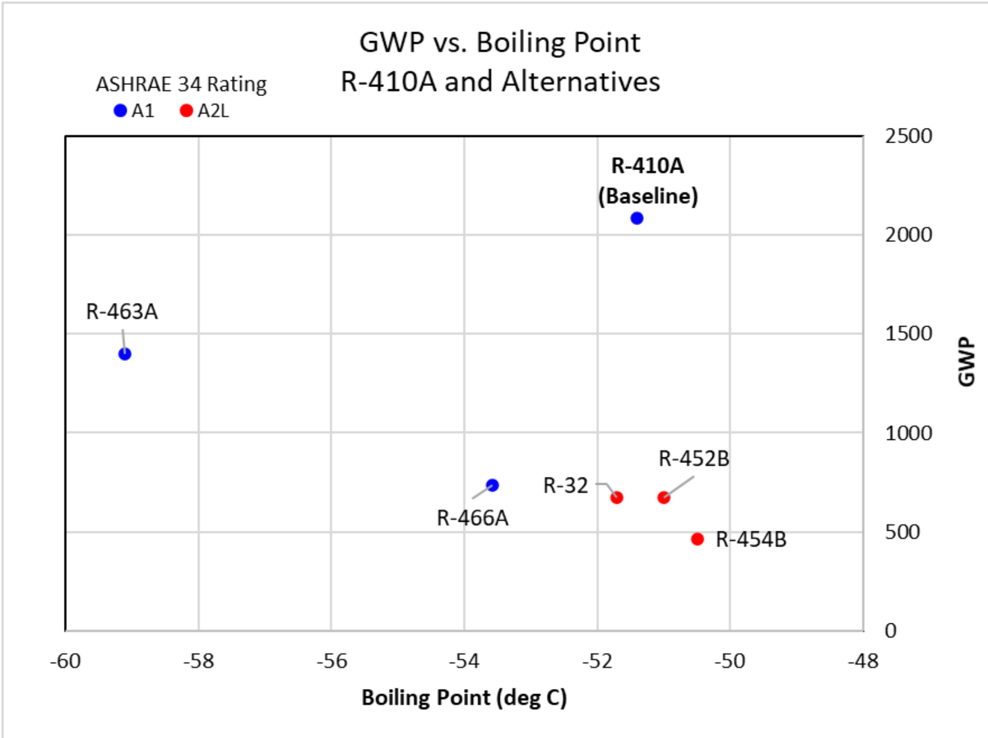

Figure 1-50. GWP vs. Boiling Point for R-410A and evaluated alternatives, including ASHRAE 34 classification

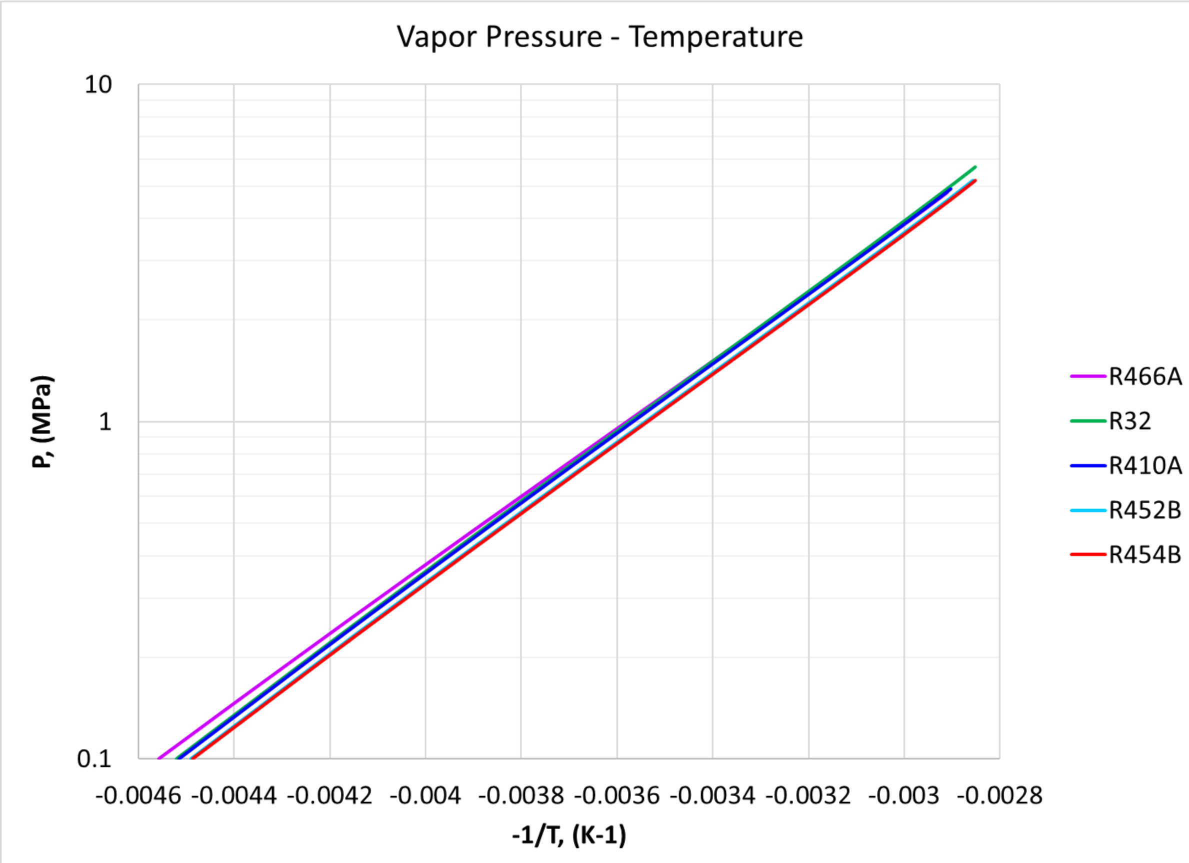

Figure 1-51. Pressure vs. -1/T for R-410A and alternatives. The legend is ordered based on the order in which the lines cross the horizontal axis, from left to right.

The alternatives on the P vs. -1/T diagram appear to have similar performance compared to R-410A (Figure 1-51). Thus, implementation of these alternatives may not require significant changes to the system equipment.

In general, R-32 stands out as a strong performer in volumetric capacity relative to the baseline, as shown in Figure 1-52, while R-463A performs rather poorly across all evaluated evaporating temperatures. As expected, alternative refrigerant performances have the least variation compared to R-410A low temperatures under 0°C. Performance also varies with application and heat exchanger fluid, as shown in the following two sections.

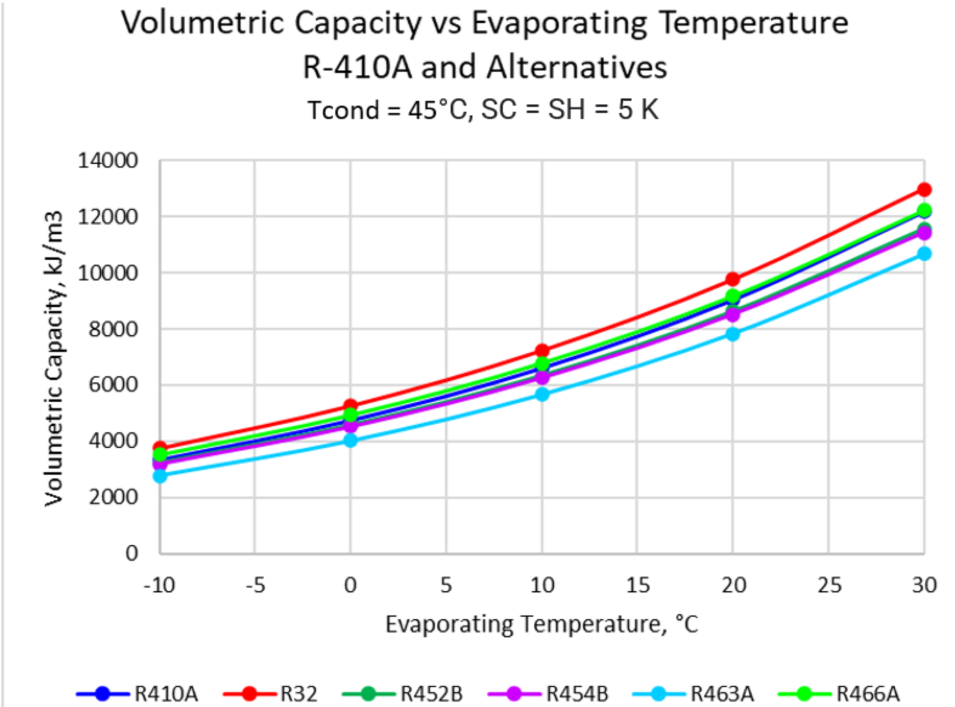

Figure 1-52. Volumetric capacity (calculated using the suction density condition) vs. evaporating temperature for R-134A and alternatives. Note that this analysis was conducted for only one of the condensing temperatures given in the assumption summary (Table 1-15). The general trend can be captured at any of the condensing temperatures.

1.6.4.2 R-410A: Heat Pumps

R-410A can be used in air-to-air, air-to-water, water-to-air, and water-to-water heat pumps. In all cases, operating pressures for the evaporator and condenser of alternatives to R-410A are fairly similar to the baseline (Figure 1-53-Figure 1-56). The refrigeration cycles for heat pumps that use water as the condensing fluid are shorter in height compared to other configurations due to higher evaporator pressures. Significant compressor downsizing can be achieved with R-32, and downsizing will also be enabled by R-452B and R-454B. This is true for all types of heat pump equipment. Unlike with R-134a chillers, changing the evaporating and condensing fluid results in notable differences in the shapes of the cycles, in terms of both the pressure and enthalpy ranges.

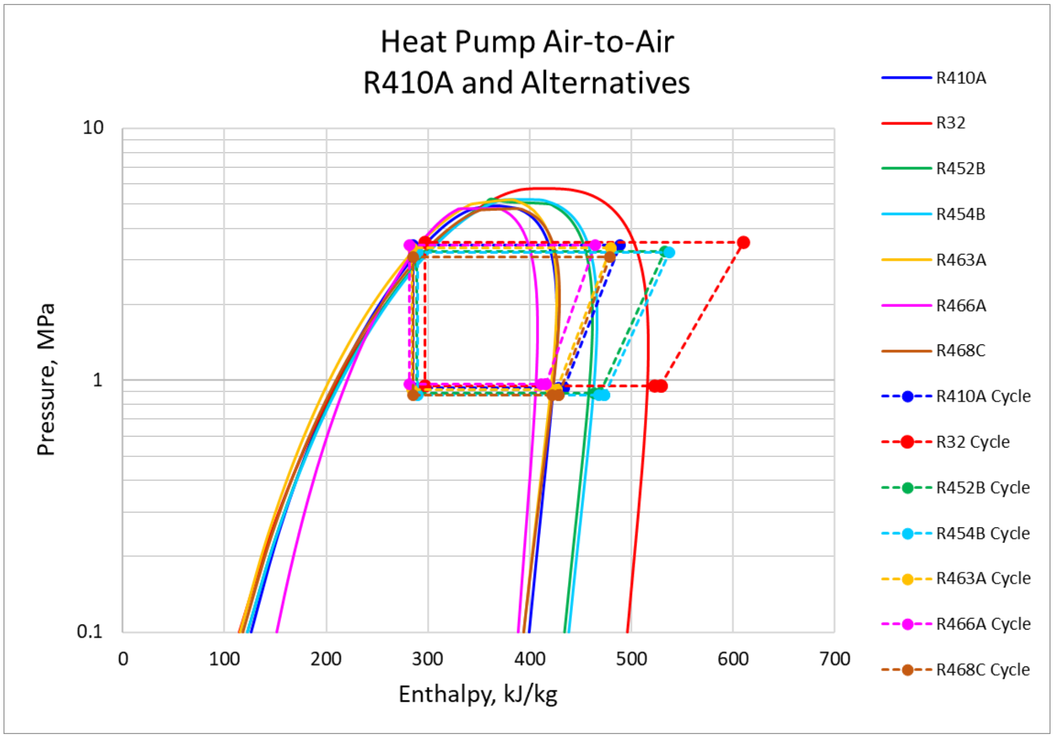

Figure 1-53. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-air heat pumps

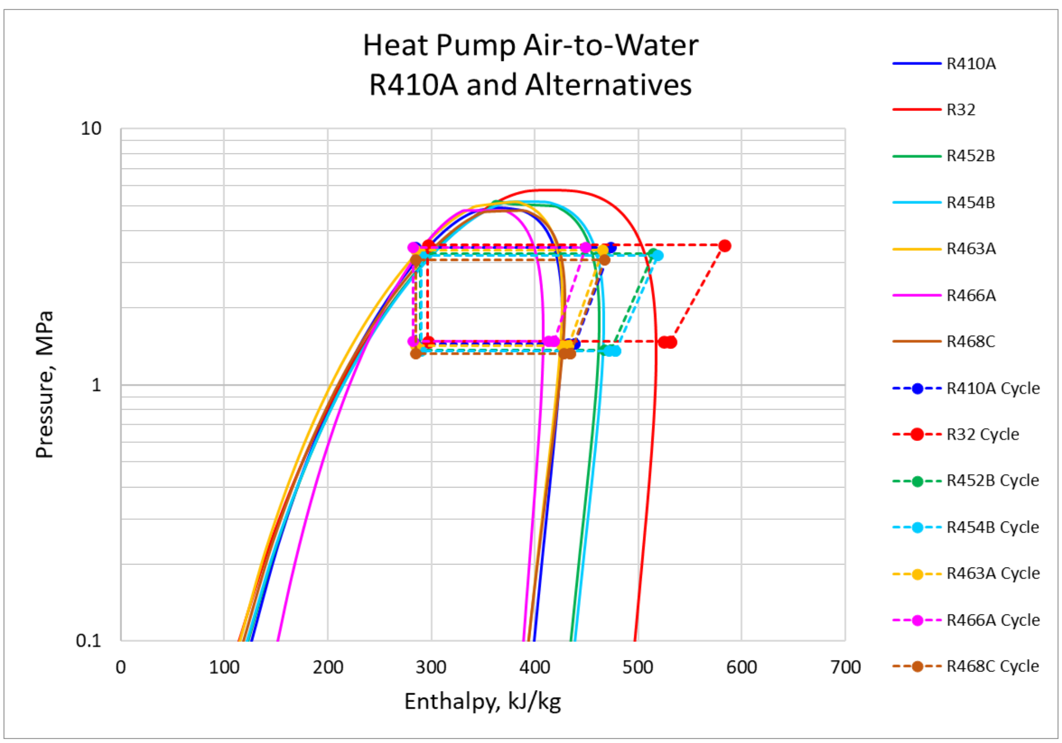

Figure 1-54. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-water heat pumps

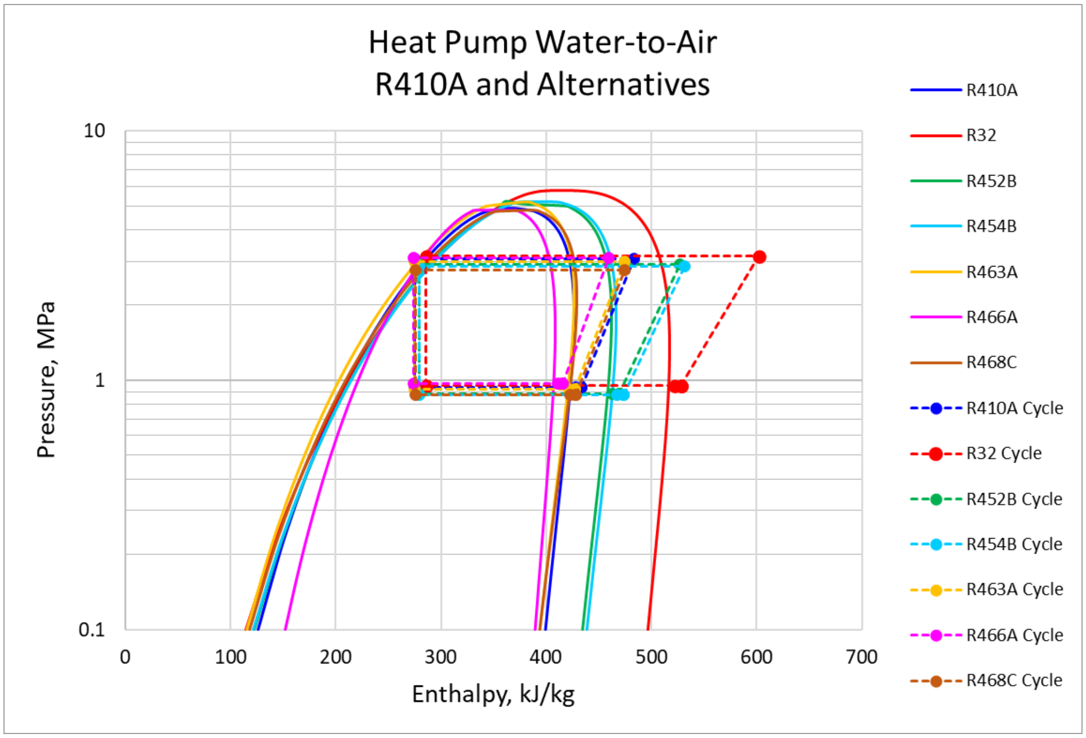

Figure 1-55. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-air heat pumps

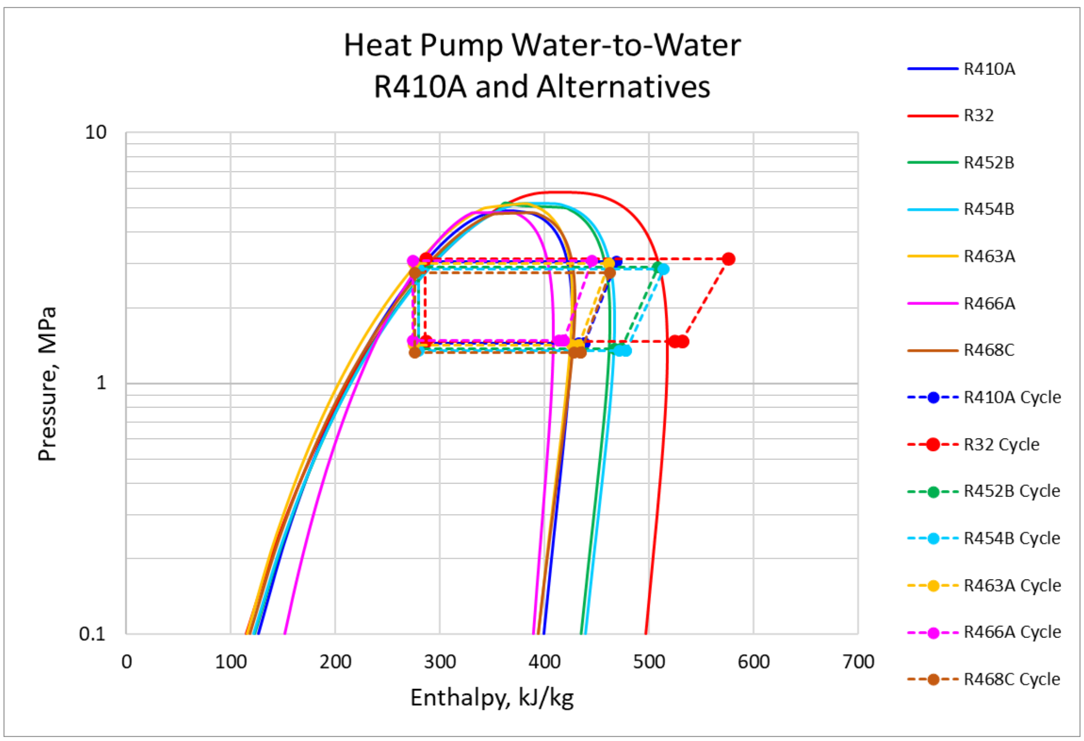

Figure 1-56. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-water heat pumps

For alternatives to R-410A in heat pumps, all of the evaluated refrigerants have COPs within 1% of the baseline COP for all four heat pump configurations (Figure 1-57-Figure 1-60). R-32 performs the best in terms of volumetric capacity (Q_V), having a Q_V 8-11% better than R-410A. A sacrifice in switching to R-32 would be its mild flammability (Figure 1-50). The poorest performance in volumetric capacity comes from R-468C, which has a Q_V consistently around 90% of the baseline. This level of performance is generally acceptable for a replacement refrigerant.

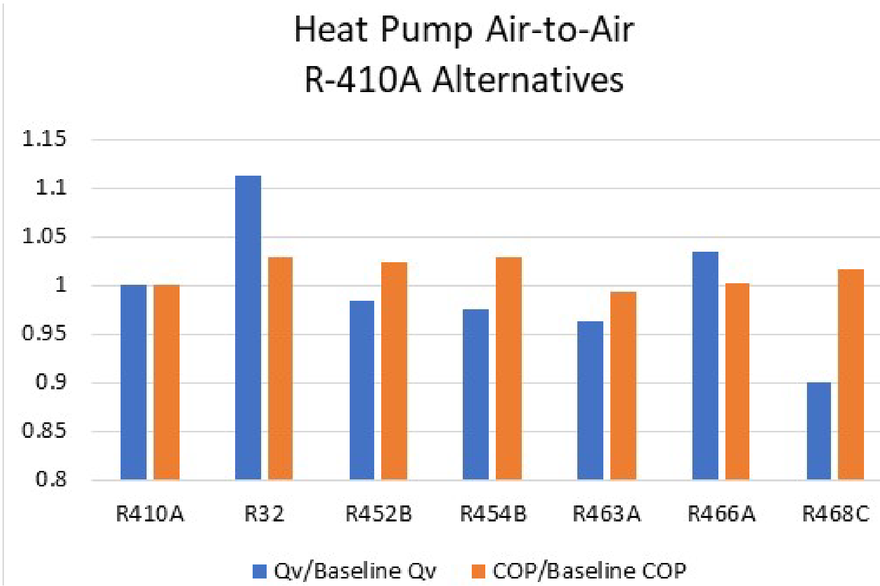

Figure 1-57. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-air heat pumps

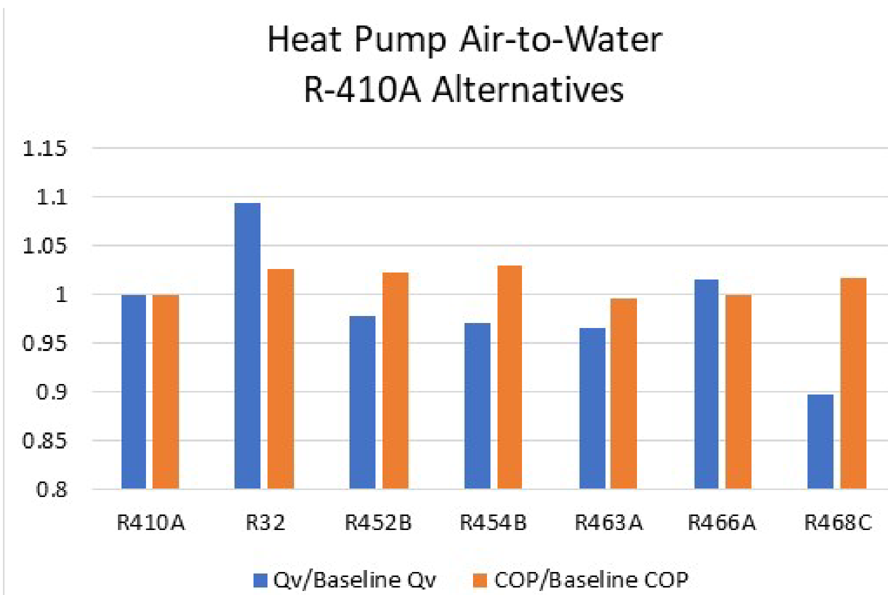

Figure 1-58. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-water heat pumps

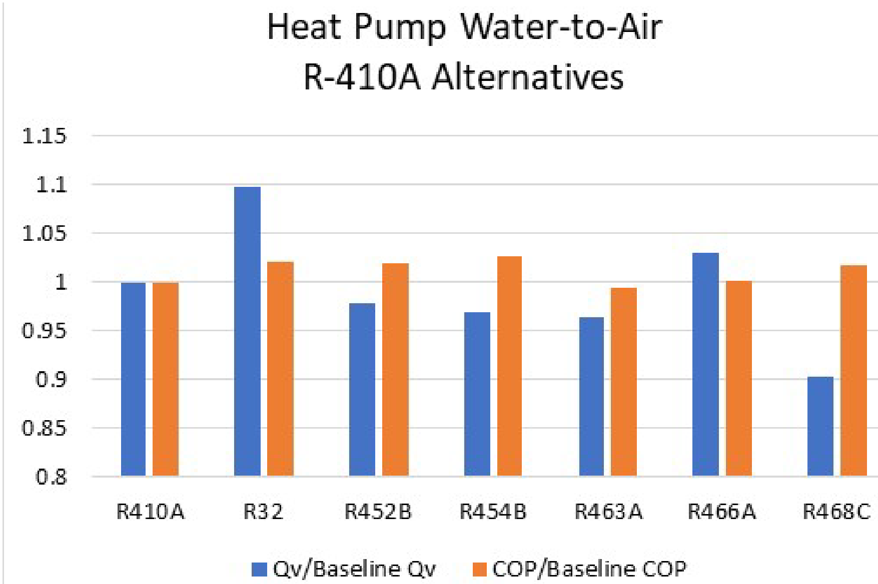

Figure 1-59. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for water-to-air heat pumps

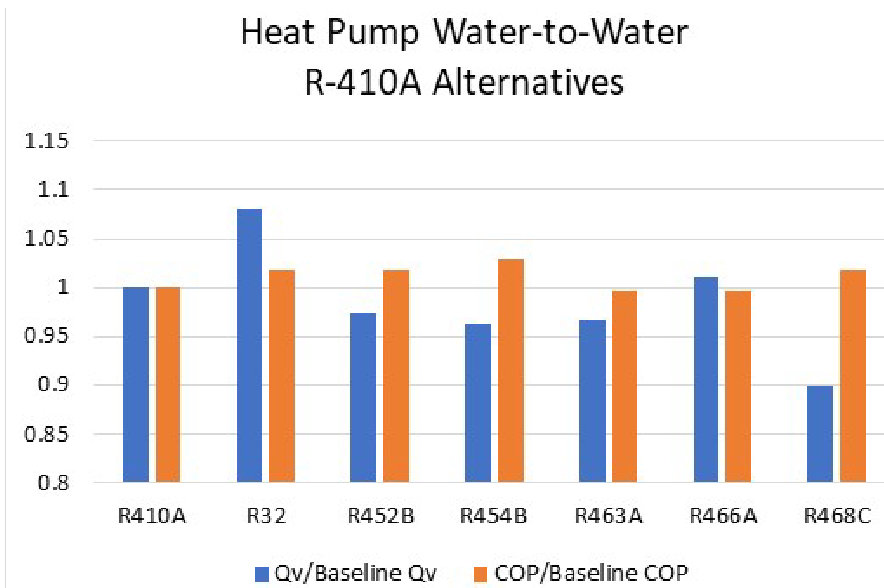

Figure 1-60. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for water-to-water heat pumps

1.6.4.3 R-410A: Building Air Conditioning

R-410A is also commonly used for building air conditioning. It was assessed for air-to-air, air-to-water, water-to-air, and water-to-water equipment.

The span of enthalpy increases is larger for A/C than it is for heat pumps for R-410A. The water-to-air configuration requires notably lower condenser pressures than the other configurations (Figures 45-48), which may reduce system costs. The other configurations are rather similar, except that water-to-water has a lower condenser pressure than the configurations where air is the evaporator fluid. The relative positions of the refrigerant cycles are very similar to those of the heat pump cycles, with R-32 having the widest cycle, and R-452B and R-454B having relatively wide cycles as well.

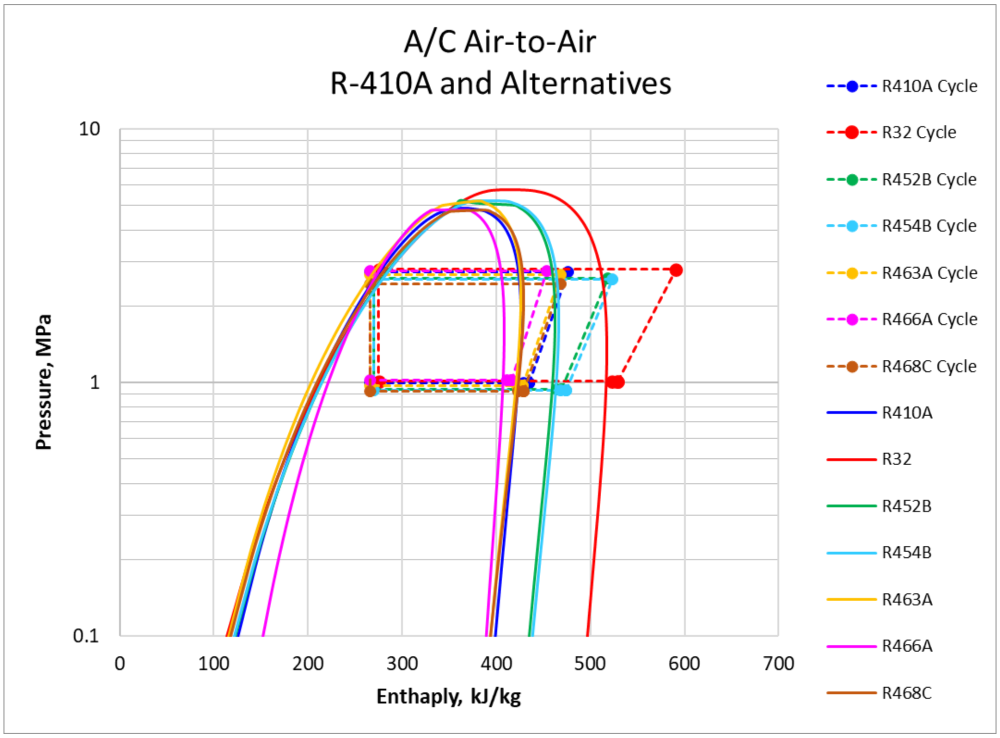

Figure 1-61. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-air air conditioning

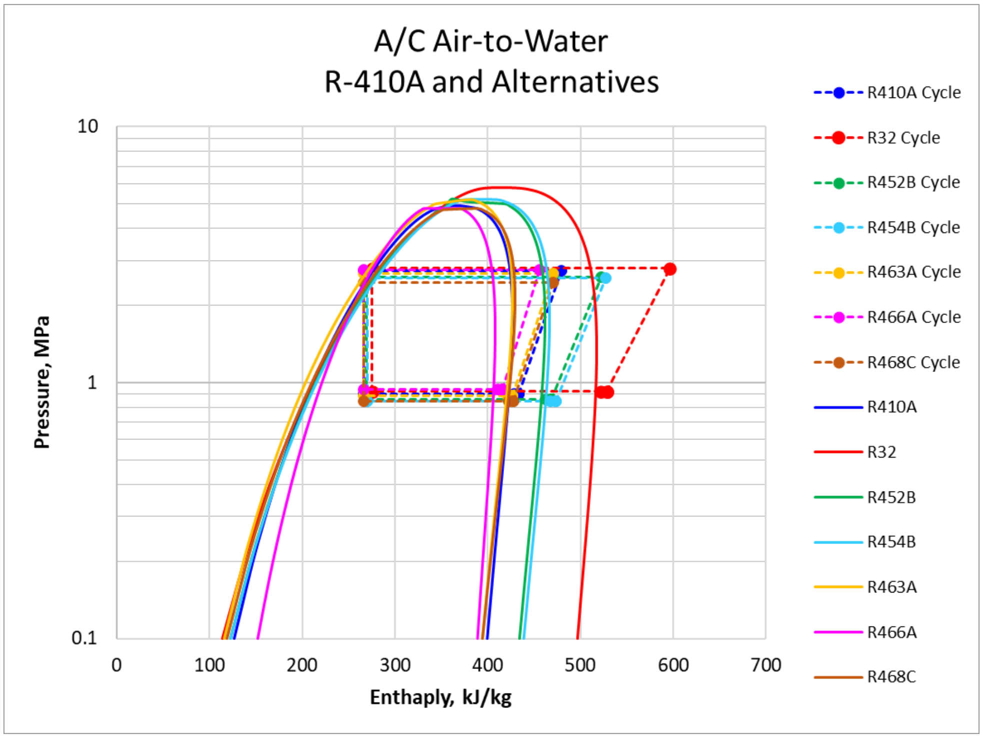

Figure 1-62. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-water air conditioning

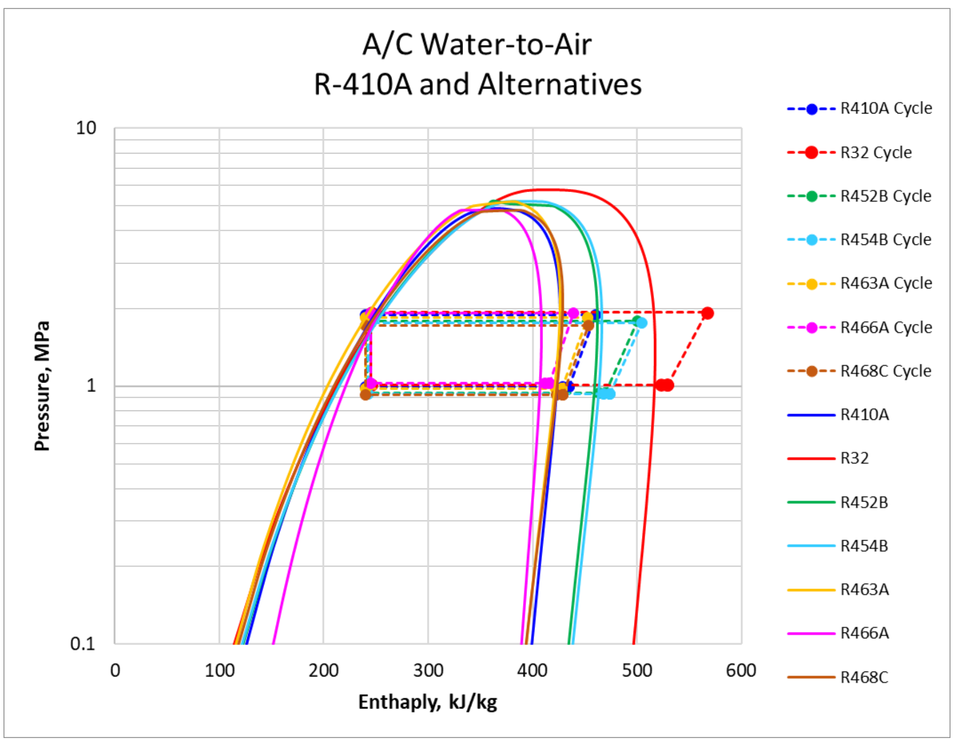

Figure 1-63. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-air air conditioning

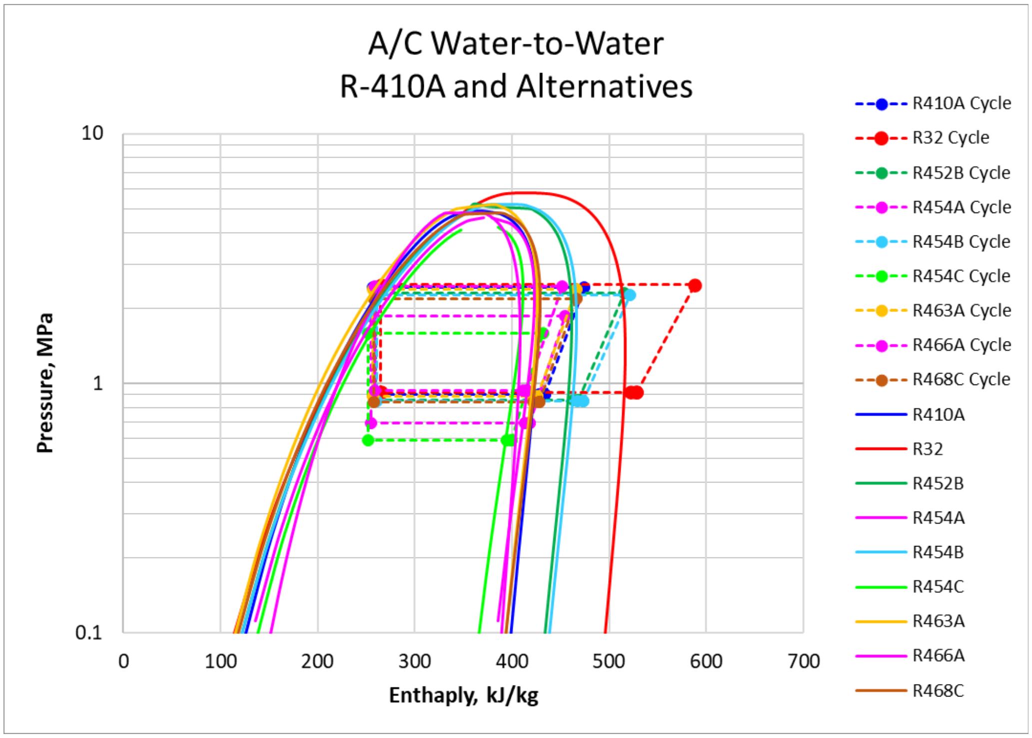

Figure 1-64. Pressure vs Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-water air conditioning

R-32 performs 6-10% better than baseline in terms of Q_V. All COPs are comparable to R-410A, with no relative COP being lower than 99% and some as high as 103% (Figures 49-52).

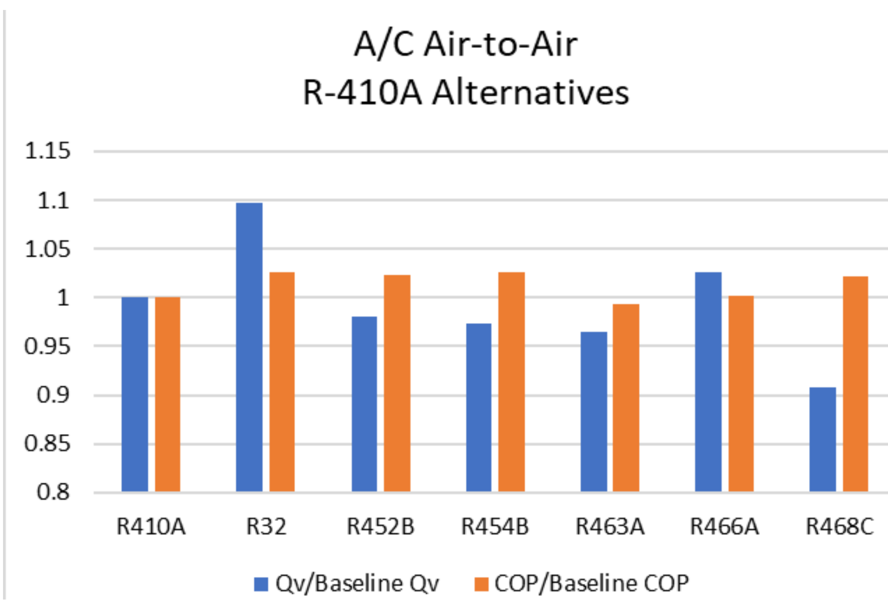

Figure 1-65. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-air air conditioning

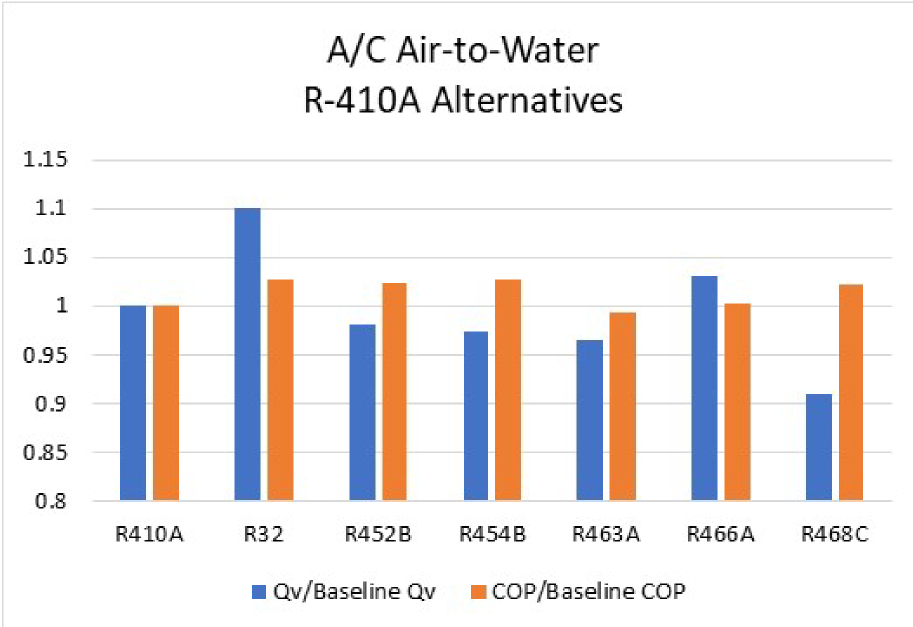

Figure 1-66. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-water air conditioning

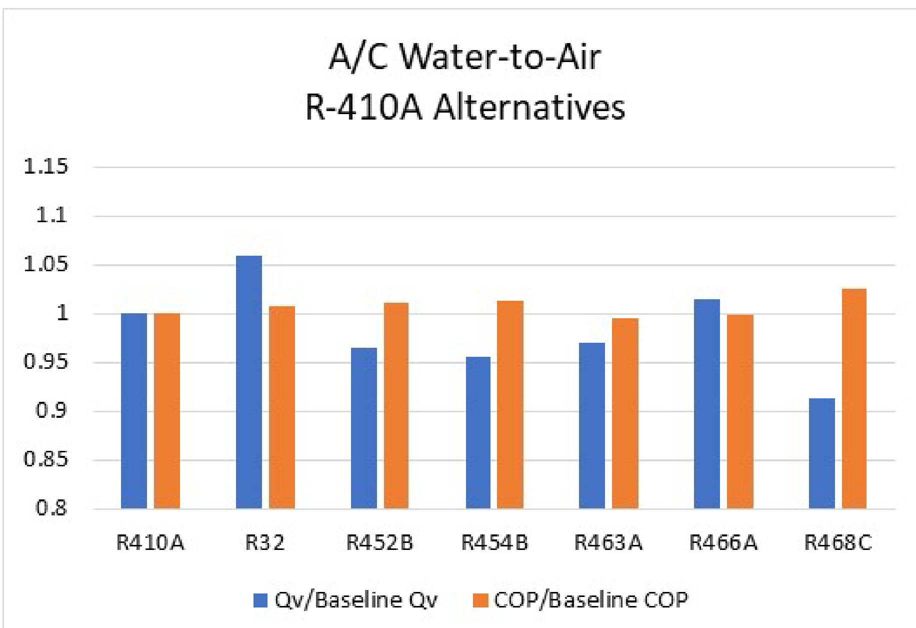

Figure 1-67. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for water-to-air air conditioning

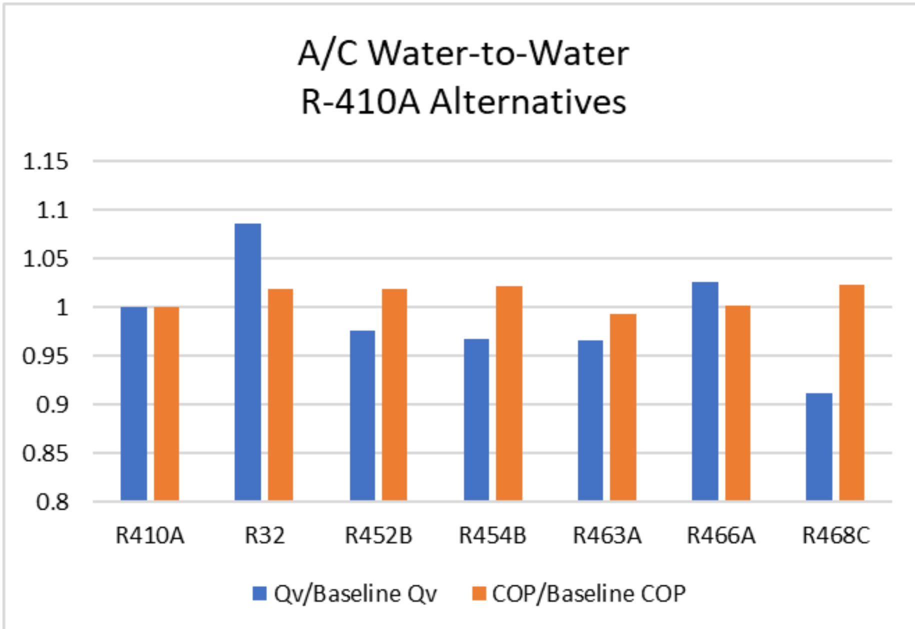

Figure 1-68. Relative volumetric capacity (Q_V) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for water-to-water air conditioning

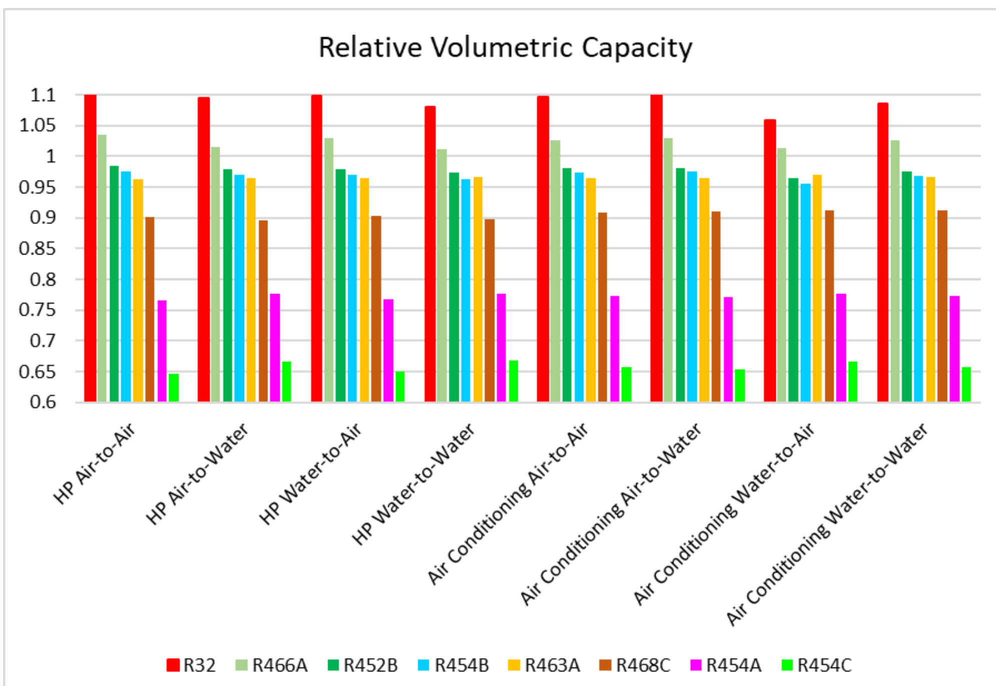

Figure 1-69. Summary of relative volumetric capacities of R-410A alternatives for all applications evaluated

In summary, across the different applications, relative Q_V follows a similar trend with small fluctuations. The performance ranking from best to worst on Q_V is: R-32, R-466A, R-410A, R-452B, R-454B, R-463A, R-468C, R-454A, R-454C (Figure 1-59).

1.6.5 Summary and Discussion

For each refrigerant category, there are viable lower-GWP alternatives with thermodynamic performances comparable to the baseline refrigerant, as summarized in Table 1-16.

Table 1-16. Summary of alternative refrigerants that meet GWP, COP, and volumetric capacity requirements

Traditional Refrigerant	Application	Short-Term Alternatives[1]	Long-Term Alternatives[2]	Meets or Exceeds Baseline COP (99.9%+)	Acceptable Volumetric Capacity (>= 90% of Baseline)
R-123	Chillers	R-1224yd(Z)	R-1233zd(E)	N/A	R-1233zd(E), R-1224yd(Z)
R-245fa	Heat Pumps			R-1233zd(E), R-1234ze(Z)	R-1234ze(Z)
R-134a	Chillers, W2W, A2W	R-1234ze(E), R-1234y, R-513A, R-456A, R-450A	R-516A, R-515B, R-444A	R-1234ze(E), R-450A, R-515B	R-1234yf, R-444A, R-456A, R-513A, R-516A
	Mobile Air Conditioning			N/A	
	Heat Pumps				
R-404A	Heat Pumps	R-448A, R449A, R449C, R-452A, R454C	R-454A, R-454C, R-455A, R-457A, R-457B, R-465A	R-448A, R-449A, R-449C, R-452A, R-454C, R-455A, R-457A, R-457B (mid- and high-temp only)	All Alternatives
R-410A	Heat Pumps	R-32	R-452B, R-454B, R-466A	R-32, R-452B, R-454A, R-454B, R-454C, R-466A (air-to-air and water-to-air only), R-468C	R-32, R-452B, R-454B, R-463A, R-466A, R-468C (air-to-air and water-to-air only)
	Air Conditioning	R-32, R-463A	R-452B, R-454B, R-466A, R-468C	R-32, R-452B, R-454A, R-454B, R-454C, R-466A (excluding water-to-air), R-468C	R-32, R-452B, R-454B, R-463A, R-466A, R-468C

Because each refrigerant has unique properties, there may be tradeoffs when considering alternatives, and manufacturers can weigh these trade-offs based on needs. For instance, a refrigerant with a lower GWP may have a significantly lower volumetric capacity, requiring investment in a larger compressor. In another case, a refrigerant may perform very well on COP

[1] Short-term alternative refrigerants are defined as those which are currently available and are being offered by at least one commercial provider.

[2] Long-term alternative refrigerants are defined as those which are listed in ASHRAE 34 and have the potential to replace the baseline refrigerant with similar performance but have not yet been commercialized.

but poorly on volumetric capacity. Flammability can also vary between alternatives, and this may become a deciding factor, in spite of performance differences.

1.6.6 Conclusions

The data, as presented throughout this report, can be used to determine the “best overall” or most viable refrigerant based on thermodynamic analysis. Again, the definition of “best” depends on the needs and priorities of the user. It is important to note that when testing these refrigerants in actual equipment, performance may differ due to the physical behavior of the system. In the next milestone (M2), four pieces of candidate equipment will be selected, and in M3, they will be rigorously analyzed with the relevant traditional and alternative refrigerants.

1.6.7 Optimization of a Residential Air Source Heat Pump Using Refrigerants with GWP <150 for Improved Performance and Reduced Emission

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1.6.7.1 Introduction

Modern cooling technologies are significant sources of greenhouse gas emissions (GHGs) with total CO_2 equivalent emissions from the HVAC sector accounting for 7.8% of global GHG emissions ([1]. Considering the commitment to reduce the impact of GHGs on climate in the HVAC&R sector, a transition from fluorinated substances to alternative refrigerants with reduced global warming potential (GWP) values is supported by F-gas Regulation ([2], the Montreal Protocol with the Kigali Amendment [3] of which 146 countries have not ratified, and the Paris Agreement [4] and the US AIM ACT. The requirements as set forth by the F-gas Regulation banned the use of refrigerants with a GWP of 2,500 or greater for high refrigerant charge stationary HVAC equipment in 2020. Beginning in 2,022, a GWP limit of 150 has been set for multi-circuit cascade systems for commercial use with a nominal capacity of 40 kW or more, and for 2025, the GWP limit for single split AC on the European Union (EU) market is set as 750. This ban will not permit the use of R410A (2088 as GWP value) in small charge system applications. In this regard, much research has been conducted to find alternative low-GWP refrigerants.

Reducing the environmental impacts of HVAC&R systems has been an important research topic due to recent severe global climate changes. In residential buildings, space heating, and cooling are the main energy consumers, which rely on vapor compression-based heat pumps. The HVAC&R industry has moved to phase out refrigerants with high global warming potentials (GWP), e.g., R410A, R22, R134a, R404A, etc. The next-generation refrigerants are mostly mixtures of HFO (Hydrofluoroolefins) refrigerants, e.g., R1234yf and R1234ze(E) combined with the HFC (Hydrofluorocarbons) refrigerant, e.g., R32. Since most of these low-GWP mixtures are in the new A2L lower flammability, research [5] has shown that promoting the use of smaller diameter tubes in heat pump systems is an effective way to reduce refrigerant charge and avoid explosion risk. But it may cause performance degradation.

One characteristic of these low-GWP alternative refrigerants is their high glide, i.e., temperature glides from the bubble point to the dew point at one pressure. High-glide refrigerants prefer multi-row, counter-flow heat exchanger configurations for a single-mode operation. If switching mode, the counter-flow heat exchanger (HX) becomes a parallel-flow heat exchanger. The reversed flow causes significant efficiency degradation. Therefore, improvements in components and system configurations are needed to make the high-glide refrigerants work for both cooling and heating modes to achieve good energy efficiency and protect the environment.

This study demonstrates a low-GWP heat pump design method with a particular focus on the use of a new system configuration for dual-mode reversible heat pumps and 5-mm, 7-mm, and 9-mm tube-optimized heat exchangers and required modification in compressors. The goals of the study include:

- Investigate the effect of a new system configuration to maintain heat exchanger flow configuration under both cooling and heating modes.

- Optimize multi-row 5-mm, 7-mm, and 9-mm tube coils with low-GWP refrigerants and compressors to determine the performance improvements.
- Analyze the annual performance indices of optimal heat pumps and assess their life cycle climate performance (LCCP) to assess carbon footprints.

1.6.7.2 Methodology

1.6.7.2.1 System Model

The DOE/ORNL Heat Pump Design Model (HPDM) [6] is used to model the performance of heat pumps. HPDM is a public-domain HVAC equipment and system modeling and design tool, which supports a free web interface and a desktop version for public use. A finite volume (segment-to-segment) tube-fin HX model is used to simulate the performance of the heat exchanger with different circuitries. This model has been validated by the experimental data [7]. The dehumidification model used in the evaporator simulation is a heat and mass transfer effectiveness-based model [8]. More details of HPDM can be found in [9]. In HPDM, REFPROP 10.0 [10] is used to produce performance look-up tables and simulate the refrigerant properties.

1.6.7.2.2 Selection of Refrigerants

In literature [11], near-term refrigerant candidates have GWP<750, and R32 and R454B can be good drop-in options. Long-term options require GWP<150, and those fluids usually do not match the incumbent fluid (R410A) properties. For fluids with GWP lower than 150, such as R454C, R455A, and R457A, they require significant changes in heat exchanger structure to address the high-temperature glide (temperature rise from the refrigerant bubble point to dew point at constant pressure) and redesign of the compressor to compensate the capacity and efficiency degradation. R1234yf and R1234ze(E) have ultra-low GWP values, and they are of interest in the long term. Table 1-17 shows the characteristics of R410A and its low-GWP alternatives for a typical residential air source heat pump. The temperature glides are evaluated at saturation pressure corresponding to 8 °C dew-point temperature

Table 1-17. Characteristics of refrigerants investigated in this research.

Refrigerant	GWP	Safety Class	Composition: Mass Fraction	Glide in Evaporator [K]	Critical Temperature [°C]
R410A	2088	A1	R32/R125: 50%/50%	0.1	72.8
R32	675	A2L	R32: 100%	0	78.1
R454B	466	A2L	R32/R1234yf: 68.9% /31.3%	1.3	78.1
R454C	146	A2L	R32/R1234yf: 21.5% /78.5%	7.7	85.7
R457A	139	A2L	R32/R1234yf/R152a: 18%/70%/12%	6.9	90.1
R455A	145	A2L	R32/R1234yf/CO_2: 21.5%/75.5%/3%	11.71	85.61
R1234yf	4	A2L	R1234yf: 100%	0	94.7
R1234ze(E)	4	A2L	R1234ze(E): 100%	0	153.7

1.6.7.2.3 Heat Pump System Configuration and Heat Exchanger Structural Parameters

In this study, a 3-ton R410A residential single-stage heat pump product on the market is modeled. Figure 1-70 shows the schematic of the baseline heat pump operating under cooling mode and heating mode. The refrigerant direction inside the heat exchangers is reversed between mode switching.

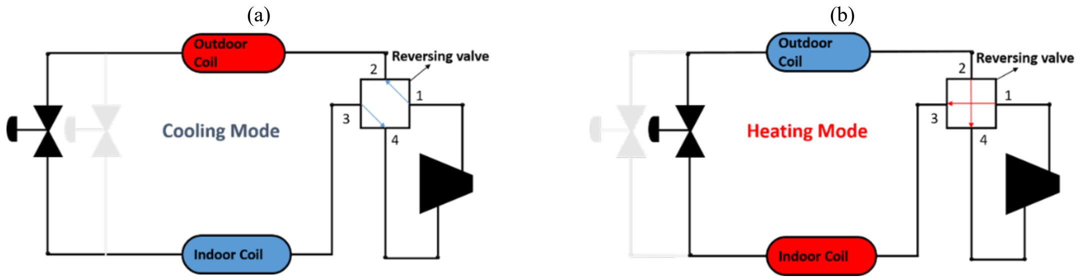

Figure 1-70. R410A Baseline Heat Pump System: (a) Cooling Mode Operation; (b) Heating Mode Operation.

HPDM has been closely calibrated against experimental data to model the baseline system. Table 1-18 lists the structural parameters of the baseline heat exchangers. The condenser fan moves 2,876 CFM of airflow across the outdoor HX and consumes 263 W of power; the indoor blower provides 1,205 CFM of supply airflow and consumes 435 W of power. The fan/blower powers and flow rates were the same among all refrigerants.

Table 1-18. Parameters of Indoor and Outdoor Units of Baseline 3-ton R410A Heat Pump.

Parameters	Indoor HX	Outdoor HX
Coil length [mm]	431.80	2496.82
Coil height [mm]	1248.20	609.60
Coil depth [mm]	35.61	22.00
Number of tubes	120	24
Number of rows	2	1
Number of tubes per row	60	24
Number of circuits	6	3
Circuit pattern (Figure 1-71**Error! Reference source not found.**)	cross mixed flow	cross mixed flow
Fin type	Louver fin	Louver fin
Fin density [fins/inch]	16	22
Tube outside diameter [mm]	7.95	9.52
Tube thickness [mm]	0.2794	0.3048
Tube horizontal spacing [mm]	17.81	22.00
Tube vertical spacing [mm]	20.80	25.40

Table 1-19. Correlations adopted in condenser and evaporator simulations

Operating Mode	Heat Transfer Correlations	Pressure Drop Correlations
Refrigerant - Liquid Phase	Dittus and Boelter (1985) [12]	Blasius (1907) [13]
Refrigerant - Two-Phase Boiling (Evaporator)	Thome and Hajal (2003b) [14]	Choi et al. (1999)[15]
Refrigerant - Two-Phase Condensation (Condenser)	Cavallini et al. (2006) [16]	Choi et al. (1999) [15]
Refrigerant - Vapor Phase	Dittus and Boelter (1985) [12]	Blasius (1907) [13]
Air	Wang et al. (1999) [17] for 7 & 9-mm tube	Wang et al. (1999) [17] for 7 & 9-mm tube
	Sarpotdar et al. (2016) [18] for 5-mm tube	Sarpotdar et al. (2016) [18] for 5-mm tube

Table 1-19 lists the empirical correlations used for local heat transfer and pressure drop. For air-to-refrigerant heat exchangers, the thermal resistance is mostly dominated by the airside.

Therefore, it is crucial to model the airside heat transfer coefficient accurately. Different correlations are used according to their suitable application range to improve the prediction of small-diameter tube and large-diameter tube heat exchangers.

Figure 1-71 illustrates 3 typical circuitry patterns by a simple six-tube HX. For the conventional system configuration as used in the baseline system, if the heat exchanger has a counter-flow configuration (Figure 1-71(a)) in cooling mode, after the refrigerant flow is reversed in heating mode, the circuitry will become parallel-flow (Figure 1-71(c)). And this yields undesirable performance degradation in heating mode.

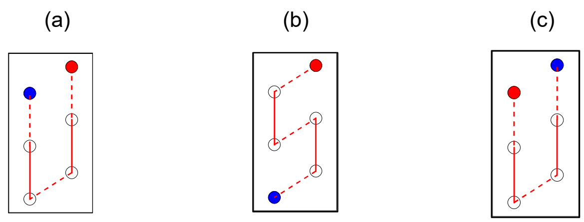

Figure 1-71. Heat exchanger circuitry patterns: (a) counterflow, (b) mixed flow, (c) parallel flow

To mitigate the heating performance degradation, the heat exchanger in the baseline system is designed as a mixed-flow configuration (Figure 1-72 (b)) as a compromise solution to balance between cooling and heating modes. The detailed circuitries of the baseline indoor and outdoor heat exchangers are shown in Figure 1-73. The indoor HX has 60 tubes per row and 2 tube rows and is divided into 6 mixed flow circuits. The outdoor HX has 24 tubes and 1 tube row and is divided into 3 circuits. Different colors represent different circuits. The baseline indoor heat exchanger uses 7.95 mm diameter tubes, and the outdoor heat exchanger uses 9.52 mm diameter tubes.

Figure 1-72. Baseline tube-fin heat exchanger circuitries: (a) Indoor HX with 7 mm tube; (b) Outdoor HX with 9 mm tube.

As mentioned in the previous section, research has shown that for low-GWP zeotropic mixtures, the counterflow configuration has the most efficient heat transfer performance due to significant saturation temperature glide [5]. With this consideration, one goal of this research is to develop a new system configuration to maintain the same refrigerant flow direction inside the heat exchangers between cooling and heating mode switching. Figure 1-73 shows the proposed reversible heat pump system configuration evaluated in the study as the optimized system. The new configuration has 4 one-way check-valves at the inlet of indoor and outdoor heat exchangers, and the heat exchangers have bi-directional distributors. Under both heating and cooling modes, this system configuration can maintain the same heat exchanger circuitry pattern.

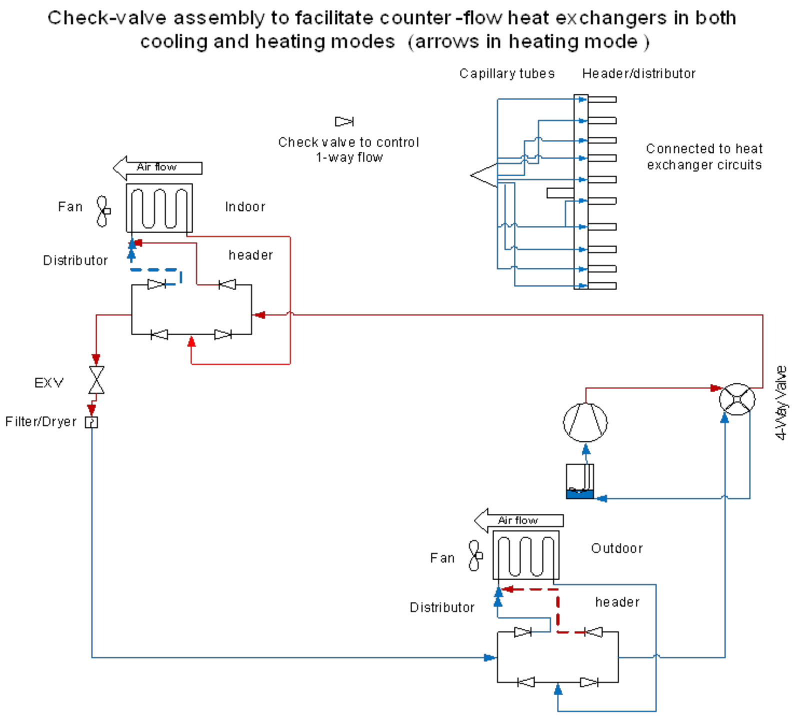

Figure 1-73. New System Configuration with Check-valves and Bi-directional distributors Maintaining Counter-flow HXs in Dual Modes

1.6.7.2.4 <u>Optimization Problem Formulation</u>

Shen et al. (2012) [19] developed an optimization framework that integrates HPDM with GenOpt ([20]), a public domain optimization package. In this research, the Particle Swarm Optimization (PSO) algorithm implemented in GenOpt is used to optimize the heat pump system. Regarding the PSO setting, the optimization runs use 100 as the population size and 200 as the number of generations.

Equation (1) shows the optimization problem formulation. The objective is to maximize the Energy Efficient Ratio (EER) of the heat pump under AHRI Standard 210/240 [21] cooling test A condition (95 °F). In Equation (1), the number of circuits for indoor and outdoor heat exchangers is two design variables, which vary between 1 and the number of tubes in each row of the heat exchangers. The number of circuits has an adaptive upper limit to coordinate with the number of tubes in each row.

$$\begin{aligned}
&Maximize: EER \\
&Subject\ to: \\
&\quad Heat\ exchanger\ tube\ diameter\ \text{varies among } 5\ mm, 7\ mm, 9\ mm \\
&\quad 1 \le N_{circuits,evaporator} \le Ntubes\ per\ bank\ of\ evaporator \\
&\quad 1 \le N_{circuits,condenser} \le Ntubes\ per\ bank\ of\ condenser \\
&\quad \Delta T_{\text{superheat, evaporator outlet}} = 10 - \frac{\Delta T_{glide}}{2}\ [R] \\
&\quad 2\ [R] \le \Delta T_{\text{subcooling, condenser outlet}} \le 15\ [R] \\
&\quad Q_{evaporator} = 10.55\ \text{kW} \\
&\quad |\ \text{SHR}_{evaporator} - \text{SHR}_{baseline,evaporator}\ | \le 1\% \\
&\quad \text{Height}_{evaporator} = \text{Height}_{baseline} \\
&\quad \text{Length}_{evaporator} = \text{Length}_{baseline} \\
&\quad \text{Height}_{condenser} = \text{Height}_{baseline} \\
&\quad \text{Length}_{condenser} = \text{Length}_{baseline}
\end{aligned} \tag{1}$$

In terms of constraints on operating conditions, the evaporator outlet superheat degree is specified based on the temperature glide of refrigerants as recommended by refrigerant OEM. The condenser outlet subcooling degree is automatically adjusted, but it is constrained between 2 R to 15 R. The cooling capacity of the evaporator is an HPDM solving target and set to be the same as the baseline 3-ton R410A heat pump. The compressor displacement volume is altered in HPDM to meet the target evaporator cooling capacity. When modeling the optimized systems using low-GWP refrigerant other than R410A, the compressor isentropic efficiency is fixed at 0.74, and the volumetric efficiency is fixed at 0.98 to be consistent with the baseline R410A system. The goal is to investigate the required displacement volume of the compressor to meet the system capacity for low-GWP refrigerants.

In terms of constraints on heat exchanger dimensions, the last four constraints in Equation (1) guarantee that the optimized indoor and outdoor heat exchangers have the same frontal shapes as the baseline heat exchangers, i.e., the optimal heat exchangers can fit into the original indoor and outdoor fan-coil unit perfectly. This can ease the retrofit effort of upgrading the R410A heat pump to the new low-GWP system by minimizing the change in manufacturing and installation processes and guaranteeing that the optimized systems have the best compatibility with end-users' house structure. As a result, the new products can be easily accepted by manufacturers and end-users.

The staggered tube layout and tube spacing for 5-mm, 7-mm, and 9-mm tubes used in the optimized coils are shown in

. These tube layouts and spacings are off-the-shelf designs from a heat exchanger manufacturer. The heat exchanger circuitry pattern is fixed as a counterflow configuration since the counterflow pattern has the most efficient heat transfer for high-glide zeotropic mixtures [5].

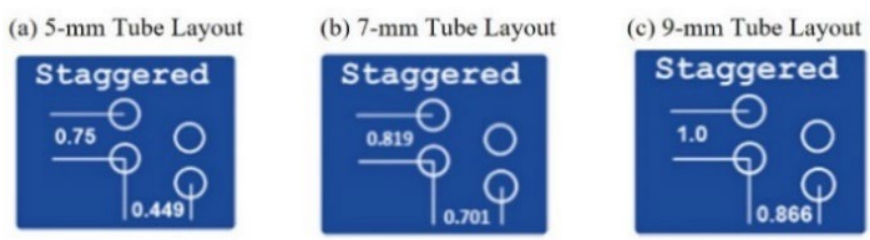

Figure 1-74. Horizontal spacing and vertical spacing in inches for (a) 5-mm tube HX, (b) 7-mm tube HX, and (c) 9-mm tube HX.

1.6.7.3 Results

1.6.7.3.1 Drop-in Performances

Without modification of the system configuration and components, shows the drop-in performance for cooling operation at 95 °F. In terms of capacity (Figure 1-75 (a)), R32 induces a capacity increase of 8.3%, and R454B shows a 1.48% capacity decrease compared with the R410A baseline. If the baseline R410A heat exchangers and compressor are used, R454C, R455A, R457A, R1234yf, and R1234ze(E) induce 29.4%, 26.8%, 32.9%, 46.9%, and 55.5% capacity degradation, respectively, without changing any component.

In terms of EER, the variation is very small. R32, R454B, R457A and R1234yf induces EER increases by 2.2%, 3.2%, 1% and 3.3%, while R454C, R455A and R1234ze(E) and induces EER decreases by 0.7%, 2.6% and 0.4%, respectively.

For the drop-in test, the compressor isentropic efficiency and volumetric efficiency are predicted by a 10-coefficient compressor map provided by the compressor manufacturer. Despite the compressor map being regressed for R410A test data, a scaling method assuming the same isentropic and volumetric efficiencies at the same suction and discharge saturation temperatures for R410A is used to predict the compressor performance when other refrigerants are used. This compressor scaling method is validated against experiment data [11] and demonstrates good prediction accuracy.

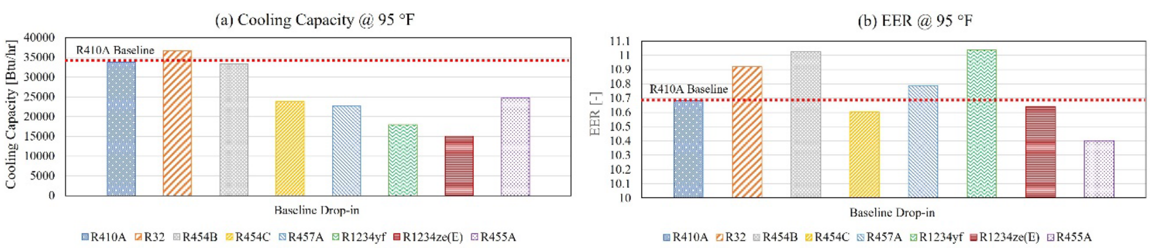

Figure 1-75. Baseline System Drop-in Cooling Performance at 95 °F (a) Cooling Capacity; (b) EER

Similar to the cooling performance, Figure 1-76 shows the drop-in performance for heating operation under AHRI 210/240 heating C_1 condition at 47 °F. R32 shows a 6.7% capacity increase and 0.03% COP increase. R454B shows comparable performance to R410A, i.e., a 3.9% capacity decrease and 1.72% COP increases. R454C, R455A, R457A, R1234yf, and R1234ze induce 6.5%, 27.6%, 38.9%, 52.3%, and 62.5% capacity degradation, respectively.

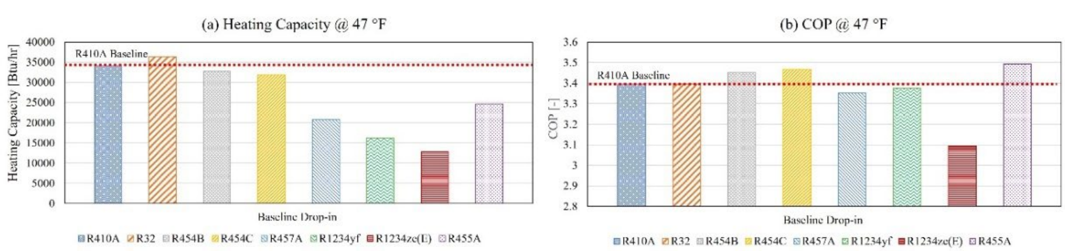

Figure 1-76. Baseline System Drop-in Heating Performance at 47 °F (a) Heating Capacity; (b) COP

In conclusion, these results show that R32 and R454B are good drop-in candidates. However, GWP < 150 refrigerants cannot be directly drop-in due to the significant capacity degradation. For low-GWP refrigerants, it is necessary to redesign the system configuration and components to meet the performance metrics. One required change for the compressor is to increase the displacement volume to increase capacity.

1.6.7.3.2 Optimization Results

The previous section shows the drop-in test results, which demonstrate the system efficiency and capacity degradation with the drop-in of low-GWP refrigerants. This section shows the results after conducting design optimization of the components and adopting the new system configuration. For all optimization cases, the suction line is sized such that the saturation temperature drop of the suction line for different refrigerants is the same as that of the baseline R410A system. The goal is to investigate the required displacement volume of the compressor to match the R410A system capacity and efficiency requirements.

All cooling-optimized systems using 5-mm, 7-mm, and 9-mm tube heat exchangers satisfy the cooling capacity requirements (i.e., 36000 BTU/hr) to match the baseline 3-ton unit. In this study, the advantage of using 5 mm tube HXs attributes to the increased heat transfer area and reduced mass flux in each circuit as shown in Table 1-20. Despite the optimized heat exchangers have the same coil width and coil height as the baseline coils, 5-mm tube heat exchangers consist of more tubes in each row and more tube rows due to smaller horizonal and vertical tube spacing as shown in Figure 1-74. Figure 1-77 shows the cooling performance comparison at 95 °F. For R410A, R32, R454B, R455A, and R457A, the optimized 5-mm tube heat exchangers yield 9.4%, 11.6%, 9.4%, 5.5%, and 5.1% EER improvements, respectively. Except for R32 and R454B, using 7-mm tube and 9-mm tube with GWP less than 150 refrigerants yields performance degradation compared with the baseline drop-in test. For refrigerants with GWP less than 150, heat exchanger design optimization does not show efficacy in improving system performance. In this study, the advantage of using 5 mm tube HXs is attributed to the increased heat transfer area and reduced mass flux in each circuit, as shown in Table 1-20. Despite the optimized heat exchangers having the same coil width and coil height as the baseline coils, 5-mm tube heat exchangers consist of more tubes in each row and more tube rows due to smaller horizontal and vertical tube spacing, as shown in Figure 1-74.

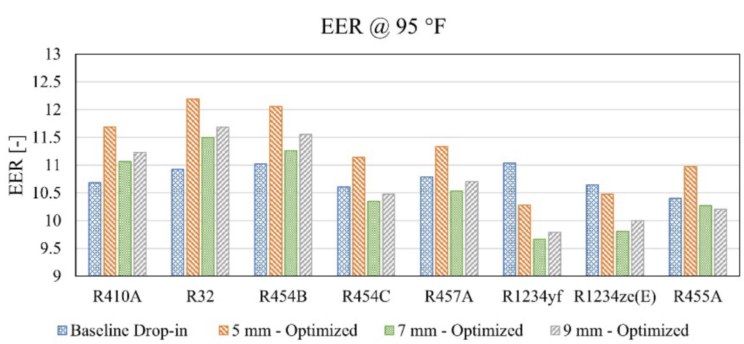

Figure 1-77. Cooling Performance Comparison of Baseline and Optimal Systems at 95 °F (a) Cooling Capacity; (b) EER

The cooling-optimized designs are evaluated under heating mode at 47 °F. Their performances are depicted in Figure 1-78. As shown in Figure 1-78(a), the optimal systems using most refrigerant fluids can satisfy the heating capacity of the R-410A baseline, except R-1234ze(E), which induces significant capacity degradation. In terms of COP (Figure 1-78(b)), systems using 5-mm tube-optimized heat exchangers show the most performance improvements, i.e., 11.4%, 12.7%, 10.2%, 0.05%, 1.2%, 6.3%, 1.0% and 10.0% for R-410A, R-32, R-454B, R-455A, R-457A, R-1234yf and R-1234ze(E), respectively. The performance of optimized 7-mm tube HX is close to the baseline system, and the performances of the system using 9-mm tube HXs are better than the baseline for R-32, R-454B, R-454C, R-457A, and R-1234ze(E) but worse than the baseline for R-455A and R-1234yf.

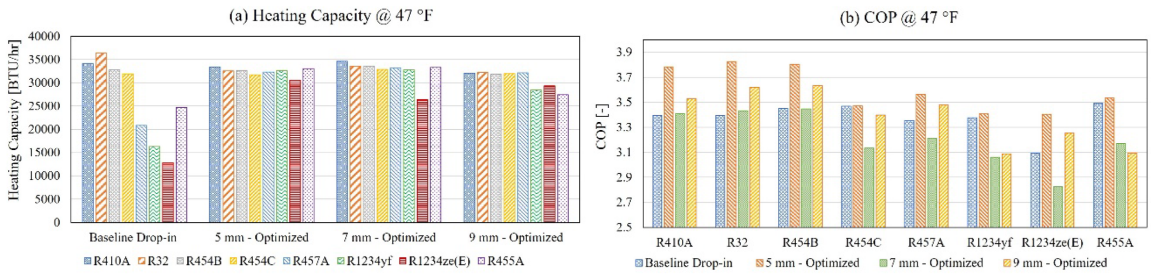

Figure 1-78. Heating Performance Comparison of Baseline and Optimal Systems at 47 °F (a) Heating Capacity; (b) COP

Figure 1-79 (a) shows the refrigerant charge of the optimized systems. The use of a 5-mm tube heat exchanger yields charge reduction. The charge reductions range from 28.4% to 43.7%. While 7-mm tube heat exchangers increase the system charge, 9-mm tube heat exchangers have a comparable charge amount compared with the baseline system. Figure 1-79 (b) shows the redesigned compressor displacement volumes for each refrigerant to match the baseline cooling capacity. The required compressor displacement volume is not sensitive to the tube diameters used in the heat exchanger, rather, it is sensitive to the property of fluids. In general, R-32 induces displacement volume decreases by 15.8% while R-454B, R-455A, R-457A, R-1234yf and R-1234ze(E) induce increased displacement volume increases by 4.7%, 40.2%, 50.7%, 62.2%, 152.6% and 224.2%. These results demonstrate the importance of compressor redesign for low-GWP heat pumps. In general, systems using low-GWP refrigerants require larger compressors than systems using R-410A.

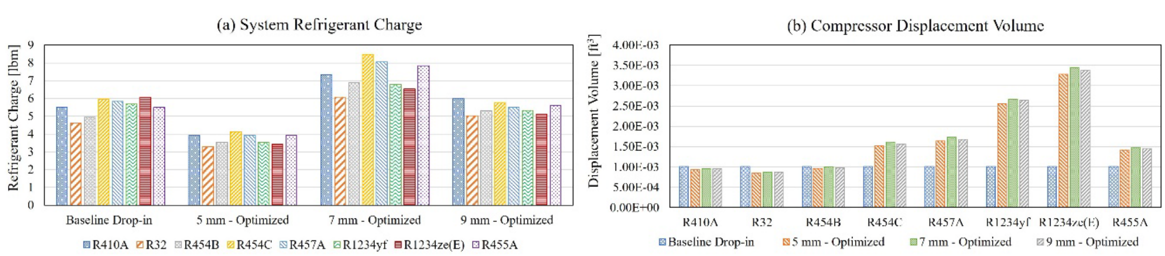

Figure 1-79. (a) System Refrigerant Charge; (b) Designed Compressor Displacement Volume.

To access the annual performance of optimized systems. The seasonal energy efficiency ratio-2 (SEER2) and heating seasonal performance factor-2 (HSPF2) are calculated according to AHRI 210/240 test standards [21]. Effective January 1st, 2023, SEER2 and HSPF-2 are used to rate system performance following a more stringent testing procedure. The new testing procedure increases the systems' external static pressure by a factor of five to better reflect field conditions of installed equipment in a typical ducted system and results in a lower numerical rating value for the same product. The performance degradation owing to frost accumulation has been considered by applying performance degradation factors, i.e., 0.91 for heating capacity and 0.985 for power consumption at the 35˚F dry bulb/33˚F wet bulb frosting condition. As shown in Figure 1-80 (a), the SEER2 of the baseline R410A system is 11.42. R-32 and R-454B optimized coils can satisfy the baseline SEER2 criteria. Among the five GWP less than 150 refrigerants, the three fluids which overperform baseline is R-454C, R-457A and R-455A using 5-mm tube heat exchangers. The optimized system using R-1234yf and R-1234ze(E) shows smaller SEER2 after dimension-constrained heat exchanger optimization and compressor sizing.

Figure 1-80 (b) shows the HSPF2 of the baseline and optimal systems. The HSPF2 of the baseline R-410A system is 7.17. R-32 and R-454B can satisfy the HSPF requirement regardless of the choice of heat exchanger tubes. When using 5-mm tube heat exchangers with GWP<150 refrigerants, HSPF2 of the optimized systems exceeds HSPF2 of the baseline.

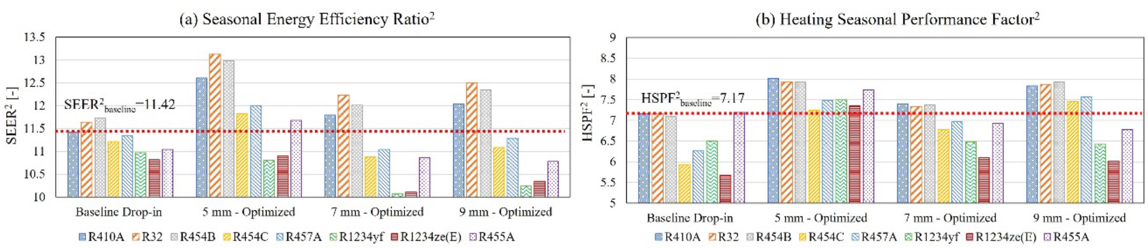

Figure 1-80. Heat Pump Systems Performance Comparison (a) Seasonal Energy Efficiency Ratio; (b) Heating Seasonal Performance Factor

To streamline the performance of different heat exchangers with different low-GWP refrigerants, the SEER2 and HSPF2 are shown in Figure 1-81. The marker shapes represent different system types, i.e., either the baseline system or optimal systems with different diameter tubes. And the marker colors show different refrigerants. The performance of the baseline R410A system (SEER2-11.4/HSPF2-7.2) is highlighted as the cross in the center. By comparing the baseline performance with optimized systems, the design space is divided into 4 regions. The system designs located in the upper right region have both cooling and heating performance superior to the baseline and the system designs located in the bottom left region have dual-mode performance inferior to the baseline. The other two regions have designs with only 1 mode better than the baseline. In the upper right region. The best performances are achieved using R-32 (SEER2-13.1/HSPF2-7.9), followed by R-454B (SEER2-13.0/HSPF2-7.9) and R-410A (SEER2-12.6/HSPF2-8.0). For the GWP <150 fluids, three design candidates using R-457A (SEER2-12.0/HSPF2-7.5), R-454C (SEER2-11.8/HSPF2-7.2) and R-455A (SEER2-11.7/HSPF2-7.7) satisfy both heating and cooling performance requirements. Table 1-20 shows the heat exchanger structures for those optimal systems in Figure 1-81. For brevity, the heat exchanger structure is denoted as *Number of Tubes per Row* x *Number of Rows* - *Number of circuits*. For example, the 7-mm tube indoor baseline heat exchanger in Figure 1-72 has 60 tubes per row, 2 rows and has 6 circuits, so its structure is 60x2-6, and the 9-mm tube outdoor baseline heat exchanger has 24 tubes per row, 1 row and 3 circuits, so its structure is 24x1-3. Following this convention, the heat exchanger structures of the optimized system are listed in Table 1-20.

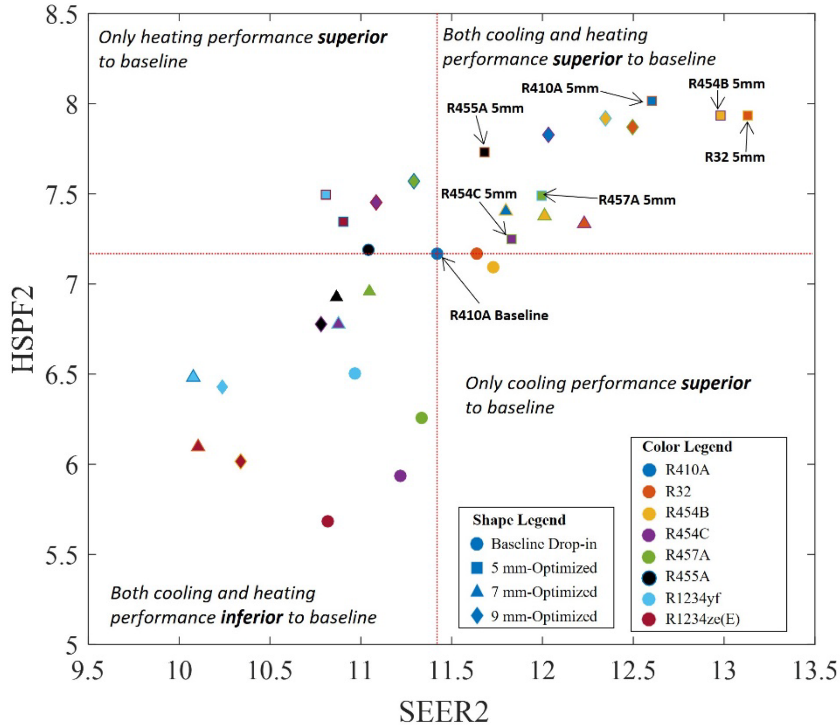

Figure 1-81. SEER2 and HSPF2 of Baseline Drop-in Tests and Optimized Systems

Table 1-20. Heat Exchanger Structure of Optimized Systems

HX Structure	5 mm - Optimized		7 mm - Optimized		9 mm - Optimized	
	Indoor HX	Outdoor HX	Indoor HX	Outdoor HX	Indoor HX	Outdoor HX
R410A	64x3-16	32x2-16	60x2-6	30x2-6	48x2-6	24x1-2
R32	64x3-16	32x2-16	60x2-6	30x2-5	48x2-6	24x1-2
R454B	64x3-16	32x2-16	60x2-6	30x2-10	48x2-4	24x1-2
R454C	64x3-32	32x2-16	60x2-12	30x2-6	48x2-6	24x1-3
R457A	64x3-32	32x2-16	60x2-12	30x2-6	48x2-6	24x1-3
R455A	64x3-32	32x2-16	60x2-15	30x2-6	48x2-8	24x1-3
R1234yf	64x3-32	32x2-16	60x2-15	30x2-6	48x2-8	24x1-3
R1234ze(E)	64x3-32	32x2-16	60x2-15	30x2-6	48x2-8	24x1-3

From Table 1-20, the optimal 5-mm tube indoor HX has two structures depending on the fluids, either 64x3-16 or 64x3-32. The optimal 5-mm tube outdoor HXs, regardless of fluids, have the same structure, i.e., 32x2-16. Compared with the baseline indoor 7-mm tube HX (60x2-6) and baseline outdoor 9-mm tube HX (24x1-3), the number of circuits is significantly increased for optimal HXs using the 5-mm tube. The optimizer increases the number of circuits to distribute refrigerant mass flow such that the refrigerant pressure drop in the heat exchangers can be reduced and the effect caused by the decreased tube cross-sectional area is mitigated.

1.6.7.3.3 Life cycle climate performance analysis

To understand the environmental impacts of optimized heat pumps using the new system configuration and the optimal HX structures with low-GWP refrigerants, life cycle climate performance (LCCP) evaluation is used to analyze the direct and indirect greenhouse gas (GHG) emissions of the system over the course of its lifetime from manufacturing to disposal. It is calculated as the sum of direct and indirect emissions generated over the lifetime of the system “from the cradle to the grave.” Direct emissions include all effects from the release of refrigerant into the atmosphere during the lifetime of the system. Direct emissions include:

- Annual refrigerant loss from gradual leaks
- Losses at the end-of-life disposal of the unit
- Large losses during the operation of the unit
- Atmospheric reaction products from the breakdown of the refrigerant in the atmosphere

The indirect emissions include:

- Emissions from electricity generation
- Emission from the manufacturing of materials
- Emissions from the manufacturing of refrigerants
- Emissions from the disposal of the unit

More details about the LCCP evaluation method can be referred to Wan et al. (2021) [22]. The input values used for evaluating the LCCP are shown in Table 1-21 including the cut-off outdoor temperature and the temperature at which the heat pump starts. A critical part of the LCCP is the assumed leak rate and end-of-life recovery. A 2% leak rate per year with 18 years lifetime and 20% end-of-life recovery are specified based on ASHRAE 189.1. To compare systems using different low-GWP refrigerants, Chicago is selected for a case study and its TMY-3 weather data is used for annual performance evaluation.

Table 1-21. Input Values for Baseline and Optimal Systems LCCP Evaluation

Factor	Value
Refrigerant	R-410A or its alternatives
Refrigerant charge (kg)	As shown in Figure 1-79(a)
Unit weight (kg)	190
Annual refrigerant leakage (%)	2
EOL leakage (%)	80
Lifetime (years)	18
Cut-off temperature (°C)	−17.8
Temperature at which the heat pump starts (°C)	−12.2
Weather data	Chicago TMY-3

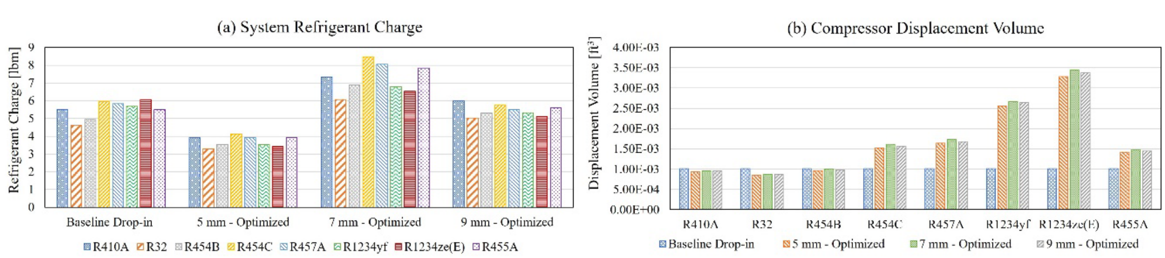

Figure 1-82. (a) System Refrigerant Charge; (b) Designed Compressor Displacement Volume

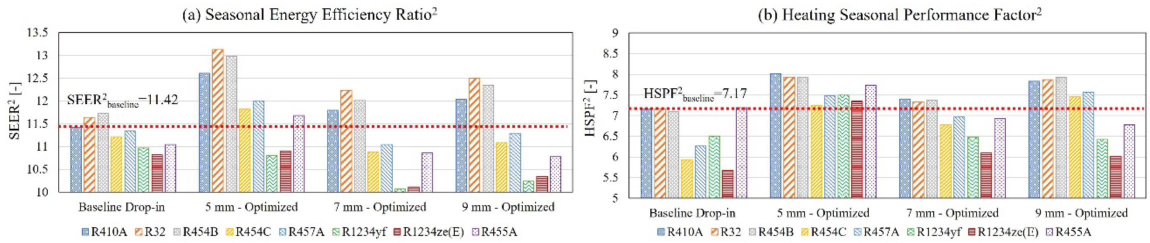

Figure 1-83. Heat Pump Systems Performance Comparison (a) Seasonal Energy Efficiency Ratio; (b) Heating Seasonal Performance Factor

Figure 1-84 shows the comparison of the direct and indirect emissions for systems. Compared to R-410A baseline, systems using other refrigerants reduce direct emissions as shown in Figure 1 86 (a). This attributes to their lower GWP value and the reduced system charge as shown in Figure 1-84(a). Figure 1-84 (b) shows the comparison of the indirect emissions. The optimized systems have 10%-16% lower indirect emissions than the baseline system due to the improved EER (Figure 1-77(a)) and improved COP (Figure 1-76). The 5-mm optimal designs with R454C and R457A induces significant indirect emission reductions. This indirect emission reduction attributes to improved SEER2 and HSPF2 as shown in Figure 1-80 and Figure 1-81. Since Chicago is a city in the heating climate region IV with both a significant number of heating and cooling days, the indirect emissions are affected by both cooling and heating performance.

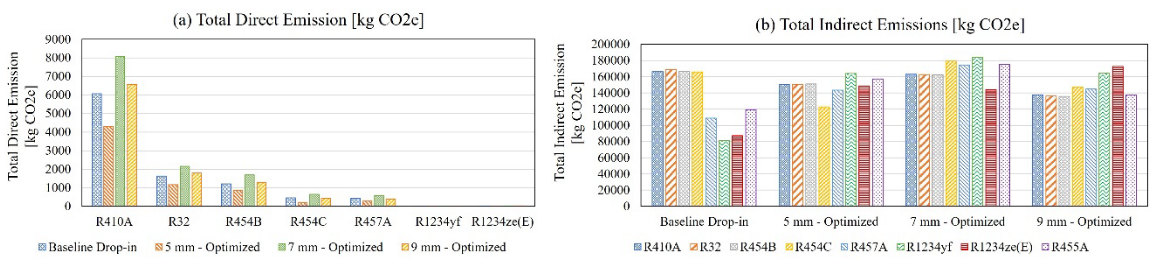

Figure 1-84. Greenhouse gas emissions of the baseline and low-GWP optimized systems (a) Direct Emission; (b) Indirect Emission.

It is worthwhile to mention that the reduction potential is different in another climate zone. For example, in a location with warm or mild climate, the SEER2, i.e., the cooling efficiency plays a more dominant role since the heat pump is mostly operated in cooling mode.

Figure 1-85 shows the comparison of the total emissions. In fact, 96%-98% of the total emission is comprised of indirect emissions. So indirect emission dominates the total emission. Although many designs using 7-mm tube and 9-mm tube heat exchangers show great emission reduction potential, they cannot meet the cooling and heating efficiency requirements as shown in Figure 1-81. The three feasible designs satisfying the efficiency requirements are systems using 5-mm tube heat exchangers with R-454C, R-455A and R-457A. The optimal 5-mm tube systems using R-454C, R-455A and R-457A yields 28.6%, 8.5% and 16.5% lifetime emission reduction compared to the baseline R-410A system.

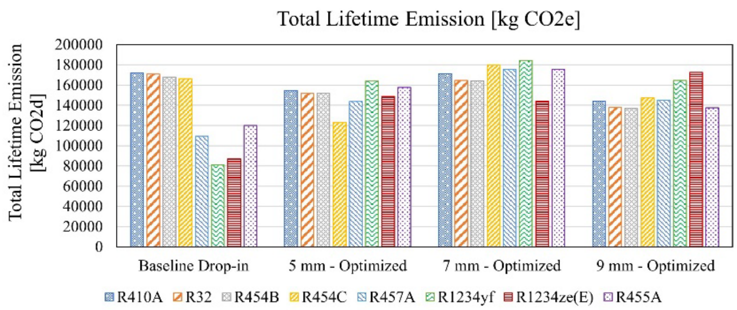

Figure 1-85. Total greenhouse gas emissions of the baseline and low-GWP optimized systems.

1.6.7.4 Conclusion

This study presents heat exchanger and system development technologies to support the transition to refrigerants with GWP lower than 150. Higher than baseline R-410A system efficiency levels in cooling and heating modes are achieved by a model-based design optimization approach based on simulation using detailed hardware information. The new reversible heat pump systems with low-GWP refrigerants adopt a new system configuration, the optimized indoor and outdoor heat exchangers, and an optimized compressor. The potential of 5-mm, 7-mm, and 9-mm tubes in heat exchangers are investigated. The optimal systems using R-454C, R-455A, and R-457A with 5-mm tube heat exchangers outperform the baseline R-410A heat pump product. Life cycle climate analysis shows that the optimized systems using GWP lower than 150 fluids reduce the lifetime CO_2 emissions by 8.5%-28.6% while guaranteeing the same or better capacity and efficiency performance than the baseline.

The optimal heat exchanger designs obtained from this research can fit into the original R410A heat exchanger ducts and chasses, which helps to minimize changes in manufacturing and installation. As a result, the system retrofit impacts on manufacturers and end users are minimized. The proposed design approach establishes a production and installation path to produce cost-effective, high-performance, low GWP reversible heat pumps.

Acknowledgments

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1.6.7.5 References

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1.7 Thermodynamic analysis and carbon emission evaluation for saturation heat pump

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1.7.1 Introduction

Low-temperature heat pumps face several challenges that can affect their performance and limit their potential applications. One of the main challenges is efficiency, as heat pumps typically become less efficient at lower temperatures, requiring more energy to extract heat from the environment. Additionally, low-temperature heat pumps have lower heat output capacity than high-temperature heat pumps, which can limit their ability to heat larger or colder spaces. Frost and ice buildup on the evaporator coils is another challenge that can reduce efficiency and potentially cause damage to the system. Low-temperature heat pumps can also be more expensive to install than conventional heating systems, although they typically offer long-term energy savings. Furthermore, low-temperature heat pumps may not be compatible with all heating systems or building designs, which can limit their potential applications. Lastly, low-temperature heat pumps may not be able to produce enough heat in extremely cold temperatures, which can require the use of backup heating systems or other heating sources. These challenges need to be considered carefully when choosing and designing low-temperature heat pump systems to ensure their optimal performance and efficiency.

The state-of-the-art (SOA) R410A HP systems face severe capacity and efficiency degradation under extremely low temperatures due to inherent thermodynamic drawbacks. For instance, under -15°F (-26°C), the COP of the HP system decreases to 2.3 with a capacity degradation of more than 30% [45]. The reasons for such lower efficiency and high capacity degradation come from 1) compressor inefficiency under a high-pressure ratio; 2) condenser inefficiency with the higher superheated vapor phase ratio; 3) selection of suitable low-GWP refrigerant and high enough critical temperature. Figure 1-86 a) describes an HP system operated under -26°C evaporation and 35°C condensing conditions. The red solid line in Figure 1-86 a) represents a conventional single-stage HP design. Typical compression ratios of single-stage mechanical gas compressors are around 2.5 to 4. However, the required total pressure ratio (PR) to achieve a high-temperature lift as high as 61 K is 8.5. Multiple stage numbers (with a smaller PR) and different superheat degrees are used to illustrate the challenges of this specific high-temperature lift on the HP system performance.

We propose a set of technologies that aim to boost both COP and heating capacity for commercial rooftop heat pumps. The saturation cycle with two injections alone leads to a heating COP of 2.7 and capacity degradation of 20% under -15°F. With an additional novel implementation of safety control, the smart CCHP system will deliver a COP of 2.7 and capacity degradation of only 12% at -15°F while supplying an air discharge temperature higher than 95°F. The proposed technology leads to cost reduction compared to competing systems that need oversized and offers a payback period of 4.3 years for a system with 10 ton capacity of a standard RTU.

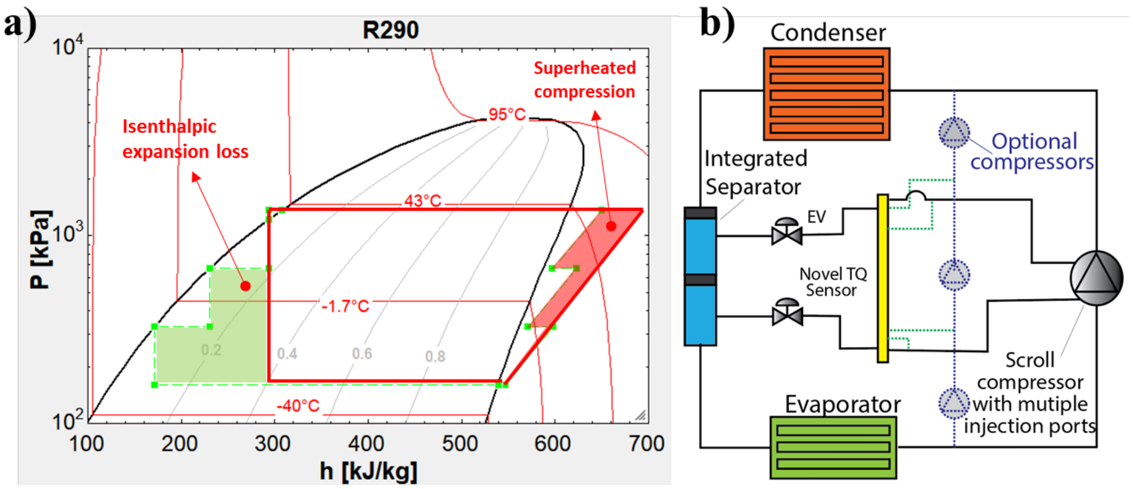

Figure 1-86: Saturation heat pump cycle: a) P-h diagram and energy loss through expansion and superheated compression; b) realization of a saturation HP system via double injections

1.7.2 Cycle description

Our previous studies have demonstrated the superior thermal performance of the saturation HP system with low-GWP working fluids for residential heat pumps [46, 47]. The COP and capacity improvement can reach 45% over non-saturation cycles. The recent development by Emerson on double injection (three stages) compressors makes the concept of saturation HPs ever closer to be commercially feasible. Figure 1-87. 10a-c succinctly shows the thermodynamic principles of saturation heat pumps. As the number of injections increases, the two major irreversible losses, throttling loss and superheated compression (indicated in Figure 1-86 a)) are reduced. Therefore, the cycle efficiency gradually approaches the theoretic Carnot limits. For example, under -26°C ambient temperature and 35°C air discharge temperature scenario, a two-stage cycle reaches COP of 2.5 and heating capacity degradation of 16%; a three-stage cycle reaches COP of 2.7 and heating degradation of 12%; and a four-stage cycle reaches COP of 2.8 and heating degradation of 10%.

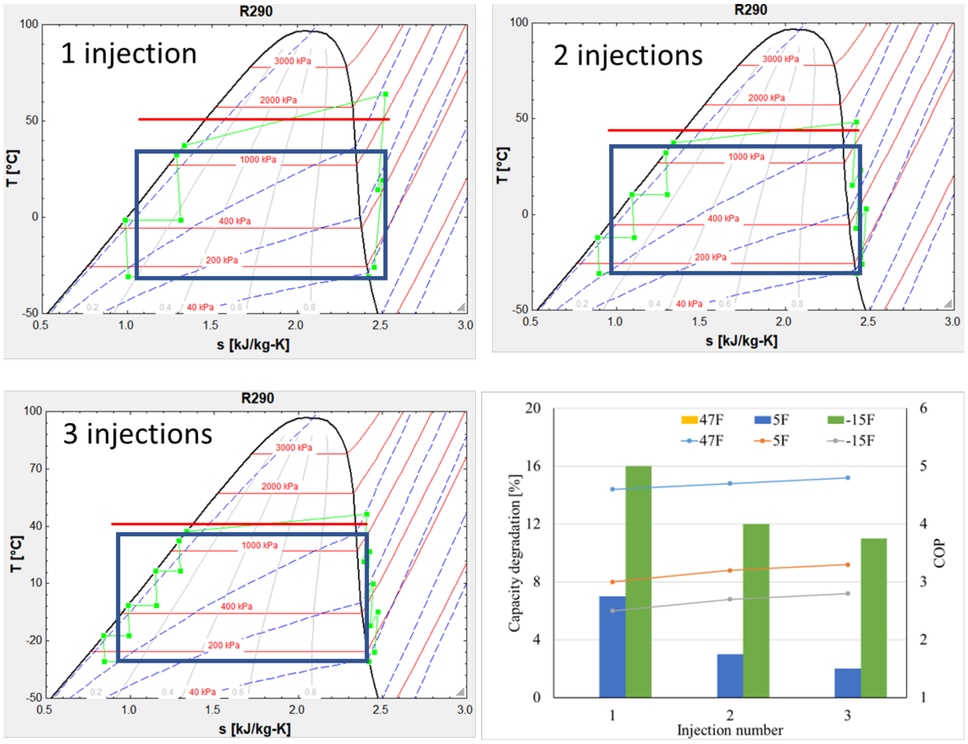

Figure 1-87: a)-c) a higher equivalent condensing temperature achieved by higher injections; d) COP and capacity degradation on different injection numbers.

Figure 1-88 a) shows how the compressor work decreases along with stage number (PR decreases sharply). The energy saving from the compressor can be easily obtained from a smaller PR and staging multiple smaller PR compression processes to deliver a sufficient temperature lift. A 15% of energy saving can be achieved for a PR lower than 3. It should be noted that 5K superheat is assumed for the compression efficiency simulation. More energy-saving potential can be obtained by optimizing the compressor design. This process can be accomplished with CEEE's established compressor modeling tools. The degree of superheating at the compressor outlet is another critical factor that should be carefully designed when it comes to an hight pressure ratio HP, as it could result in excessive compressor power and failure. It is observed from the simple cycle simulation that the compressor outlet temperature of the single-stage cycle can reach around 85°C. By adopting a low-GWP refrigerant as a working fluid (eg. R290) for a heat sink temperature of 35°C with a evaporating temperature of -26°C. Figure 1-88 b) shows the amount of heat delivered by a superheated vapor in terms of superheat increase. The mass flow rate of the refrigerant decreases with the increase in overall heating capacity due to higher superheating. However, the heat transfer area of the condenser increases almost linearly with a degree of superheating since the heat transfer coefficient of the two-phase region is one order higher than that of the superheated vapor.

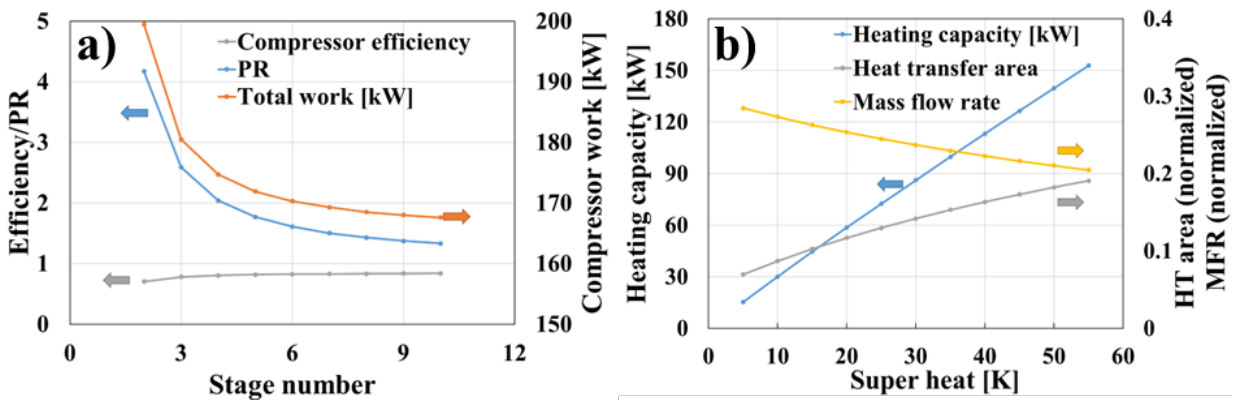

Figure 1-88: Challenges for a conventional HP system a) stage number influence on compression efficiency; b) Superheat influence on HX performance

One of the most critical design directions for the cold climate system is to reduce the excessive pressure ratio and the degree of superheating to their optimal values. To achieve this, the compression process can be divided into multiple stages with multiple refrigerant injections. Therefore, the design should incorporate multiple injection port locations to satisfy the requirements of the various compression stages. By doing so, the system can improve its overall efficiency and performance while reducing energy consumption and operating costs.

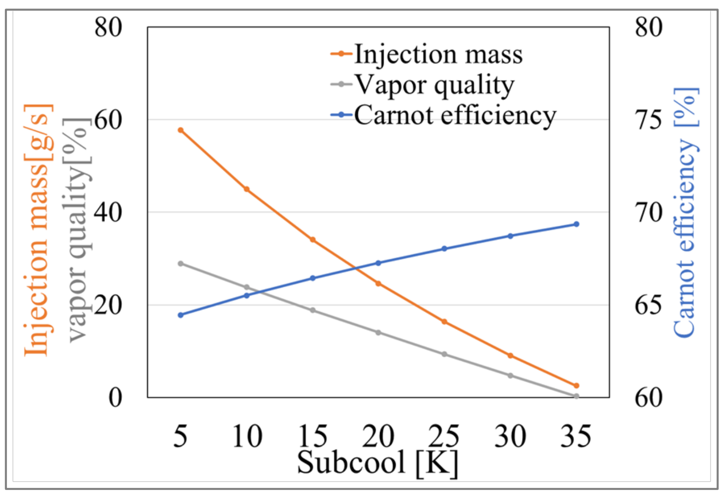

Figure 1-89. Subcool influence on system performance

To optimize the system efficiency, it is important to consider the subcooling degree of the last compression stage, as indicated in Figure 1-89. A lower subcooling degree leads to higher vapor quality after the last expansion, which in turn requires more two-phase refrigerant to satisfy the same superheat at the last compression inlet. While increasing the condensing capacity, this also results in a higher power consumption for the last compression process. On the other hand, a higher subcooling degree is preferred to improve system efficiency, but excessive subcooling can have a negative impact on condenser performance. For the simulation work, a subcooling degree of 5K was adopted, and further optimization is needed to determine the most suitable subcooling degree for optimal system performance.

1.7.3 CO_2 emission analysis

Around one-third of global energy- and process-related CO_2 emissions are directly and indirectly attributed to the buildings sector. In 2021, approximately 8% of these emissions resulted from the use of fossil fuels in buildings, while 19% resulted from the generation of electricity and heat used in buildings. An additional 6% was related to the manufacturing of construction materials such as cement, steel, and aluminum. Therefore, it is crucial to address the impact of buildings and construction on CO_2 emissions by imposing emission restrictions throughout the entire value chain. These dedicated saturation HP systems can offer a total energy saving of 1.51 Quads, as the solid bar plot in Figure 1-90, which is 85% and 21% more energy efficient than fossil fuel-based systems and SOA IHP systems. As a result, the saturation system can reduce 62.4 MMT of CO_2 (stripped bar plot in Figure 1-90) if switching from conventional system. The energy saving will defeat the SOA efficiency by 28% (black line chart). This carbon emission saving is equivalent to that of the energy consumption by 15.6 million U.S. homes. The payback period will decrease as the injection number increases at the beginning and slightly increases when too many injection ports are applied. This payback trend is caused by the requirement of multiple compressors and the challenge of manufacturing multi-port on a single compressor. Considering the on-shelf injection compressors and optimized low-cost heat exchangers, the proposed saturation system concept will still offer a payback period of 2.3 years (blue line chart in Figure 1-90).

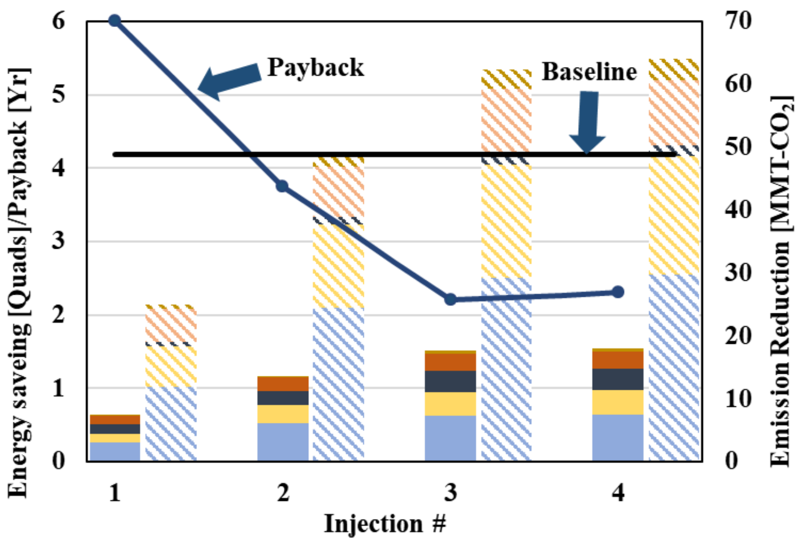

Figure 1-90: Energy, emission, and payback performance

1.8 Enhanced Summary: Optimization of Air-to-Refrigerant Heat Exchanger Shape & Topology for Low GWP Refrigerants

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As nearly all HVAC&R systems utilize air-to-refrigerant and liquid-to-refrigerant Heat eXchangers (HXs) as the main heat transfer components, improved HX performance is of utmost importance to improving overall HVAC&R system performance. Specifically, compact HXs, i.e., those with large heat transfer area to envelope volume ratio, have the potential to increase energy efficiency while reducing environmental impact. Specifically, finless HXs with small diameter tubes (e.g., <5.0 mm) have been shown to be more compact, utilize less refrigerant and material, and potentially outperform finned HXs. Moreover, Westphalen et al. noted that heat transfer enhancement in the form of doubling the condenser airside heat transfer coefficient could reduce cycle energy consumption by 10-15%, which would, in turn, reduce building energy consumption by 4.6-6.9%. HX modeling and optimization is a key component of the design process, allowing researchers to investigate novel HX geometries and configurations prior to prototyping and experimentation, thus saving significant resources. Moreover, advancements in computational tools such as Computational Fluid Dynamics (CFD), Finite Element Analysis (FEA), optimization algorithms/techniques such as multi-objective genetic algorithms (MOGA) and Approximation-Assisted Optimization (AAO), along with improvements in advanced manufacturing such as Additive Manufacturing (AM) have led to a paradigm shift in HX design ideology.

This research sheds light on practical issues in HX modeling, design, optimization, and commercialization such as novelty challenges (improved performance for significant charge reduction; significant modeling expertise & time investment), manufacturing challenges (product qualification, e.g., burst pressure testing, operation in extreme environments, etc.), and operational challenges (flow maldistribution, frosting, fouling & wetting, noise & vibration). For example, a >20% improvement on one (or more) HX-level performance metrics (e.g., envelope volume, airside pressure drop, face area, capacity, refrigerant charge, weight, and cost, to name a few) must be achieved before an HX design is considered for commercialization. Continuing with the previous year; we summarized the major findings in Table 1.

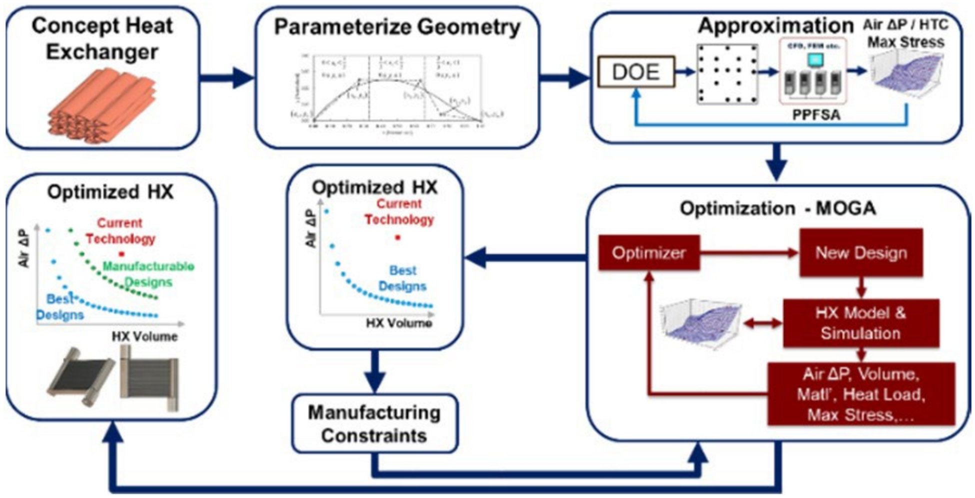

Figure 1-91 Numerical optimization framework.

1.8.1 Methodology

1.8.1.1 Design Optimization Framework

The HX design optimization framework (Figure 1-91) utilizes Approximation-Assisted Optimization (AAO) [20] involving automated Computational Fluid Dynamics (CFD) [21] and Finite Element Analysis (FEA) simulations [10-11], Kriging metamodels [22], and optimization with a multi-objective genetic algorithm (MOGA) [23].

1.8.1.2 Problem Description

High performance HXs featuring novel, non-round tubes have been investigated for many applications. In this research, the optimization framework is utilized to design novel, finless, air-to-refrigerant condensers for residential A/C applications using lower-GWP alternative refrigerants. The baseline HX is a commercially- available, state-of-the-art nominal 5.28 kW (1.5-Ton) tube-fin air-to-R410A condenser. Three refrigerants are investigated: (i) R410A, (ii) R32, and (iii) R454B. The inlet conditions for each refrigerant are taken from commercially available residential A/C units with nominal 5.28 kW (1.5-Ton) capacity. Sample schematics for a representative tube-fin HX and a finless HX with shape-optimized tubes are shown in Figure 1-92.

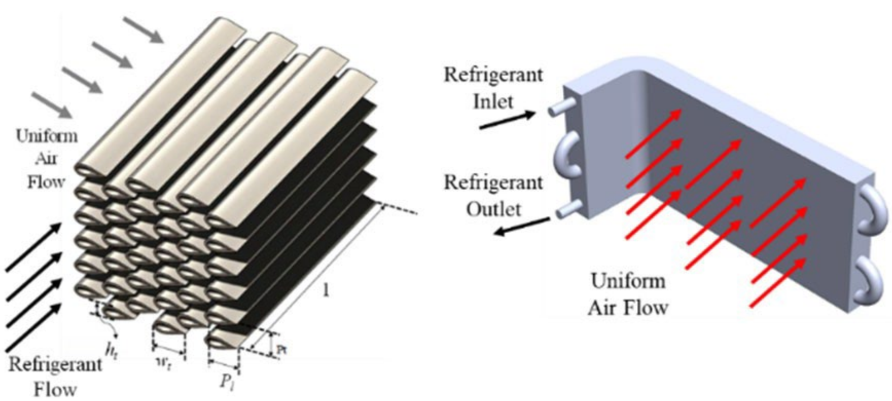

Figure 1-92. (Left) Generic HX with shape-optimized tubes; (Right) Generic multi-pass tube-fin condenser.

The optimization framework was applied to multiple design problems (Table 1) with a wide variety of refrigerants, including conventional refrigerants (R410A), next-generation replacement refrigerants (R32, R454B), and low-GWP and natural refrigerants (R290, supercritical carbon dioxide (sCO_2)), and applications (A/C system, heat pump system, gas coolers). It is clear that the optimization framework is capable of designing HXs utilizing non-round, shape- and topology-optimized tubes which show significant improvements over state-of-the-art baseline HXs across a wide variety of HX geometric and thermal-hydraulic performance metrics, including satisfying the objectives of >20% smaller and >20% improved thermal-hydraulic performance. Of all cases considered, only one optimization did not result in HXs which completely outperformed the baseline across all metrics (face area for sCO_2 gas cooler with a microchannel HX baseline). In this case, it was noted that the HX core envelope volume reductions were due to large HX depth reductions resulting from the small tube size, and some potential methods to find additional designs with reduced face areas would be to allow for more tube banks in the depth-wise direction and/or to consider a HX with multiple slabs. The generality of the optimization framework developed herein allows such considerations and should be explored in the future to find HX designs with even greater improvements beyond those listed in Table 1-22.

Table 1-22: Summary of completed HX optimization studies (with References).

Reference	Optimization Study	Application	Tube Shape	Air ΔP	Core Envelope Volume	Face Area	Core Internal Volume
Tancabel et al.	R410A Condenser (A)	Nom. 5.28 kW A/C system	NTHX1	43%↓	47%↓	31%↓	31%↓
	R410A Condenser (B)	Nom. 5.28 kW A/C system	Variable	46%↓	90%↓	27%↓	83%↓
Tancabel et al.	R410A Condenser (C)	Nom. 5.28 kW A/C system	NTHX1	62%↓	53%↓	34%↓	43%↓
Internal Study (A)	R410A Evaporator	Nom. 5.28 kW A/C system	NTHX1	82%↓	68%↓	16%↓	70%↓
Internal Study (B)	R410A Indoor Unit HX (A)	Dual-mode heat pump	NTHX1	62%↓	N/A	40%↓	N/A
	R410A Indoor Unit HX (B)	Dual-mode heat pump	Variable	77%↓	N/A	37%↓	N/A
Tancabel et al.	R32 Condenser	Nom. 5.28 kW A/C system	NTHX1	47%↓	57%↓	50%↓	44%↓
	R454B Condenser	Nom. 5.28 kW A/C system	NTHX1	63%↓	47%↓	34%↓	41%↓
Tancabel et al.	R290 Condenser	Nom. 2.4 kW A/C system	NTHX1	43%↓	69%↓	14%↓	49%↓
Tancabel et al.	sCO_2 Gas Cooler (A)	FTHX Baseline	Variable	N/A	74%↓	7%↓	74%↓
	sCO_2 Gas Cooler (B)	MCHX Baseline	Variable	79%↓	85%↓	133%↑	73%↓

1.8.2 Summary

In summary, the framework was utilized to develop novel HXs with a wide variety of refrigerants, including conventional refrigerants (R410A), next-generation replacement refrigerants (R32, R454B), and low-GWP and natural refrigerants (R290, supercritical carbon dioxide (sCO2)), and applications (A/C system, heat pump system, gas coolers). These HXs consistently met the following design targets compared to commercially available, high-performance HXs for stationary systems: >20% weight reduction,>20% envelope volume reduction, and >25% refrigerant charge reduction.

1.9 Comprehensive review of life cycle climate performance (LCCP) for air conditioning systems

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The climate is changing faster than predicted, with self-reinforcing feedback loops in the climate system that risk pushing the planet past irreversible and catastrophic tipping points (The Climate Reality Project, 2020). Air conditioning (AC) systems are both one of the main contributors to global warming (Yang et al., 2021) and potentially part of the solution. The challenge is to drastically increase energy efficiency even as electric supply shifts from fossil fuel to wind, solar, and hydro-electric sources, and to simultaneously shift from high global warming potential (GWP) refrigerants to lower ones even as leakage rates are minimized and refrigerants are fully recovered at end of (product) life (EOL).

A holistic evaluation of an AC system's global warming impact during its life cycle needs to account for direct refrigerant greenhouse gas (GHG) emissions, indirect fossil fuel GHG emissions, and embodied equipment emissions (Andersen et al., 2018). The Life Cycle Climate Performance (LCCP) concept was conceived by Andersen (1999) to calculate the "cradle to grave" climate impact of direct, indirect, and embodied GHG emissions. Later, the Society of Automotive Engineers (now rebranded as SAE International) developed an LCCP model for automotive/mobile air conditioning (MAC) systems (Hill and Papasavva, 2005), and the Institute of International Refrigeration (IIR) developed the LCCP evaluation guideline for stationary AC systems. The IIR's LCCP guideline adopted a rigorous approach to identifying and quantifying the direct and indirect environmental impacts of a MAC system over a stated life cycle. The LCCP has evolved over the past decades, as shown in Figure 1-93. Other environmental evaluation metrics existed for AC systems before the LCCP. Global warming potential (GWP), a famous ecological metric for refrigerants, only accounts for the direct emission from the refrigerant (Makhnatch and Khodabandeh, 2014). Total Equivalent Warming Impact (TEWI) is the summation of carbon-equivalent direct refrigerant and indirect power generation GHG emissions (Sand et al., 1997). LCCP, a more comprehensive metric, was proposed by adding carbon-equivalent and embodied emissions. Life Cycle Assessment (LCA) is a broad classification of systematic tools to evaluate a product's life cycle environmental impact, including TEWI and LCCP (Hwang, 2013). LCA is also widely used in other fields, like building and aviation sectors s (Bachmann et al., 2017; Vilches et al., 2017). Therefore, LCA is more formal and comprehensive in comparing the holistic environmental impact of technologies, but LCCP focuses more on climate impact (Andersen et al., 2005). Recently, some re- searchers came up with a new concept: enhanced-LCCP (EL-LCCP) (Andersen et al., 2018), which considers local climate, heat islands, and local power supply characteristics.

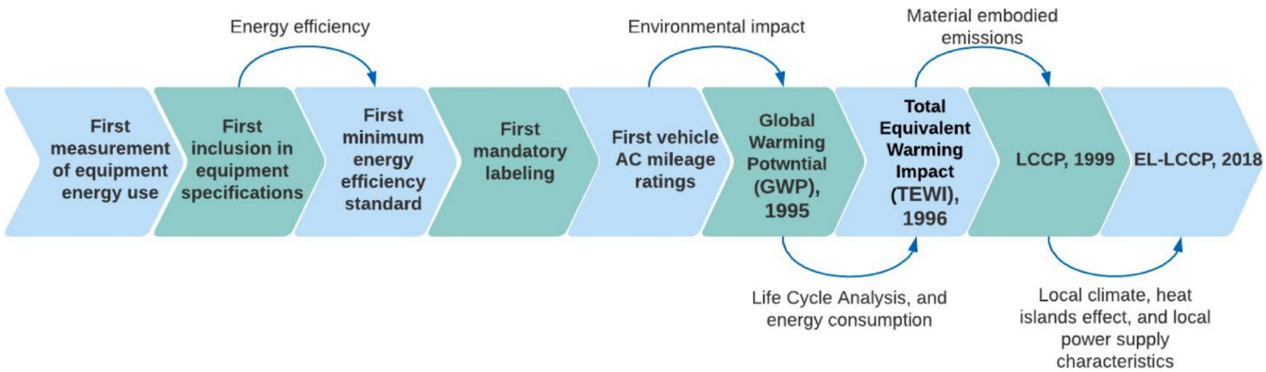

Figure 1-93. Invention and evolution of LCCP.

LCCP has had some successful applications. In 2009, SAE International approved standard J2766-200902 "GREEN-MAC-LCCP" (SAE International, 2009), which was the basis for the industrial decision to shift from HFC-134a to HFO-1234yf rather than to other refrigerant candidates available at that time (Andersen et al., 2013). Horie et al. (2010) assessed the LCCPs of residential heat pumps in Japan. Zhang et al. (2011) developed an LCCP tool for a residential heat pump for four United States (US) cities. Li (2015a) evaluated various Packaged Air Conditioners (PAC) involving microchannel heat exchangers for typical US cities. Troch (2016) and Lee et al. (2016) conducted an LCCP evaluation for the same heat pump system in five US cities. Choi et al. (2017) developed an LCCP model and evaluated it for South Korean weather conditions. Wu and Jiang (2017) developed LCCP-calculation software to analyze different climate regions in China. Botticella et al. (2018) establish a multi-objective thermo-economic and environmental opti- mization of 5 kW residential space heating split system using LCCP. Kim et al. (2018) applied a Neural Network algorithm to predict LCCP value using three different US weather conditions. A new application of LCCP analysis in domestic refrigerators was reported (Aprea et al., 2016).

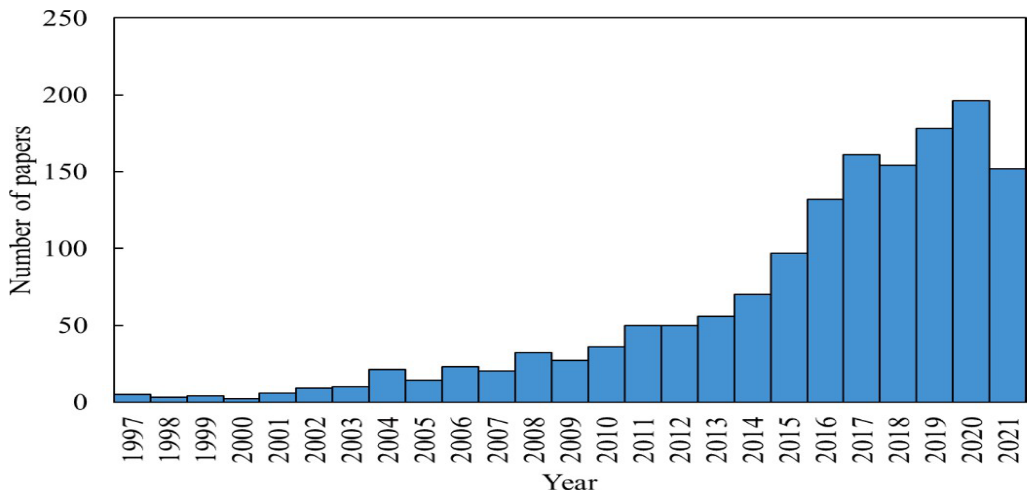

Figure 1-94. Histogram of annual life cycle assessment publications for AC and HP systems (ScienceDirect, assessed 2021).

AC systems' life cycle assessment has grown steadily over the last 20 years and has become a mainstream topic, but the comprehensive LCCP metric isn't always used. Figure 1-94 shows the number of papers reported on ScienceDirect using the keywords "life cycle assessment" for "heat pump" and "air conditioning." Among the 1,508 papers, only 84 papers mentioned the LCCP concepts. Makhnatch and Khodabandeh (2014) reviewed LCCP, TEWI, and GWP publications in 2014, but mainly focused on different calculations of carbon-equivalent refrigerant effect. Chau et al. (2015) and Sharma et al. (2011) reviewed LCA for buildings, but although the AC systems were mentioned in their work, they did not analyze AC equipment in detail. Mota-Babiloni et al. (2020) discussed the utilization of equivalent warming impact metrics in refrigeration, air conditioning, and heat pump systems. They concluded that the TEWI metric had been more widely adopted than LCCP and provided the TEWI metric application recommendations. Nevertheless, they ignored the material and refrigerant embodied emissions.

This subsection: 1) reviews the invention and evolution of LCCP, including a comprehensive timeline and bibliography; 2) documents the successful application of LCCP in the replacement of HFC-410A with HFC-32 in room air conditioners; 3) compares the conceptual frame- works and the operational approaches; and 4) reflects on the drawbacks of current LCCP research. Our study could be considered as a supple- ment to the guideline. In the IIR LCCP guideline, the recommendation values were based on North American regions, and some assumptions used in the guideline were not realistic enough. Our study will review how these assumptions have been handled in the latest literature. For the current research gap, we will provide our suggestions on future works

Table 1-23: Recent LCCP evaluation research

Author (year)	System	Refrigerant	Country
Horie et al. (2013)	1.3 kW HP	R-410A, R-32, R-1234yf	Japan
Zhang et al. (2014)	11 kW HP	R-410A, R134a, R-1234yf	US
Li (2015)	13, 14 kW AC	R-410A, R-22	US
Troch et al. (2016)	11 kW HP	R-410A	US
Lee et al. (2016)	11 kW HP	R-410A, R-32, R-290, DR5, L41, D2Y60	US
Choi et al. (2017)	11 kW VI HP	R-410A, R-32, R-290	Korea
Wu and Jiang (2018)	-	R-410A	China
Kim et al. (2018)	12.4 kW VI HP	R-410A	US

1.9.1 LCCP Calculation Process

Troch et al. (2016) wrote IIR's guideline (2016) summarizing the calculation process of LCCP. The symbols in Troch's work were adopted in this paper. LCCP consists of direct and indirect emissions and is typically calculated in kg CO_{2e} unit, as shown in eq. 1:

$$LCCP = Direct\ emissions + Indirect\ emissions \tag{1}$$

Direct emissions are the refrigerant emissions during the usage phase in the equipment's lifetime and end of life (EOL) phase. Direct emissions can be calculated by eq. 2:

$$Direct\ emissions = C \times (L \times ALR + EOL) \times (GWP + Adp.GWP) \tag{2}$$

Where C means a refrigerant charge (kg); L means average life of the equipment (yr); ALR means annual leakage rate (percentage of refrigerant charge); EOL means End of Life refrigerant leakage (percentage of refrigerant charge), GWP means Global Warming Potential (kg CO_{2e}/kg), Adp.GWP means GWP of Atmospheric Degradation Product of the Refrigerant (kg CO_{2e}/kg).

Indirect emissions include emissions from the power plants by consuming electric power for the equipment operation, manufacturing of materials, manufacturing of refrigerant, and disposal of the unit, as shown in eq. 3:

$$\begin{aligned} Indirect\ emissions = L \times AEC \times EM + \sum(m \times MM) + \sum(mr \times RM) \\ + C \times (1 + L \times ALR) \times RFM + C \times (1 - EOL) \times RFD \end{aligned} \tag{3}$$

Where AEC means Annual Energy Consumption (kWh); E.M. means CO_2 produced/kWh (kg CO_{2e}/kWh), which is the Grid Emission Factor (GEF) if electricity is the only energy source; m means a mass of unit (kg); MM means CO_2 Produced/Material (kg CO_{2e}/kg), which is also known as ECC; mr means the mass of recycling material (kg); R.M. means CO_2 produced/ recycled material (kg CO_{2e}/kg); RFM means refrigerant manufacturing emission (kg CO_{2e}/kg); RFD means refrigerant disposal emissions (kg CO_{2e}/kg); L, C, ALR, and EOL have the same meaning as the ones in eq. 2. Figure 1-95 shows the relationship between each sub-emission of LCCP.

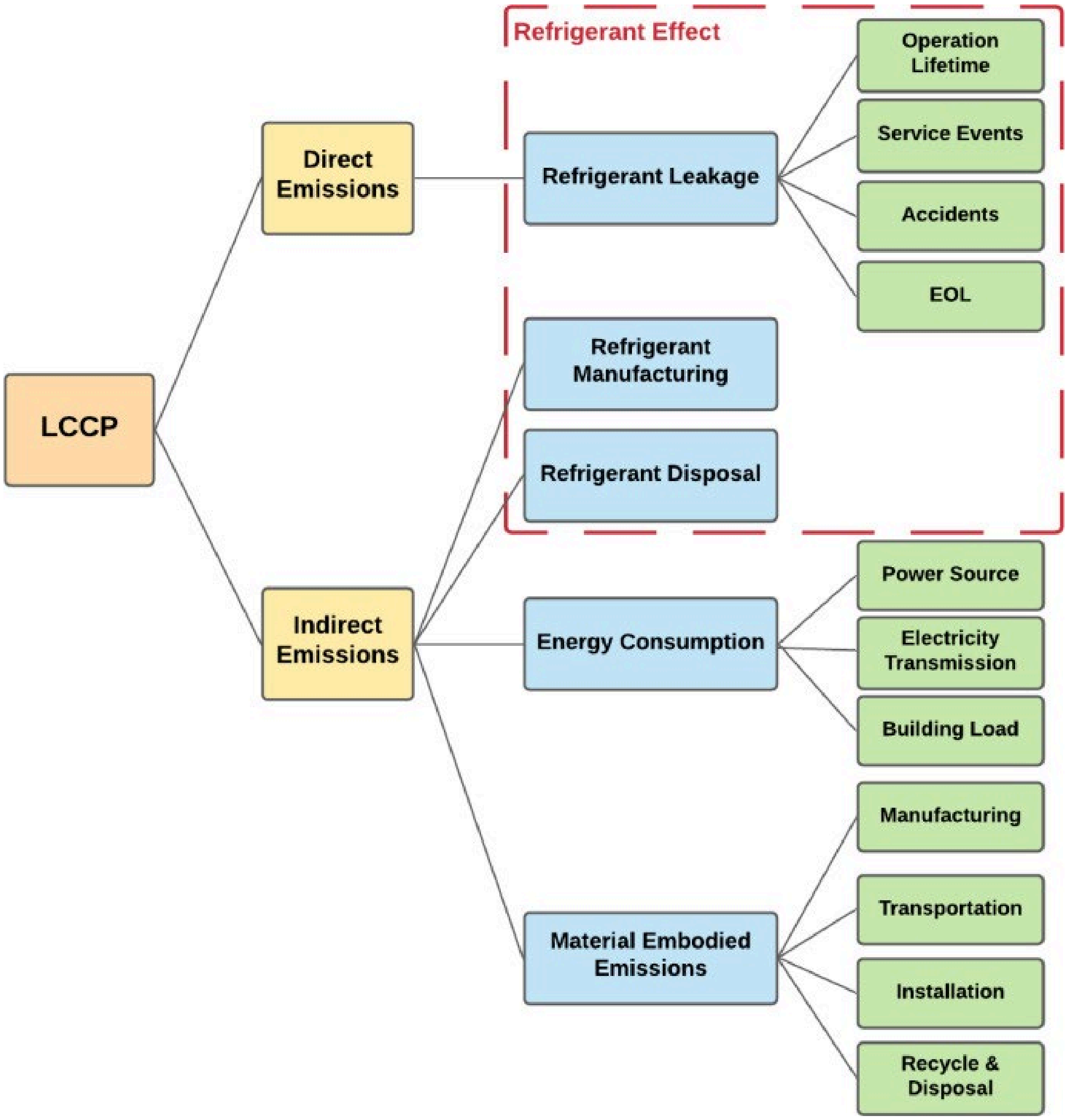

Figure 1-95. Carbon emissions category

1.9.2 Refrigerant effects

Table 1-24 shows some common refrigerants and their properties. The refrigerants used in AC systems have evolved over several decades from ozone-depleting hydrochlorofluorocarbon (HCFC) or refrigerant (R)-22 GHG to chlorine-free/ozone-safe hydrofluorocarbon (HFC) R-410A GHG and now to lower-GWP R-32. The US Environmental Protection Agency (EPA) reported that current AC systems using R-410A still contribute 36.7 million metric tons of carbon dioxide equivalent to the atmosphere (EPA, 2015). Selecting an appropriate alternative refrigerant for an AC system involves a trade-off among heat transfer characteristics, compressor efficiency, flammability, and cost (Gilmour and McNally, 2010). LCCP can assist in selecting alternative working fluids that reduce GHG emissions from AC systems. Since only certain nations use R-22 nowadays, this review used R-410A as the baseline. We discuss several alternatives to R-410A based on the refrigerant safety group, namely A1, A2L, A3, and B groups according to ASHRAE Standard 34, (2019). The refrigerant leakage, emissions associated with the refrigerant manufacturing process, and emissions from disposal of the refrigerant can affect LCCP calculation. These effects will be discussed in detail below.

Table 1-24 Refrigerants of interests

Safety class	Refrigerant	GWP[3] (kg CO_2e/kg)	Adp.GWP (kg CO_2e/kg)	Concerns	Introduced Year	Reference
A1	R-410A	2,088	-	High GWP	Honeywell, 1991	(Goto et al., 2001; Wang et al., 2009)
A1	R-466A	730	-	High cost	Honeywell, 2018	(Cooling Post, 2018a, 2018b; Deveciog˘luand Oruç, 2020)
A1	R-1234yf	< 1	3.3	Low pressure	DuPont, 2011	(Baral et al., 2013; Myhre et al., 2014)
A1	R-134a	1,300	1.6	High GWP	DuPont, 1930[2]	(Baral et al., 2013; Myhre et al., 2014)
A1	R-404A	3,943	-	High GWP	-	(ASHRAE Standards, 2019)
A1	R-22	1,760	-	High GWP Mild ODP	General Motors, 1928[1]	(ASHRAE Standards, 2019)
A2	R-32	675	-	Mildly Flammable	Japan Ministry of Economy, Trade, and Industry (METI), 2011	(Mota-Babiloni et al., 2017; Pham and Rajendran, 2012; Xu et al., 2013)
A2	R-452B	676	-	Mildly Flammable	Ingersoll Rand, 2015	(Kedzierski and Kang, 2016; Kujak et al.,2014)
A3	R-290	< 1	-	Highly Flammable	-	(Wu et al., 2012)

1.9.2.1 Refrigerant leakage

The main factors affecting direct emissions are the annual leakage rate (ALR) of the refrigerant, the GWP of the refrigerant, and the GWP of the Atmospheric Degradation Product (ADP) of the refrigerant (ADP.GWP). Direct emissions are the refrigerant emissions during the usage time in the lifetime and EOL process. By expanding the terms in Eq. 3, direct emissions can be calculated by Eq. 4:

$$Edirect = C \times (L \times ALR + EOL) \times (GWP + ADP.GWP) \qquad (4)$$

where C means a refrigerant charge (kg); L means average life of the equipment (yr); ALR means annual leakage rate (percentage of refrigerant charge); EOL means End of Life refrigerant leakage (percentage of refrigerant charge); GWP means Global Warming Potential (kg CO2e/kg); ADP.GWP means the GWP of Atmospheric Degradation Products of the refrigerant (kg CO2e/kg). GWP has been widely reported in the literature (Bobbo et al., 2018). C is typically marked on the product label.

1.9.2.2 Global Warming Potential

Global Warming Potential (GWP) of a refrigerant is an index to compare the relative radiative forcing of different gases relative to the reference gas CO2, which is set equal to 1. A significant number of researchers have studied the GWP of refrigerants (Bobbo et al., 2018; Lin and Kedzierski, 2019). However, ADP.GWP was rarely mentioned. Although previous research has pointed out that Adp.GWP needed to be considered in LCCP calculations (Yang et al., 2021; Andersen et al., 2018; Choi et al., 2017), only Baral et al. (2013) reported the ADP.GWP values for R-1234yf and R-134a. Yang et al. (2021) used zero (0) for R-410A and R-32 but didn't mention the reference or reason. Thus, more work is needed to perfect ADP.GWP values.

Table 1-25 Recommended ALR, EOL, and L values (Troch, 2016).

System Type	ALR (%)	EOL (%)	L (yr)
Residential Packaged AC Units	2.5	15	15
Residential Split AC Units	4	15	15
Packaged Refrigeration System	2	15	15
Supermarket Direct AC System	18	10	7–10
Supermarket Indirect AC System	12	10	7–10
Commercial Refrigeration System	5	15	15
Commercial Packaged AC Units	5	15	10
Commercial Split AC Units	5	15	10
Chillers	5	15	15
Marine	20	15	15

1.9.2.3 Unit lifespans, annual leakage rate, and EOL leakage rate

The most accurate values for L, EOL, and ALR in stationary AC applications can often be obtained from the manufacturers. Average unit lifetimes can be taken from IPCC AR4 or AR5 reports or from the Montreal Protocol's Refrigeration, Air Conditioning and Heat Pumps Technical Options Committee's 2002 report (Troch, 2016). The US limits the amount of refrigerant released from an appliance to 15% of the total charge for units with a charge of 22.7 kg or less (Troch, 2016). Troch summarized the recommended value for L, EOL, and ALR for different types of AC systems (Troch, 2016). Table 1-25 shows these values for stationary applications. It is worth noting that SAE International Standard J-2727 (2012) prescribes a method to estimate the leakage rate based on a standard configuration while taking into account fittings, seals, and refrigerant permeation of flexible hoses.

1.9.2.4 Refrigerant manufacturing

Refrigerant leakage contributes to direct emissions, while the refrigerant manufacturing and refrigerant disposal process are related to indirect emissions from the energy used to recover, transport, and recycle or destroy the refrigerant. The leakage in these two processes is counted in the leakage calculation in section 3.2. The emissions we discuss here are from the energy usage in these two processes. The Eref, man can be calculated by eq. 5:

$$Eref\ ,man\ =\ C\ \times\ (1\ +\ L\ \times\ ALR)\times\ (1\ -\ R)\times\ RFM \quad (5)$$

where RFM means refrigerant manufacturing emission (kg CO_{2e}/kg); R means the fraction of the refrigerant in the system, which is reclaimed refrigerant. Refrigerant manufacturing emissions rates are shown in Table 1-26. They will need to be updated as more efficient methods of manufacturing are developed. So far, manufacturing emissions for R-466A have not been reported.

Table 1-26 Refrigerant manufacturing emission of refrigerants of interests.

Refrigerant	RFM (kg CO_2e/kg)	Reference
R-410A	10.7	(Spatz and Motta, 2004)
R-466A	n/a	n/a
R-32	7.2	(Spatz and Motta, 2004)
R-452B	8.9	(Troch, 2016)
R-290	0.05	(Hill and Papasavva, 2005)
R-1234yf	13.7	(Hill and Papasavva, 2005)
R-134a	5	(Banks and Sharratt, 1996)
R-404A	16.7	(Papasavva et al., 2010)
R-22	390	(Chen, 2008)

1.9.2.5 Refrigerant disposal

Similarly, Eref, EOL can be calculated by eq. 6, where RFD means refrigerant disposal emissions (kg CO_{2e}/kg). This process also includes the recovery of the refrigerant. Although RFM values were widely reported, as listed in Table 1-26, almost no LCCP study stated RFD values. This may explain why almost all the literature mentioned emissions from the refrigerant disposal process but did not count them in their calculation.

$$E_{ref,EOL} = C \times (1 - EOL) \times RFD \quad (6)$$

1.9.2.6 Summary

The environmental impact of refrigerants has been widely studied. The current challenge is data limitation. First, the ADP.GWP values were rarely mentioned. Second, for emissions produced during the refrigerant manufacturing process, the RFM values of newly reported refrigerants like R-466A were seldom reported. Third, many researchers noted the emissions during the refrigerant disposal or recycling process, but only GREEN-MAC-LCCP adds the refrigerant disposal emissions in the LCCP calculation.

1.9.3 Load Prediction

AHRI 210/240 provided an approach to estimate the load and power consumption by ASHRAE standard test system efficiency. This approach is called the temperature bin method. However, this approach is only applicable to fixed-speed systems. If the system had a variable speed compressor, the control logic of the compressor frequency would also affect the result. Thus, in this study, for a fixed speed system, we will use the temperature bin method. For the variable speed system, we developed a model in MATLAB to simulate the hourly power consumption in one typical year.

1.9.3.1 Temperature bin method

As for the process of calculating the thermal load using the temperature bin method, Troch et al. [11] and Lee et al. [12] gave a clear description in their papers. Interesting readers can also use AHRI 210/240 as a reference. The temperature bin definition is different in different regions. China (GB 21455-2019), Japan (JIS C 9612:2013), and the EU (Noo 206/2012) use an interval of 1 °C for each temperature bin. However, the USA (AHRI 210/240-2017) uses an interval of around 2.7 °C (5 °F). Figure 1-96 shows a comparison between the USA, China, Japan, and the EU. Several typical cities were chosen as an example. The weather data was from the National Centers for Environmental Information (NOAA) Integrated Surface Database (ISD) [42,43]. We can see that the US had a wider range than other countries. Many countries in Europe have a relatively lower temperature in the cooling season. Thus, the lower bound of the bin temperature (17 °C) is also lower than that in China and Japan (24 °C).

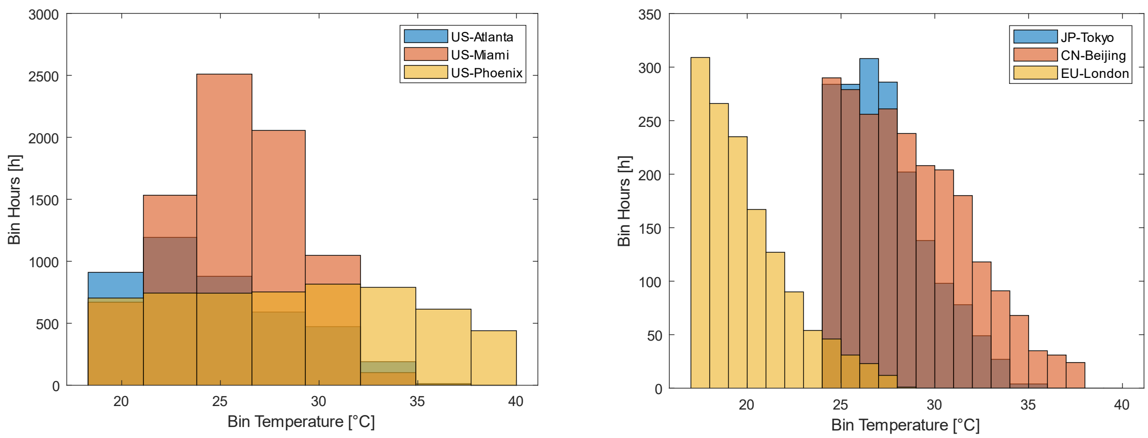

Figure 1-96 a) US temperature bins b) China, Japan, and EU temperature bins

1.9.3.2 Simulation-based method

As the weather data is available, we can simulate the load of the target building or a room. In this study, we consider a 10m×10m room facing south in the Northern Hemisphere. Two windows were installed facing south and north, respectively. The top and roof were assumed adiabatic. Other parameters could be found in Table 1-27. We also made the following assumptions: First, the optical depth parameters for the location were constants throughout the year; Second, the solar radiant time series were the same throughout the year; Third, windows had no shading. We used the method introduced in Wijeysundera's book to calculate the cooling and heating load of the target room [44].

Table 1-27 Parameters for simulation

Item	Value
Height	3 [m]
Window Wall Ratio (WWR)	0.6 [-]
Ground Reflectivity	0.25 [-]
Solar Absorptivity of Wall	0.8 [-]
Wall	Brick and a layer of insulation board
U of Wall	0.58 [Wm-2K-1]
Window	Double-glazed
People	75 W for sensible heat, 55 W for latent heat
Occupant Ratio	0.1 [m-2]
Equipment Ratio	13.5 [m-2]
Light Ratio	4.5 [Wm-2]
Work Time	9:00-19:00

Table 1-28 Load prediction

City	Latitude	Heating		Cooling	
		Load	Ambient temperature	Load	Ambient Temperature
CN_Beijing	39.9042	-5880.78	-3	9259.854	29
CN_Shanghai	31.2304	562.9471	3	8756.95	29
EU_SW_Kallax	65.517	-7903.94	-10.7	641.416	15.9
EU_SZ_Basel	47.5596	-5668.99	-0.66947	5491.785	21.04053
EU_UK_London	51.4074	-5307.59	0.773999	1745.069	18.651
JP_Tokyo	35.6762	-3873.53	5	8289.877	27
US_Atlanta	35.6762	-671.876	3.9	9394.784	28.3
US_Miami	35.6762	819.9277	0	11073.12	27.2
US_Phoenix	33.749	-2917.59	18.3	11183.06	30

1.9.3.3 Power Factor and Material-Embodied Carbon Coefficients

The emissions due to electricity generation are a dominant factor in the LCCP calculation. Different countries and different regions have different power plant emission factors due to the resource portion difference [3]. Carbon Footprint summarizes the country-specific electricity grid carbon emission factor in June 2019 [45]. The data for Asian countries is from G20 Green Report 2018 [16], for European countries is from the Association of Issuing Bodies [46], and for the USA is from the US Environment Protect Agency database [47]. The second column of Table 1-29 shows the power factors that we will use in this study.

Table 1-29 Power factor, Material usage, and Embodied Carbon Coefficients

<table>
<tr><th colspan="2"></th><th>Power Factor kg CO_{2e}/kWh</th><th>Aluminum kg CO_{2e}/kg</th><th>Copper kg CO_{2e}/kg</th><th>Plastic kg CO_{2e}/kg</th><th>Steel kg CO_{2e}/kg</th></tr>
<tr><td colspan="2">Percentage of usage</td><td>-</td><td>12%</td><td>19%</td><td>23%</td><td>46%</td></tr>
<tr><td colspan="2">Average around world</td><td>0.623</td><td>13.1</td><td>2.71</td><td>3.31</td><td>3.02</td></tr>
<tr><td rowspan="3">EU</td><td>UK</td><td>0.2773</td><td rowspan="3">6.58</td><td rowspan="3">2.71</td><td rowspan="3">-</td><td rowspan="3">1.8-2.89</td></tr>
<tr><td>SW</td><td>0.012</td></tr>
<tr><td>SZ</td><td>0.014</td></tr>
<tr><td colspan="2">CN</td><td>0.6236</td><td>14.6</td><td>-</td><td>-</td><td>3.5-4.5</td></tr>
<tr><td rowspan="3">US</td><td>Miami</td><td>0.4667</td><td rowspan="3">5.65</td><td rowspan="3">3</td><td rowspan="3">2.8</td><td rowspan="3">1.8</td></tr>
<tr><td>Phoenix</td><td>0.425</td></tr>
<tr><td>Atlanta</td><td>0.457</td></tr>
<tr><td colspan="2">JP</td><td>0.4916</td><td>10.6</td><td>-</td><td>-</td><td>1.64</td></tr>
<tr><td colspan="2">KR</td><td>0.517</td><td>11.9</td><td>-</td><td>-</td><td>-</td></tr>
</table>

The carbon emissions during the manufacturing phase of the system are another factor that may affect the LCCP calculation. Some previous studies used the same emissions values for the material in every country. For example, Choi et al. [3] used the values in the IIR guideline [48] to estimate the LCCP in Korea. However, IIR only provides the recommended values in the US. Some researchers, especially those working on LCA of buildings, have developed databases for different material's Embodied Carbon Coefficients (ECC) in different countries [49]. For this study, we used the Inventory of Carbon & Energy database developed by Hammond et al. [50]. For plastic and steel, many types existed. We used the general values for these two materials. Some ECCs are not found in the literature for some regions. The value around the work was used as a substitute in this study. As we can find in Table 1-29, the ECC for Aluminum in the US is around one-third of the value in China. Thus, ECCs could be a crucial factor in the LCCP calculation for different countries.

1.9.4 Weather Station data and On-site Weather data

Current studies, especially building simulation studies, have applied data collected from weather stations. The most used database includes the EnergyPlus built-in weather data, NOAA weather data, and TMY weather data. The first two datasets are the AMY dataset, while the last one is the TMY dataset. Some researchers have studied the difference between the applied AMY dataset and the TMY dataset. All the data was collected from weather stations, most of which were built around the airports. Some studies pointed out the temperature gaps between a city and an airport. Such temperature gap in an urban area or metropolitan area due to human activities is called an Urban Heat Island (UHI) [51]. The main cause for this effect is the modification of land surfaces [52] and waste heat generated [53]. Santamouris et al. [54] studied this effect from 220 projects and concluded that 31% of the peak temperature drop below 1 °C, 62% below 2 °C, 82 % below 3 °C and 90 % below 4 °C. Munck et al. [55] found that the increase in temperature was 0.5 °C in the situation with current heat releases, 1 °C with current releases converted to only sensible heat, and 2 °C for the future doubling of air conditioning waste heat released to air in Paris. This temperature gap could bring some differences in LCCP calculation. Thus, we would compare the LCCP results using weather station data and weather data corrected by Santamouris's statistics, respectively.

We also measured the ambient temperature ourselves. Thermocouples were installed next to an outdoor unit of an air conditioner in a campus building at UMD, College Park, US. The thermocouples were exposed in the air facing north and had no shadings. The ambient temperature tested was compared with the temperature data from Airport, College Park, US. The distance between the two places was 1.8 km. Figure 1-96 shows the comparisons between the two data. Figure 1-96 a) shows the daily temperature tested on Jan 15th, 2019. Figure 1-96 a) shows the daily temperature tested on Jul 15th, 2019. The blue line is the temperature tested in the campus reading through LabVIEW. The red line is the temperature tested in the airport from NOAA database. We can see that in winter, the temperature on campus was 10-20 °C higher than the temperature in the airport. Since the campus building sensors had no shading, solar radiation would have a significant effect on it. As a comparison, the temperature sensors used by the weather stations are usually stored in boxes, which have less effect from the radiation. The built-in sensors of the outdoor units are usually directly exposed to the air. Thus, the campus case should be closer to the field test case. This temperature gap could also be caused by human activity and other AC outdoor unit outlet waste heat. However, during the summer, the temperature on campus had a higher peak but lower valley than the temperature at the airport. The reason could be that the heat capacity of the air in the airport is larger than that on campus. Figure 1-98 shows a histogram of the two temperatures in 2019. 118-hour data points in the campus testing dataset and 43-hour data points in the airport dataset were not validated due to the power shortage or database breakdown. We deleted these data points when we drew Figure 1-97. Thus, 8,599 data points exist in this figure. We will use these two datasets separately to calculate the LCCP and discuss the differences in the next chapter. The findings here are that the gaps between the on-site air conditioner sensors and the weather station data might be much more larger than previous researchers assume.

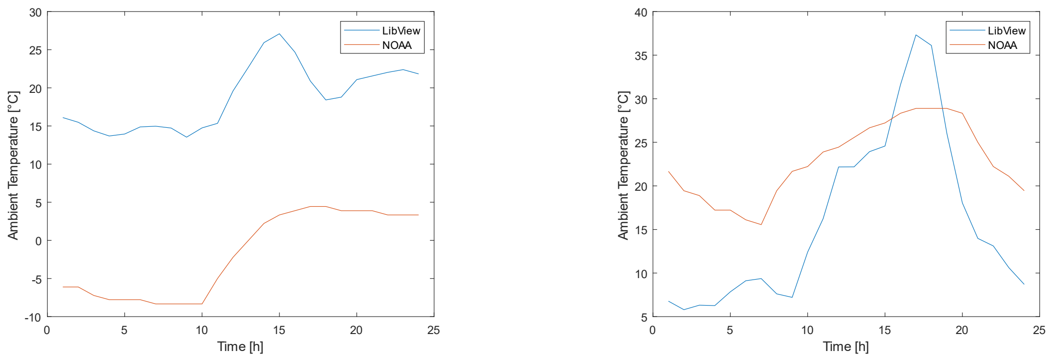

Figure 1-97 a) January 15th Ambient Temperature b) July 1st Ambient Temperature

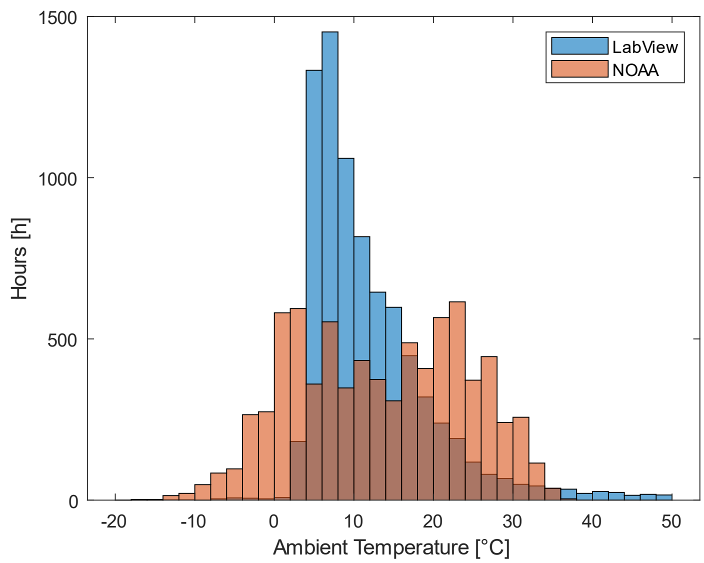

Figure 1-98 Histogram of year 2019 ambient air temperature

1.9.5 Results

1.9.5.1 Different regions

Figure 1-99 shows the LCCP results in different regions for R410A as an example. From the figure, we can see that the LCCP for Basel and Kallax is very small. The reason is that the power factors of Sweden and Switzerland are very small. Only for the two regions is the annual leakage the major factor of the LCCP. For all other regions, the annual energy consumption is the main factor for the LCCP.

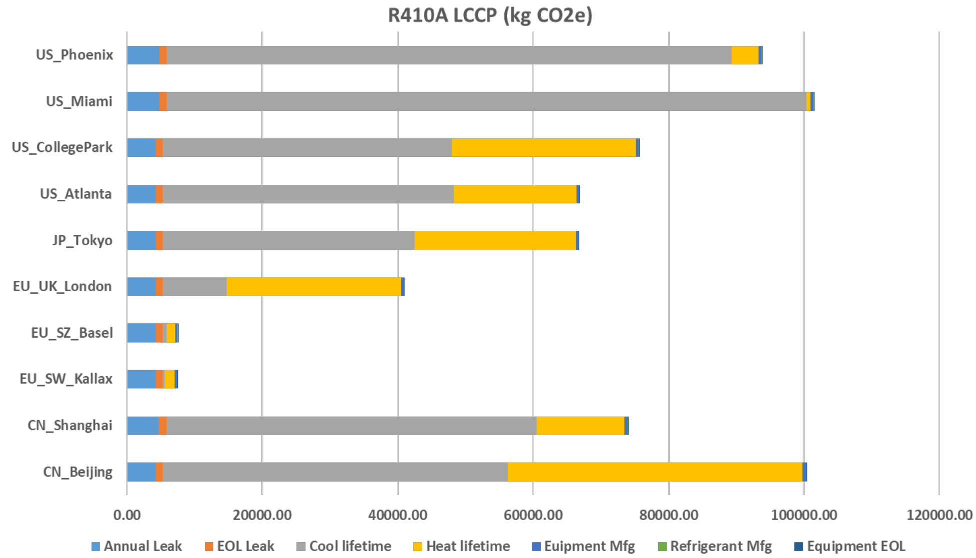

Figure 1-99 LCCP results for R410A

1.9.5.2 Different refrigerants

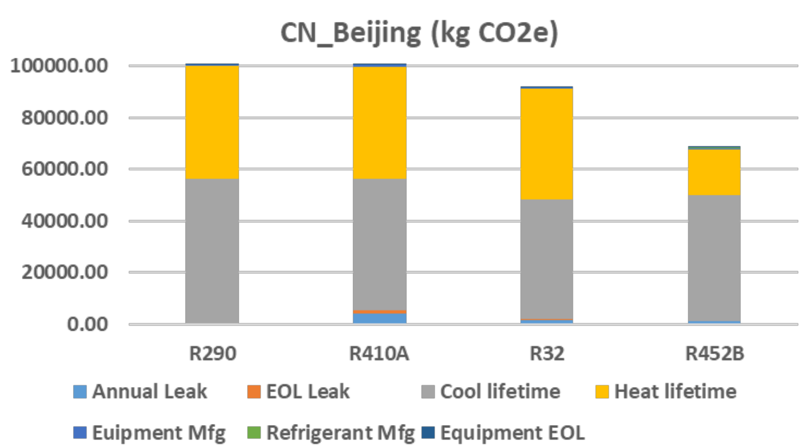

Figure 1-100 Beijing, China LCCP results

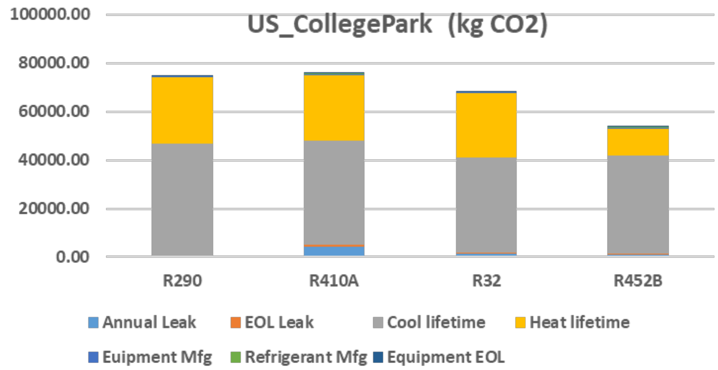

Figure 1-101 College Park, US LCCP results

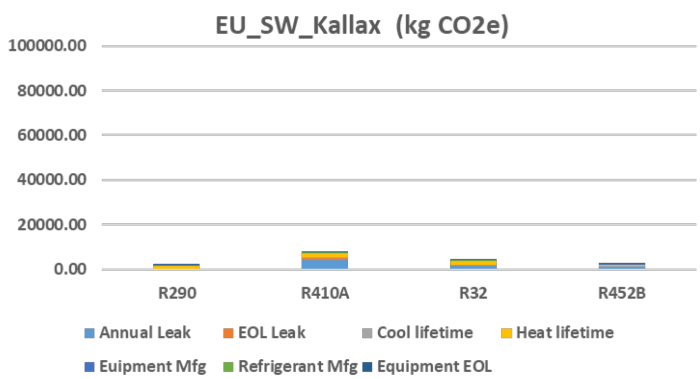

Figure 1-102 Kallax, Sweden LCCP results

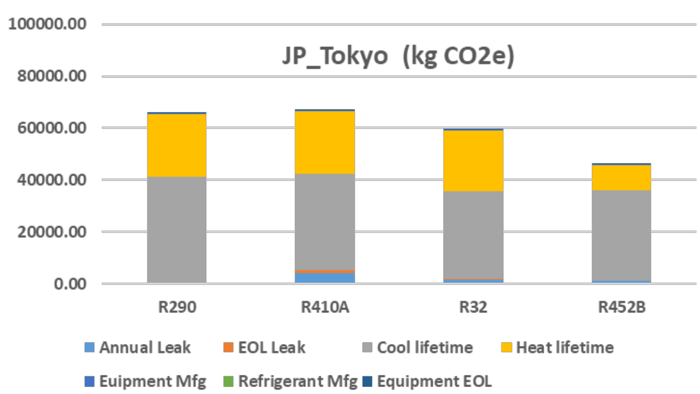

Figure 1-103 Tokyo, Japan LCCP results

Figures 1-100 to Figure 1-103 show the LCCP in four different cities for four different refrigerants. We can see that replacing R-410A with R-290 may not always be helpful in decreasing the LCCP. For example, for the regions with high power factors like Beijing, the LCCP for R-290 and R-410A are very close. R-452B is always a good substitution for R-410A with lower LCCP.

1.9.5.3 Ambient Temperature Study

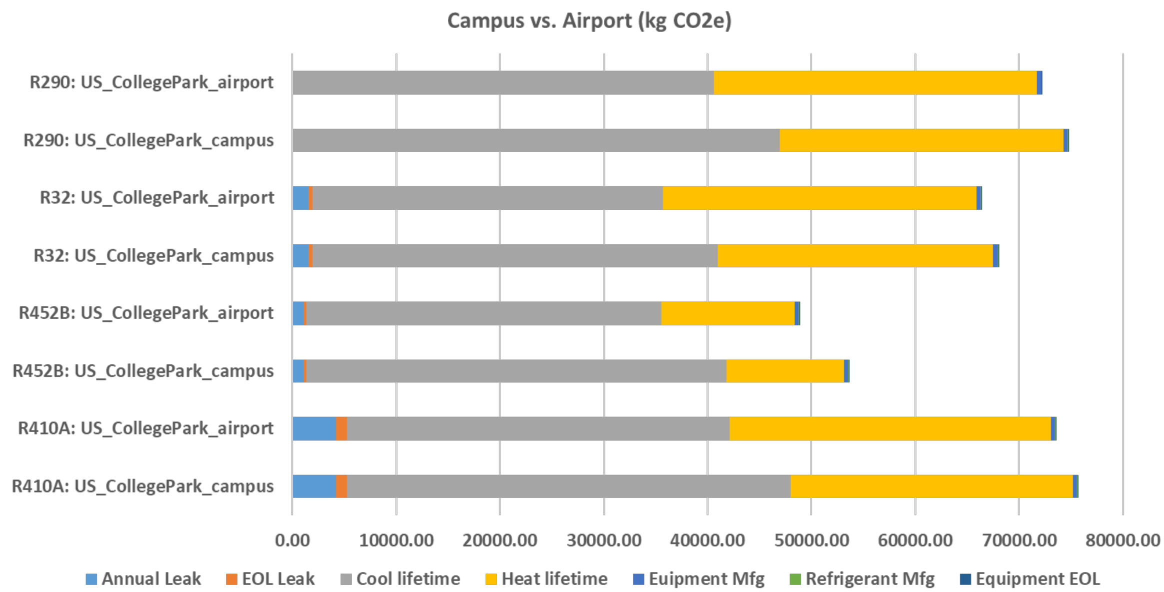

Figure 1-104 Campus vs. Airport

Figure 1-104 shows the comparisons between the LCCP calculation based on campus ambient temperature and airport ambient temperature. We can find that for all refrigerants, the LCCP calculation using the airport ambient temperature will cause the result to be 2% to 10% lower than using the campus ambient temperature.

1.9.6 Conclusions

In this study, we reviewed the weather data, the power factors, the manufacture embodied carbon emissions for refrigerants and materials, and the percentage of material used in the LCCP calculation. We can evaluate the LCCP for both fixed-speed systems and variable-speed systems. We compare the LCCP in different regions for different refrigerants. The conclusions are listed as follows: i) For most countries, the annual energy consumption is the major factor; ii) Except for emissions caused by heating and cooling, the reset parts are very close for each region; iii) Low GWP refrigerants may not be able to decrease the carbon emissions in all regions; iv) We compared the weather data collected from the campus and from the airport.

As for future work, we will add more new refrigerants like R-466A. We will also compare more regions around the world. We may also study the relationship between LCCP and the cost and optimize the LCCP based on the cost.

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1.10 Review of low GWP refrigerant for HP system

As highlighted by the speaker at the event of the HFC lifecycle management workshop [1], reducing hydrofluorocarbons (HFCs), super-polluting greenhouse gases used in refrigeration, air-conditioning, insulating foams, and aerosol propellants, plays an important role for decarbonization. HFCs are hundreds to thousands of times more powerful than carbon dioxide and contribute to global warming. The global implementation of the Kigali Amendment to the Montreal Protocol requires a phase-down of HFCs by more than 80% over the next 30 years, which can prevent more than 70 billion metric tons of carbon dioxide equivalent emissions by the middle of the century, leading to a reduction in global warming by 0.1°C by 2050 and up to 0.5°C by the end of the century. It is crucial to emphasize the significance of replacing high-global-warming-potential HFCs with next-generation refrigerants and enhancing energy efficiency in order to decrease emissions. The speaker highlighted the benefits of increasing the capture and reuse of HFCs already inside refrigeration and air conditioning equipment. The United States has ratified the Kigali Amendment and is leading the way in reducing HFCs domestically through the 2020 American Innovation and Manufacturing Act. Collaboration with industry stakeholders and working together towards achieving a net-zero economy is essential for a successful transition.

Theresa [2] presents the results of a study on refrigerant use for air conditioning (AC) equipment maintenance in the United States. The study collected data from a survey of over 1,000 AC contractors and technicians, as well as an analysis of refrigerant sales data from a major supplier. The survey found that the majority of technicians (68%) do not perform refrigerant recovery during AC maintenance, and instead simply release the refrigerant into the atmosphere. The analysis of sales data found that the most commonly sold refrigerants were R-22 and R-410A, which are both high global warming potential (GWP) refrigerants that contribute to climate change. The study also estimated that the amount of refrigerant released into the atmosphere during AC maintenance is equivalent to the emissions from over 9 million cars each year. The authors argue that better training and education for technicians, as well as stronger regulations and enforcement, are needed to reduce refrigerant emissions from AC maintenance and address the climate impact of these emissions.

As discussed before [3], the harmonization of life cycle climate performance (LCCP) methodology is used to assess the greenhouse gas (GHG) emissions of products or systems throughout their life cycle. The authors highlight the importance of standardizing the methodology to ensure consistent and accurate GHG emissions calculations. They review different LCCP approaches and methods, such as the PAS 2050, the ISO 14044, and the Product Category Rules (PCR), and their strengths and limitations. The article emphasizes the need for a standardized, transparent, and flexible LCCP methodology that can be adapted to different sectors and product types. The authors also discuss the challenges of data collection and quality, the role of technology and innovation, and the importance of stakeholder engagement in the development and implementation of LCCP methodologies. The article concludes that harmonization of LCCP methodology can contribute to more informed decision-making and promote sustainable practices across different industries.

1.10.1 Rrefrigerants selection & evolution

Mark [4] discussed the evolution of refrigerant molecules, the constraints and regulations that have driven the need to consider new molecules, and the advancements in the tools and property models used to identify new molecules and design equipment using them. The authors highlight three comprehensive searches for new refrigerants that took place in the 1920s, 1980s, and 2010s, which sometimes identified new molecules, but more often validated alternatives already under consideration. The paper also notes that there is little that is truly new in the search for new refrigerants and that most of the "new" refrigerants were reported in the chemical literature decades before they were considered as refrigerants. The authors also review the evolution of the NIST REFPROP database for the calculation of refrigerant properties. Bell [5] provides a survey of existing data for refrigerant blends containing halogenated olefins and presents the data in graphical form. The data is primarily taken from the NIST SOURCE database. The conclusion is that there is a lack of experimental data for blends containing halogenated olefins and some classes of data are particularly sparse. The second part of the study compares the thermodynamic models in NIST REFPROP against the experimental data sets and identifies systems for which refitting of the thermodynamic model is required.

1.10.2 Alternative refrigerants

R290 (propane) and R1234yf (a low GWP refrigerant) are considered promising options as alternatives to traditional refrigerants due to their good thermodynamic properties, low environmental impact, and energy efficiency. These refrigerants have the potential to significantly reduce greenhouse gas emissions from the refrigeration and air conditioning industry. However, their flammability and toxicity properties should also be considered and appropriate safety measures must be implemented during installation and maintenance. Additionally, further research is needed to fully understand the potential impact of these refrigerants on the environment and human health. Zhang compares the performance of R290 and R1234yf, in heat pump water heaters with low charge [6]. The experimental results show that both refrigerants have comparable heating capacity and COP at low charge levels, with R290 showing slightly better performance. However, R1234yf has a lower flammability risk and a lower impact on the ozone layer, making it a safer and more environmentally friendly option. The study concludes that R1234yf can be a good alternative to R290 in heat pump water heaters with low charge, especially in regions with stricter regulations on flammable refrigerants.

The Fraunhofer Institute for Solar Energy Systems is developing a low refrigerant charge refrigeration circuit with a propane heat pump as part of the "LC150" project funded by the German Federal Ministry of Economics and Climate Protection [7]. The research team has been building prototype brine heat pumps with reduced amounts of refrigerant by adjusting factors such as the internal volume of the heat exchangers, the amount of oil required, and reducing the number of components such as sensors. The final version of the refrigeration circuit is expected to achieve a maximum output of 8 to 10 kilowatts and a maximum charge of 150 grams of refrigerant, with a more balanced system using a fully hermetic compressor and slightly more refrigerant.

Alfares [8] provides a review of the experimental application of R-410A alternative refrigerants, with a focus on low-GWP refrigerants. The authors collect a wide range of experimental data from the literature and analyze the performance and environmental impact of these refrigerants. They find that DR-55 (R-452B) provides an overall performance closer to R-410A in existing refrigeration systems. Additionally, the authors conclude that DR-55 (R-452B) and DR-5A (R-454B) are the most suitable candidates for residential air conditioners and heat pumps. Overall, the paper provides valuable insights into the use of low-GWP refrigerants as alternatives to R-410A in refrigeration systems.

Table 1-30. Alternative refrigerants normalized ranking parameters [8]

Refrigerant	Ranking parameters				Total
	Heating ratio	Cooling ratio	COP ratio	Discharge temperature ratio	
DR-5A	-0.04	-0.04	0.01	0.12	0.04
DR-55	-0.03	0.02	0.02	-0.05	-0.04
R-32	0.02	0.05	0.02	-0.19	-0.11
L41A	-0.09	-0.07	0.00	-0.06	-0.22
L41-2	-0.13	-0.11	-0.01	-0.09	-0.34
D2Y-60	-0.30	-0.19	-0.03	0.07	-0.45
L41-1	-0.50	-0.46	-0.05	-0.06	-1.08

Yasuharu [9] performed an in-silico screening of 10,163 molecules using quantum chemical calculations, machine learning, and database search to find suitable candidates. The author discussed the urgent need for the development of new refrigerants with low GWP while meeting conventional requirements such as cooling performance, safety, and non-destructiveness to the ozone layer. It highlights the tradeoffs among various chemical properties and the need to properly control these properties to fulfill the requirements.

Rydkin [10] reviews a novel refrigerant, GWP 1 HFO refrigerant 1132E, and its binary mixture with R-1234yf. The author covered the critical parameters, flammability classification, stability studies, initial materials, and oil compatibility screening, and laboratory-validated refrigerant cycle performance properties of the binary blend. The binary blend is intended for use as a replacement for R-134a and R-1234yf in heat pump applications, as part of the phasedown of HFCs under the Kigali amendment and the US AIM Act.

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1.11 Component Research Review and Heat Exchanger Optimization for Residential Air-conditioning Systems with Low-GWP Refrigerants

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1.11.1 Component Research Review

1.11.1.1 Introduction

Throughout the last few decades, there has been a significant effort to improve the environmental impact of refrigerants used in residential air conditioning systems. It started with the phasing out of Ozone Depletion Substances, the most common one being R-22. With the elimination of R-22, then came R-410A. It showed promising signs from the beginning, but with the advancements in technology and increasing awareness of global warming, it quickly began to show its faults. R410A has a GWP 2088. Consequently, when it leaks out of its air conditioning system, it has a tremendous effect on the buildup of greenhouse gases. Fortunately, researchers and engineers across the world are actively working to find a low-GWP solution to R-410A.

In recent years, there have been numerous literature reviews on possible low-GWP refrigerants. Furthermore, much of these literature reviews have been primarily focused on the adoption of low-GWP refrigerants in residential air conditioning systems. The following are a few of the recent reviews carried out by leaders in both industry and the academic communities:

The following are studies with a primary focus on testing and analyzing potential alternative refrigerants. Heredia-Aricapa et al. (2020) [1] reviewed the most up-to-date work on HFC, HFO, HC, and R744 as replacement refrigerants for R134a, R404, and R410A. Their work revealed that R32 appears to be one of the leading replacement candidates but has some drawbacks that some mixtures can overcome, like high discharge temperature and final energy performance. Harby (2017) [2] presented their findings from analyzing studies regarding hydrocarbons as refrigerants in refrigeration, air conditioning, heat pumps, and automobile air conditioning. They determined that hydrocarbons are not only an eco-friendly option but also provide the ability to reduce energy consumption and possess the necessary properties to be an excellent drop-in replacement for halogenated refrigerants. Additionally, they recommended that future research should be focused on HFC/HF mixtures, low charge large capacity, and equipment reliability. Mota-Babiloni et al. (2017) [3] summarized recent investigations that resulted from F-regulations on refrigerants and had a primary focus on lower GWP synthetic alternative refrigerants. When considering residential air conditioning systems, they noted that there are no significant advantages to using mixtures over R32 as a replacement for R-410A. Pabon et al (2020) [4] reviewed the most relevant work of the last decade on the potential of R-1234yf. In the end, they found most studies endorsed the use of R-1234yf as a replacement for R134a. However, they added that optimization on the circuit would be necessary in order to achieve optimal operating conditions. Kasaeian et al. (2018) [5] discussed recent theoretical and experimental studies surrounding the application of low-GWP, such as hydrocarbons, hydrofluorocarbons, R744 (carbon dioxide), hydrofluoroolefin, and nano refrigerants. The authors found that nano refrigerants offer many advantages, such as their thermodynamic performance and effects on lubricants. Despite they highlighted the fact that there needs to be much more research concerning their potential for being a replacement. Bolaji and Huan (2013) [6]) examined research on the possibilities of utilizing natural refrigerants in the areas of refrigeration and air conditioning. Through their research, the authors found that natural refrigerants are the most favorable for long-term replacement in both refrigeration and air-conditioning systems.

Here is one of the very few reviews on heat exchanger optimization. Tancabel et al (2018) [7] conducted a literature review on shape and topology optimization for the design of Air-to-Refrigerant Heat Exchangers. Through their research, they concluded that there is a need for more studies on Heat exchanger optimization using low-GWP refrigerants.

Table 1-31: Summary of recent reviews on refrigerant research

Author	Focus	Highlights
Bolaji and Huan (2013) [6]	Natural refrigerants as environmentally friendly alternative refrigerants	Natural refrigerants are the prospects for long-time replacements
Mota-Babiloni, Makhnatch, and Khodabandeh (2017) [3]	Low global warming potential synthetic alternative refrigerants.	No major advantages of using mixtures over R32.
Harby (2017) [2]	Hydrocarbons as a replacement in refrigeration, AC and automobile AC	Comprehensive review of latest hydrocarbon research efforts
Tancabel, Aute, and Radermacher (2018) [7]	Air-to-refrigerant HX shape and topology optimization	Lack of heat exchanger optimization studies for Low-GWP refrigerants
Abas et al. (2018) [8]	Natural and synthetic refrigerants.	Utilizing Refrigerant Qualitative Parametric (RQP) quantification model in the refrigerant choice decision
Kasaeian et al. (2018) [5]	Evaluate the potential of hydrocarbons, hydrofluorocarbons, R744 (carbon dioxide), hydrofluoro olefin, and nano refrigerants as environmentally friendly refrigerants.	Nano refrigerants have many advantages like their thermodynamic performance and lubricant interaction
Pabon et al. (2020) [4]	R1234yf as working fluid in compression systems	R1234yf endorsed as possible replacement
Heredia-Aricapa et al. (2020) [1]	Replacing R134a, R404A and R410A with HFC/HFO/HC/R744 refrigerant mixtures.	Mixtures have ability to overcome the drawbacks that R32 possess

Overall, much of the effort on summarizing recent work regarding low-GWP refrigerants has been concerned with the ability for them to be replacements to current refrigerants. As a result, there are many gaps in information provided by the reviews. First many of the existing reviews are not specifically dedicated to the application of low-GWP refrigerants in residential applications. Secondly, much of the studies examined only discuss alternative refrigerants as drop in or performance in terms of the entire system, only a very limited amount discussed specifics down to a component level. Lastly, there are hardly any reviews devoted solely to component optimization with new refrigerants. For this reason, the objective of this report is to present a thorough review of all the recent progress and developments in the areas concerning component optimization of low-GWP refrigerants in residential air conditioning systems. This report is intended to serve as the most current update on the R-410A phase-down for engineers and researchers. The review is structured into the following three sections: heat exchangers, compressors, and expansion valves/ other components. For the heat exchanger portion the two main categories are tube-fin and microchannel. In those two-sub section we further categorized the sources as either a simulation or experiment study. For the compressor the section was only divided between simulation and experimental studies. As a result of the limited number of expansion valve/other component studies these have just been listed under the category for other.

1.11.1.2 Review of Component Optimization Progress

1.11.1.3 Heat Exchangers

1.11.1.3.1 Tube-fin Heat Exchangers

Tube-fin heat exchangers are one of the most common types of heat exchangers investigated in residential air conditioning systems, mostly because of their simplistic design and long commercial usage history. However, tube-fin heat exchangers do have their disadvantages. To begin with, because of the design of tube-fin heat exchangers, they require a lot of material. Additionally, compared to other heat exchanger designs, tube-fin heat exchangers require a higher refrigerant charge. Researchers and engineers have been working on ways to overcome these drawbacks by exploring alternative refrigerants to R-410A and the benefits they provide. Overall, R32, R290, and HFO blends are thought to be the best replacement candidates when using tube-fin heat exchangers. Lastly, researchers and engineers have been exploring the possibility of implementing small-diameter copper tubes into the heat exchanger designs and using simulation software like ISHED (Intelligent System for Heat Exchanger Design) to optimize the design of the tube-fin Heat exchangers for specific refrigerants. Table 1-32 provides a summary of the review of recent tube-fin heat exchanger studies.

1.11.1.3.2 Simulation

Domanski and Yashar (2007) [9] described the outcomes of applying the ISHED (intelligent system heat exchanger design) simulator to optimize tube-fin condensers. The authors explained that ISHED has the capability to produce optimized circuitry architecture. Their results showed that R-32 performed the best, while R-600a performed the worst. Additionally, the ratio of condenser capacity between the two was calculated to be 1.18.

Shabtay et al. (2014) [14] presents simulation studies on how small-diameter copper tube heat exchangers can be applied to low-GWP alternative refrigerants. The authors concluded that all four alternatives, R-290, R-744, HFO blends, and R-32, all require less charge and provide lower costs. It was found that using R-32 reduced tube sizes by 30% compared to R-410A.

Kamada et al. (2016) [15] conducted simulations to optimize heat exchangers in heat pumps for R-32 and HFO mixtures. In particular, they focused on how to overcome the temperature glide to achieve a better performance by optimizing the number of passes. They concluded that HFO blends, like R-32 /R-125/R-1234yf, that have smaller temperature glides, will have a reduced ratio of improvement of heat transfer by optimizing the number of passes.

Besher et al. (2016) [16] examined three different optimization simulations on heat exchangers. The author focused on using genetic algorithms to optimize the small diameter tubes in the heat exchangers in order to optimize the designs of heat exchangers for the use of both R410A and R-32. They concluded that It is possible to reduce the heat exchangers' cost by 60% without having to reduce the heat exchangers' performance. Additionally, the authors noted that the system's refrigerant charge can be decreased by 35% by using the ASHP system.

Cho and Domanski (2016) [17] examined the performance of twelve R22 and R410A low-GWP alternatives after optimizing the tube-fin heat exchangers using first-principle-based simulation models and ISHED, Intelligent System for Heat Exchanger design. They reported that all the alternative refrigerants benefited from the optimization of tube-fin heat exchangers. At the same time, R-744 and R-717 saw the highest performance improvements, with their capacities increasing by at least 13% and 10%, respectively.

Li et al. (2019) [19] presented a novel integer permutation-based Genetic Algorithm (IPGA) for optimizing Tube-fin heat exchanger circuitry. Through their simulations, they found that a 2.4–14.6% increase in heat exchange capacity resulted from applying IPGA to an A-shaped indoor unit. Additionally, they noted in comparison to other optimization methods, IPGA results in higher capacity, lower pressure drop, and better manufacturability.

Kim et al. (2020) [18] conducted drop in simulations to determine the optimal tube-fin heat exchanger, primarily focusing on the tube diameter and the number of paths. Two alternative refrigerants that were examined, R32 and L-41a. The authors observed R-32 and L-41a to have Energy Efficiency Ratios over R410A of 104.9% and 98.5% respectively. Furthermore, their Coefficients of Performances were 102.7% and 100.8% compared to R410A.

1.11.1.3.3 Experiment

Tao et al. (2010) [10] provided experimental data on how working conditions affect the COP of a transcritical R744 residential air conditioning system using both a tube-fin heat exchanger and an internal heat exchanger. Through their experiments, they found that the COP increases by about 27% when the air inlet velocity is increased from .68 m/s to 1.8 m/s. As a result, it was concluded that increasing the heat transfer coefficient is a proficient way of enhancing the system's performance.

In et al. (2014) [11] performed experimental studies on tube heat exchangers using two alternative refrigerants to R-410A, R-32 and L-41b (HFO blend). When analyzing the HFO blend it was discovered that its performance was much worse than that of R-410A. The authors justified the reason for this as being a result of the HFO blend having a higher viscosity, which causes a larger pressure drop and a lower mass flow rate. It was suggested that partial optimization of the tube size and the number of passes in the heat exchanger would increase the performance of the HFO blend.

Cheng et al. (2014) [12] explored R-32 and R-290 as possible alternatives to R-410A in small tube heat exchangers. They noted that R-290 had the smallest refrigerant charge, half of the charge required for R-22, R-32, and R-410A. Additionally, the authors described how R-290 could be the best alternative refrigerant for small tube heat exchangers in air conditioners.

Table 1-32: Review of Recent Tube-Fin Studies

Author	Type	Focus	Highlights
Domanski and Yashar (2007) [9]	Simulation	Optimize tube fin heat exchangers with ISHED	R600a, R134a, R290, R22, R410A, R32 examined. R32 performed the best. R600a performed the worst.
Tao et al. (2010) [10]	Experiment	Examined how working conditions affect the COP of a trans critical R744 residential air conditioning system	Increasing the heat transfer coefficient is a proficient way of enhancing the system performance.
In et al. (2014) [11]	Experiment	Experimental optimization of tube-fin heat exchangers using R32 and L-41b (HFO Blend).	HFO blend's performance much worse than R410A.
Cheng, Wang, and Liu (2014) [12]	Experiment	Explore possible alternative refrigerants to R410A in small tube fin heat exchangers.	R290 had the smallest refrigerant charge.
Ren et al. (2014) [13]	Experiment	Effect of suction line heat exchanger on R290 Air conditioner with small diameter copper tubes.	Improved the cooling capacity by 5.3%. Reduced the refrigerant charge by 6%.
Shabtay, Black, and Kraft (2014) [14]	Simulation	The use of small diameter copper tube heat exchangers with low-GWP refrigerants.	R32 reduce the tube diameter size by 30% compared to R410A R290, R744, HFO blends and R32 all require less charge and provide lower costs.
Kamada, Haikawa, and Taira (2016) [15]	Simulation	Optimize heat exchangers in heat pumps for R32 and HFO-mixtures	HFO blends with smaller temperature glide will have a reduced ratio of improvement of heat transfer by the optimizing the number of passes
Beshr, Aute, and Radermacher (2016) [16]	Simulation	Optimize tube fin heat exchangers using small diameter tubes for R410A and R32, from performance, charge and cost aspects.	60% heat exchanger cost reduction potential without degrading the performance. 35% charge reductions using ASHP systems.
Cho and Domanski (2016) [17]	Simulation	Twelve R22 and R410A alternatives examined in tube fin heat exchangers optimized with ISHED.	R744 and R717 have the highest performance improvements with their capacities increasing by at least 13% and 10% respectively.
Kim et al. (2020) [18]	Simulation	Determine the optimal number of paths and tube diameter in tube fin heat exchangers using R32 and L-41a.	R-32 and L-41a having Energy Efficiency Ratios (EER) over R410A of 104.9% and 98.5% respectively. COPs were 102.7% and 100.8% compared to R410A.

Ren et al. (2014) [13] investigated experimentally the effect of a suction line heat exchanger on an R-290 Air Conditioner with a small diameter copper tube. Their experiment data showed that the suction line heat exchanger improved the cooling capacity by 5.3% and reduced the refrigerant charge by 6%.

1.11.1.4 Microchannel Heat Exchangers

Over the past two decades, researchers and engineers have started to explore the possibility of utilizing microchannel heat exchangers with R410A alternative refrigerants. They have concluded that R-32, R-744, and HFO blends contain the best characteristics to achieve the best results when using microchannel heat exchangers. Additionally, their work has shown that microchannel heat exchangers provide two major advantages. The first advantage is that microchannel heat exchangers reduce the required refrigerant charge, and secondly, they have the ability to increase the cooling capacity. Table 1-33 provides the review summary of recent microchannel heat exchanger studies.

1.11.1.4.1 Simulation

Jain and Bullard (2004) [20] reviewed simulations to obtain optimal heat exchanger geometries using R290 with the purpose of reducing refrigerant charge and achieving an equivalent system efficiency. Their simulations resulted in the following conclusions: microchannel heat exchangers can reduce the system charge by about a factor of 5 compared to the tube-fin heat exchangers, tube-fin heat exchangers hold about 70% of the system charge, whereas microchannel heat exchangers only make up for about 20%.

Yun et al. (2007) [21] investigated microchannel heat exchanger designs for R-744 air-conditioning systems. It was determined that the microchannel heat exchanger performance can be enhanced by increasing the two-phase region, modifying the fin spacing, and varying the flow rate of the refrigerant to the slab.

Shen et al. (2016) [25] presented that microchannel heat exchangers are capable of reducing the refrigerant charge up to 55% for some of the refrigerants, with R-290 requiring the least amount of charge. They were also able to conclude it would be possible to use R-290 in a small capacity room air-conditioner with only microchannel heat exchangers.

López-Belchí and Illán-Gómez (2017) [27] investigated the use of R-32 as a replacement for R410A using a microchannel condenser. They calculated through numerical analysis that if R-32 were directly used a drop-in replacement could either have a positive or negative effect on the refrigerant charge and energy efficiency. However, it was concluded that if the system was modified to run with R-32, then the refrigerant charge would decrease, and the energy efficiency would increase.

Zanetti et al. (2018) [28] discussed the importance of using a microchannel heat exchanger with low-GWP refrigerant like R-32 to reduce refrigerant charge. In addition, they showed through a numerical simulation that using a microchannel heat exchanger as a condenser reduces the refrigerant charge by about 30% compared to a tube-fin heat exchanger.

Table 1-33: Review of Recent Microchannel Studies

Author	Type	Focus	Highlights
Jain and Bullard (2004) [20]	Simulation	Optimize heat exchanger geometry for R290 to reduce refrigerant charge.	Microchannel heat exchangers can reduce the system charge by about a factor of 5. Tube fin heat exchangers hold about 70% of the system charge whereas microchannel heat exchangers only make up for about 20%.
Yun, Kim, and Park (2007) [21]	Simulation	Investigate possible microchannel heat exchanger designs for R744.	There are several ways to improve microchannel heat exchanger performance.
Pham and Rajendran (2012) [22]	Experiment	Compare R32 and HFO blends to R410A through drop in tests.	When using a microchannel heat exchanger design the refrigerant charge is reduced by 30~40% compared to using a tube-fin heat exchanger.
Fuentes and Hrnjak (2012) [23]	Experiment	Explore potential charge reduction for several refrigerants using a microchannel evaporator.	R744 showed the highest potential for possible refrigerant reduction.
Tian et al. (2015) [24]	Experiment	Compare tube-fin heat exchanger to a microchannel heat exchanger using R32/R290 as the refrigerant.	MHX reduced power consumption by .4%. MHX increased cooling capacity by 6.4%.
Shen, Bhandari, and Rane (2016) [25]	Simulation	Extend the application of MHXs as evaporators in split, room air conditioners (RAC) using low-GWP refrigerants.	MHXs are capable of reducing the refrigerant charge up to 55% for some of the refrigerants.
Xu et al. (2016) [26]	Experiment	Examine a novel low charge microchannel condenser	System refrigerant charge decreased by 28.3%. Cooling capacity increased by 1.6%.
López-Belchí and Illán-Gómez (2017) [27]	Simulation	Investigated the use of R32 as a replacement for R410A using a microchannel condenser.	With a modified system R32 will reduce the refrigerant charge and increase the energy efficiency.
Zanetti et al. (2018) [28]	Simulation	Microchannel heat exchanger working as evaporator and condenser.	Using a microchannel heat exchanger as a condenser reduces the refrigerant charge by about 30% compared to a tube-fin heat exchanger

1.11.1.4.2 Experiment

Pham and Rajendran (2012) [22] presented data comparing R-32 and HFO blends to R-410A through various drop-in tests using different heat exchangers. They reported that in their experiment, it was found when using a microchannel heat exchanger design, the refrigerant charge is reduced by 30~40% compared to using a tube-fin heat exchanger.

Fuentes and Hrnjak (2012) [23] displayed experimental results of potential charge reduction for several refrigerants using a microchannel evaporator. Their experiment was designed to compare the refrigerant charge vs the hydraulic diameter of the evaporator for a 1 kW refrigeration system, causing a 2% difference from the ideal COP due to the evaporator pressure drop. They concluded that R-744 showed the highest potential for possible refrigerant reduction.

Tian et al. (2015) [24] explored swapping a tube-fin heat exchanger with a microchannel heat exchanger and compared them through experiment testing using R-32/R-290. They found by using the microchannel heat exchanger the power consumption was reduced by .4%. Additionally, the microchannel heat exchanger increased the cooling capacity by 6.4%.

In another review Xu et al. (2016) [26] discussed experiments comparing traditional microchannel heat exchangers with novel low-charge microchannel heat exchangers. The experimental data showed that the system refrigerant charge decreased by 28.3% and that the cooling capacity increased by 1.6% when using a novel heat exchanger.

1.11.1.5 Other Designs

1.11.1.5.1 Simulation

Bacellar et al. (2017) [29] conducted studies on finless, novel-shaped heat exchangers and if they can outperform a microchannel heat exchanger. Through experiments with a 3d printed prototype, they found that the finless design can achieve more than 50% reduction in size, material, and pressure drop compared to the baseline microchannel heat exchanger.

1.11.1.6 Compressors

In respect to the number of studies done on heat exchangers, there are much less about compressors. Still, there has been an increasing amount of work done on compressors in recent years. The work is really split into two major categories. One is modifications and drop-in tests using new and old compressor hardware. The second is researchers taking a look at possible new lubricants to improve the performance of low-GWP refrigerants in compressors and systems as a whole. In this section, our findings are split into two sub-sections: simulation studies and experimental studies.

1.11.1.6.1 Simulation

Table 1-34 provides the review summary of recent simulation studies on compressors. Sethi and Motta (2016) [30] discussed simulations that show the benefits of using low-GWP alternatives over R-410A in stationary air conditioning systems. They found that both the alternative refrigerants researched, R-447B and R-452B, had discharge temperatures within 10°C and 5°C of R410A respectively. As a result, discharge temperature mitigation is not required for both R□447B and R-452B.

Pham and Monnier (2016) [31] examined low-GWP replacement refrigerants for R-410A. From their simulations they concluded that the replacements all have high compression heat leading to a high discharge temperature. They listed possible solutions to reduce the discharge temperature as 1) optimizing compressor internal design, features, and materials of construction 2) employing compressor vapor injection (VI) or liquid injection (LI) cycle, or 3) improving the oil to enable higher maximum allowable discharge temperature

Table 1-34: Review of Recent Simulation studies on Compressors

Author	Type	Focus	Highlights
Sethi and Motta (2016) [30]	Scroll, Centrifugal	Benefits of Low-GWP Alternatives over R410A in stationary air conditioning systems.	Compressor discharge temperature mitigation not required for R447B and R452B.
Pham and Monnier (2016) [31]	Scroll	Explored possible low-GWP refrigerant solutions to R410A.	Recommended reducing discharge temperature by optimizing compressor internal design, employing an injection system, improve oil properties.
Mota-Babiloni et al. (2017) [32]	Scroll, Rotary	R32 as a possible alternative for R410A.	R32 resulted in too high of a discharge temperature as a direct replacement.
Berkah Fajar et al. (2020) [33]	Reciprocating	Analyzed the energy and exergy of a small vapor compression system with R410A and R290.	R290 cut the compressor power consumption by 35.7% compared to R410A.

Mota-Babiloni et al. (2017) [32] examined the possibility of using R32 as a possible alternative to R-410A. In their research, they found that the compressor discharge temperature for a direct replacement using R32 was too high. It was suggested that modifying the system to utilize a liquid, vapor or two-phase refrigerant injection system in order to decrease the compressor discharge temperature. Additionally, they noted that using an R-32 mixture could decrease the compressor discharge temperature.

Berkah Fajar et al. [33] analyzed the energy and exergy of a small vapor compression system with R-410A and R-290. Their simulations showed that the power consumption of the compressor with R-290 compared to R410A decreased by 35.7%.

1.11.1.6.2 Experiment

Table 1-35 provides the review summary of recent experimental studies on compressors. Pham and Rajendran (2012) [22] focused on potential R-410A low-GWP replacements for air conditioning and heat pump systems. One of their experiments compared the compressor discharge temperature of R410A with an optimized compressor and R-32 with a non-optimized compressor. The results showed that R32's discharge temperature was about 22% higher than that of R-410A. They suggested the following optimizations for the R-32 compressor to improve the discharge temperature: optimize the compressor's internal design, use compressor vapor injection, and improve the oil.

Barve and Cremaschi (2012) [34] conducted an experimental comparison of the energy performance of refrigerants R-32 and R-1234yf in an R-410A heat pump split system for residential applications. Their results for the efficiency of the compressor showed the R-32 was generally higher than R-410A and R-1234yf was lower than R-410A.

Karnaz (2014) [35] investigated how low-GWP refrigerant performs with current POE lubricants. Their experiment showed that some refrigerants like R-32 and R-1234ze will require lubricant optimization on the current POE lubricants.

Table 1-35: Review of Recent Experimental Studies on Compressor

Author	Type	Focus	Highlights
Pham and Rajendran (2012) [22]	Scroll	R410A low-GWP replacements for air conditioning and heat pump systems	Suggested optimizations to improve discharge temp. Optimize compressor internal design. Use compressor vapor injection. Improve the oil.
Barve and Cremaschi (2012) [34]	Hermetic reciprocating	Experimental comparison in energy performance of refrigerants R32 and R1234yf to R410A.	R32 had higher compressors efficiency than R410A.
Karnaz (2014) [35]	Scroll, Reciprocating	Low-GWP refrigerant performance with current POE lubricants.	R-32 and R-1234ze require lubricant optimization on the current POE lubricants.
Hessell, Urrego, and Benanti (2014) [36]	Not Specified	Challenges of finding lubricants for low-GWP refrigerants like R32 and blends.	Advanced POE'S (APOE) best option for R32 and blends.
In et al. (2014) [37]	Swing	Partial optimization of a compressor using low-GWP refrigerants.	R32 system's compressor discharge temperature was 20-25% larger than the R410A system.
Schultz (2014) [38]	Scroll	Low-GWP refrigerant tests in heap pump designed for R410A.	Cycle and hardware modifications need to reduce R32's discharge temperature.
Yuuichi Yamamoto et al. (2015) [39]	Swing, Rotary	Compared an optimized swing compressor with R32 to a compressor with R410A.	COP for R32 was 101.2% of that of R410A.

Hessell et al. (2014) [36] discussed the challenges of finding lubricants for low-GWP refrigerants like R-32 and blends. Through experimental tests, they found that advanced POEs (APOE) are the best option. For this reason APOEs have the ability to be made to have acceptable miscibility in R-32 while maintaining the same performance as prior lubricants with R-410A.

Schultz (2014) [38] conducted numerous tests on a heap pump design for R-410A using low-GWP refrigerants. They found that R-32 offered potential benefits in its ability to increase capacity and efficiency. However, R-32 was shown to have a high discharge temperature when using a scroll compressor designed for R-410A. The authors noted that cycle and hardware modification would be required to improve the operating conditions for R-32.

In et al. (2014) [37] detailed experimental results from the partial optimization of a compressor using low-GWP refrigerants. They found that the R-32 system's compressor discharge temperature was 20-25% larger than the R-410A system.

Yamamoto et al. (2015) [39] described an experiment that compared an optimized swing compressor with R-32 to a compressor with R-410A. The compressor was optimized by creating spaces above and below the compression chamber to reduce the suction overheating losses. Additionally, the discharge port diameters were optimized in order to benefit from the reduction in compression loss. In the end, it was found that the COP for R-32 was 101.2% of that of R410A.

1.11.1.7 Other Components

Altogether, there has not been a lot of research and studies published on the work being done to optimize the expansion valves and other novel components in order to improve the performance of low-GWP refrigerants.

1.11.1.7.1 Valves

Barve and Cremaschi (2012) [34] conducted experiments and discussed the results of using an optimized expansion valve with R-32 and R-1234yf. Through their experiments, it was found that by optimizing the expansion valve, the capacity for R-1234yf was improved by up to 10%.

1.11.2 Conclusions

This report presents information surrounding the current status of component optimization of low-GWP refrigerants in residential air conditioning systems. This is an extension of facts and evidence that is intended to help the decision on a possible solution to R-410A.

Much of the effort has been spent on the optimization of tube-fin and microchannel heat exchangers. For tube-fin heat exchangers, improvements have been made to reduce the refrigerant charge and improve the cooling capacity. One approach has been to utilize small-diameter tubing. It was reported that the small-diameter tubing has the ability to reduce the cost by up to 60% without reducing the performance. On the other hand, microchannel heat exchangers have also been investigated. One of the major advantages of microchannel heat exchangers is their ability to reduce the refrigerant charge. It was mentioned that microchannel condensers reduce the refrigerant charge by 30% compared to a tube-fin.

A lot of work has also been spent on compressors. For each type of compressor (scroll, screw, rotary, swing types). The objective has been the same: to decrease the discharge temperature. Many of the reviews illustrated how the low-GWP refrigerants increase the discharge temperature. In fact, many suggested further optimizations of hardware and cycle along with utilizing compressor vapor injection. Additionally, there has been a lot of attention devoted to finding a lubricant/oil for Low-GWP applications. One of the major contenders is just optimizing current POEs for the new refrigerants or using an advanced POE.

Based on the summaries, the following areas are recommended for the future research:

- Optimization of heat exchangers with the sole use of low-GWP refrigerants
- Evaluate the benefits of employing a compressor vapor injection system
- Optimizing expansion valves for improved system performance
- Comprehensive evaluation of best lubricants/oil for new refrigerants

1.11.3 Heat Exchanger Optimization

1.11.3.1 Introduction

Tube-fin heat exchangers (TFHXs) are prominent components in air conditioning and heat pump systems. This type of heat exchanger consists of a bundle of tubes with fin sheets. The performance of TFHXs is greatly affected by a large number of structural parameters (tube diameter, tube length, fin type, fin thickness, etc.), and therefore, conducting optimization on such parameters can significantly improve its performance (Huang et al., 2015). In addition to those parameters, the configuration of tube connections, i.e. refrigerant circuitry, determines the refrigerant flow path and also significantly impacts the heat exchanger performance (Chwalowski et al., 1989; Wang et al., 1999; Bigot et al., 20 0 0; Liang et al., 2001; Ding et al., 2011; Ye and Lee, 2012; Joppolo et al., 2015). For an existing TFHX design with a given geometry, performing optimization on refrigerant circuitry is more convenient and cost-effective than varying other structural parameters. For example, changing the circuitry is a matter of changing U-bend length and orientation, whereas changing tube spacing can require new fin dies, which is costly and has a much longer lead time.

A novel Genetic Algorithm is developed and used to solve the tube-fin heat exchanger circuitry optimization problem. A validated finite volume heat exchanger model, CoilDesigner® (Jiang et al., 2006) is used to simulate the performance of TFHX with different circuitries. This model has been validated with multiple sets of measured data from different sources (Singh et al., 2008a, 2008b; Singh et al., 2011; Alabdulkarem et al., 2015). The primary contribution of this paper is to present a new integer permutation-based GA approach. This new approach can generate valid circuitry designs without requiring extensive domain knowledge. Manufacturing constraints are incorporated to guarantee that the optimal designs are manufacturable. The optimization method can handle various operating constraints, such as limits on pressure drop, refrigerant states, etc., and also handle various geometric constraints, such as limits on envelope volume, face areas, etc.

1.11.3.2 Integer permutation based Genetic Algorithm

1.11.3.2.1 Representation of HX circuitry

The first step in using a Genetic Algorithm to solve a problem is to find a way to mathematically represent a candidate solution as an individual in a population. A good individual representation (chromosome) can not only reduce the search-space but can also represent the design by simulating the nature of the problem (Griffiths et al., 2005). It has been shown that a meaningful and appropriate chromosomal representation of the problem can speed up Genetic Algorithm to converge to a global optimal (Kargupta et al., 1992). For the tube-fin heat exchanger circuitry optimization problem, the proposed approach represents tubes in a circuit as a sequence of integers, with each integer representing a tube in a given flow path. The optimization technique is independent of the actual numbering of the tubes. For an integer permutation, each integer (i.e., tube number) appears exactly once, thus, any chromosome generated by the Genetic Algorithm can be mapped to a valid circuitry and the size of the search space is dramatically reduced by the elimination of redundant and cyclic designs. It should be noted that the current implementation of IPGA does not include splitting and merging of circuits. Two different chromosome representations are developed and implemented in the new optimization framework. Consider a 15- tube, 4-circuit HX as an example (shown in Figure 1-105 (a)). The red circle indicates the inlet refrigerant streams and blue circle indicates the outlet refrigerant streams. A solid line represents a U-bend on the front end, while a dotted line represents a U-bend on the farther end of the heat exchanger. The first type of chromosome is called a “Two-Part Chromosome” as shown in Figure 1-105 (b), in which the first part of the chromosome denotes tube sequences, the second part denotes the number of tubes in each circuit. This type of chromosome works well when optimal number of circuits can be determined before conducting the optimization by using preliminary analysis such as the one presented in (Lee et al., 2016) or using rules of thumb or application of specific knowledge. However, in most of the applications, especially for new product designs, the optimal number of circuits cannot be easily derived, so a more general representation is de- sired. In this paper, the ‘Split Circuit Chromosome’, as shown in Figure 1-105 (c), is therefore proposed. The Split Circuit Chromosome uses the concept of jagged arrays, with each element of this array rep- resenting the tube sequence in each circuit. Each element can also contain a different number of tubes. With Split Circuit Chromosome, the number of circuits is flexible, i.e. the optimal number of circuits is also an output from optimization. Intuitively, Split Circuit Chromosome represents a refrigerant circuitry in a more realistic manner because it "physically" separates different circuits from each other.

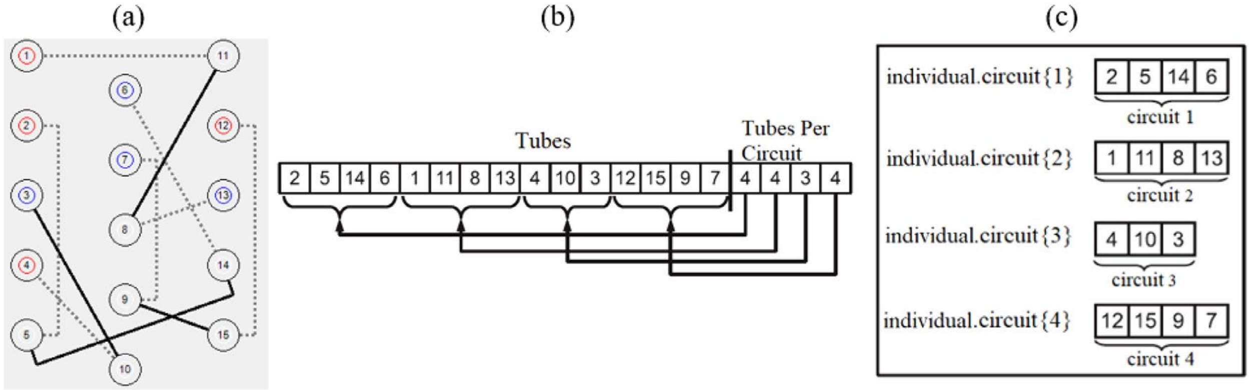

Figure 1-105: Circuitry representation: (a) 15-tube TFHX; (b) two-part chromosome; (c) split circuit chromosome

1.11.3.2.2 Selection, Crossover and Mutations

The selection operator selects superior individuals in a population and forms a mating pool. The common selection methods from Genetic Algorithm literature are tournament selection, proportionate selection (i.e., Roulette wheel selection) and ranking se- lection (Goldberg, 1989). The purpose of selection is to pick the above-average individuals from the current population and insert duplicates of those elite individuals in the mating pool in a probabilistic manner. In this paper, a tournament selection operator with tournament size 2 is used along with an efficient constraint handling method. Goldberg and Deb (1991) have shown that the tournament selection has equal or better convergence and computational time complexity properties compared with other selection operators in literature.

The selection operator selects good individuals, while the creation of new individuals relies on the genetic operators. Conventional Genetic Algorithm uses crossover and mutation operators. There exists a number of crossover and mutation operators (Spears and Jong, 1998). For the conventional crossover operator, two individuals are randomly picked from the mating pool and some portions of their chromosomes are exchanged to create two new individuals. Since each chromosome represents one heat exchanger and the chromosomal representation of heat exchanger circuitry has the characteristic that each integer appears exactly once, it is obvious that exchanging genes among two individuals will under- mine the structure of integer permutation and potentially generate many infeasible individuals. Coit and Smith (1996) have shown that constraint handling method such as penalty method cannot efficiently avoid the infeasible individual if Genetic Algorithm generates too many infeasible individuals than feasible ones and the optimization process will remain stagnant. For these reasons, the integer permutation-based chromosome requires new genetic operators. In this study, six novel genetic operators are developed. These six genetic operators are classified into two groups, 'in-circuit operators' and 'cross-circuit operators'. The first group of operators manipulates tubes (i.e., genes) inside one randomly selected circuit, while the second group manipulates tubes across different circuits. By transforming the selected individual to a new individual with potentially better fitness, these Genetic Algorithm operators direct the search and drive the optimization process.

1.11.4 Results

In order to address the above-mentioned concerns, different combinations of constraints are introduced in the optimization problem. Figure 1-106 shows the optimal solutions from three constrained optimization runs and Table 1-36 presents a detailed analysis of those optimal designs. In Table 1-36 , ‘U-bends L1’ and ‘U-bends L2’ are the number of U-bends that span 1 tube row and 2 tube rows, respectively. ‘U-bends ≥L3’ is the number of long U-bends which span more than 2 tube rows. Case (c) in Table 1-36 shows an optimization case in which manufacturing constraints, i.e., inlets and outlets on the same side, preventing long U-bends and preventing partial overlap U-bends crossovers, are enforced. Figure 1-106 (c) shows its optimal solution. This has resulted in a 13.8% capacity improvement. The optimal solution has 2 circuits, which still results in a 12 times higher pressure drop than the baseline. Nevertheless, the circuitry can be readily manufactured. Case (d) in Table 1-36 is another constrained optimization run in which only refrigerant pressure drop constraint is applied. This constraint restricts the designs to have equal or less refrigerant pressure drop than the baseline. Figure 1-106 (d) shows its optimal solution, which has 4 circuits, the same as the baseline, a 4% increase of capacity than the baseline, and a slightly lower pressure drop than the baseline. It is worthwhile to mention that some existing methods (Domanski et al., 2004; Ploskas et al., 2017) do not demonstrate pressure drop constraints, although various manufacturability constraints were investigated. Case (e) in Table 1-36 is a highly constrained optimization run in which both manufacturability constraints and the pressure drop constraint are applied. Figure 1-106 (e) shows its optimal solution, which achieves a 2.4% capacity increase and 1% fewer pressure drops than the baseline. The circuitry is manufacturable without any long U-bends or U-bend crossovers. Compared with other cases, the optimal solution from case (e) has the desired thermal, hydraulic, and manufacturability performance. The optimal evaporators in cases (d) and (e) provide a similar amount of latent cooling as a baseline since all three evaporators have similar sensible heat ratios (SHR). Since GA is a stochastic optimization approach, different runs can converge into different solutions. To assess the robustness of the proposed optimization approach, the aforementioned optimization case study with four different constraint combinations (case (b) to case (e) in Table 1-36) are repeated 15 times per case. The small variations of results between multiple optimization runs for each case indicate that the proposed approach has good robustness.

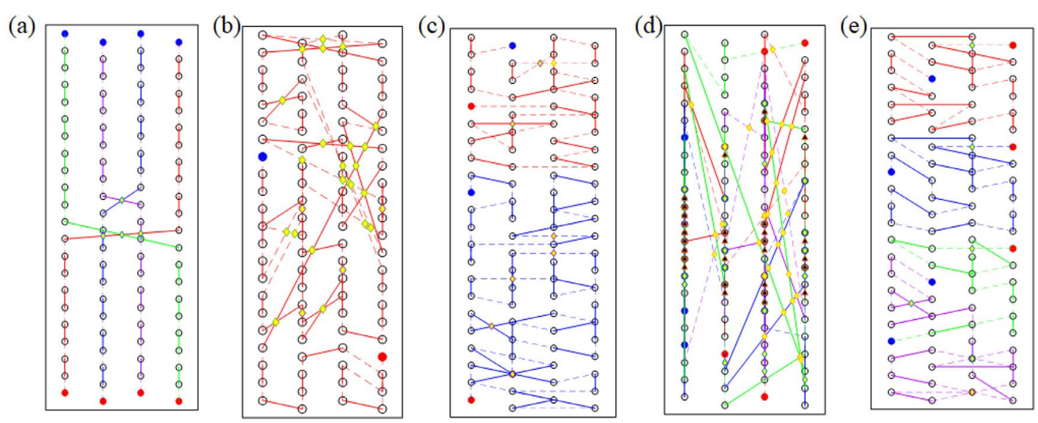

Figure 1-106: Optimal circuits under different constraints (a) baseline; (b) unconstrained; (c) with mfg. constraints; (d) with refrigerant DP constraint; (e) with mfg. and refrigerant DP constraints.

Table 1-36: Evaporator optimization results with different constraints

Case Constraints	Baseline (a) –	(b) No constraints	(c) Mfg. constraints	(d) Refrigerant DP constraint	(e) Mfg. constraints and DP constraint
Capacity [W]	5294	6065(14.6%↑)	6027(13.8%↑)	5497 (3.8%↑)	5421 (2.4%↑)
Ref. DP [kPa]	11.8	972.5 (81x↑)	160.5 (12x↑)	11.4 (3.6%↓)	11.7 (1.0%↓)
SHR	79.6%	67.6%	72.5%	79.8%	80.6%
U-bends L1	82	61	51	17	62
U-bends L2	0	4	35	9	22
U-bends ≥ L3	2	2	0	58	0
Collinear U-bends	0	3	0	43	0

1.11.5 Conclusions

A novel integer permutation-based GA approach is presented to solve the tube-fin heat exchanger circuitry optimization problem. Six novel genetic operators are designed to generate feasible circuitries with potentially better performance. The manufacturability aspect is handled using an efficient constraint-dominated sorting method, which avoids long and overlapping U-bends with adjustable U-bend length limits. The case studies on an experimentally validated evaporator show that the proposed optimization approach can generate circuitry designs with capacities superior to circuitries designed manually while guaranteeing good manufacturability. An exhaustive search on a small heat exchanger was used to verify that the proposed Integer Permutation-based GA can find optimal or near-optimal refrigerant circuitry designs using a relatively low population size and number of iterations. Overall, a capacity increase of 2.4–14.6% is observed with different constraints. Comparison with other optimization methods in the literature shows that the proposed approach can find designs that are better than optimal designs obtained from other methods.

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1.12 Overall Conclusions

This comprehensive report has examined the transition to low-GWP refrigerants within the HVAC&R industry, covering state-of-the-art technologies, case studies, design optimizations, and future outlooks. Each section of the report provides valuable insights and recommendations to guide the industry toward a more sustainable and environmentally friendly future.

The HVAC&R industry is making significant strides in adopting low-GWP refrigerants, driven by regulatory mandates and the pressing need to reduce global warming impacts. The availability of A1 refrigerants is limited, with CO_2 and R-466A showing promise but requiring further validation and system design modifications. A2L refrigerants, particularly R-32 and its mixtures, are extensively researched and offer a balanced performance. The development of advanced heat exchangers, compressors, and expansion valves enhances system efficiency and reliability. Regulatory frameworks like the Kigali Amendment, EU F-gas regulations, and the US EPA SNAP program are pivotal in driving this transition, while comprehensive safety research ensures the safe adoption of new refrigerants.

The case studies presented in this report highlight the practical implementations of low-GWP refrigerants, demonstrating their potential as viable alternatives to high-GWP refrigerants. R-516A shows comparable performance to R-134a with significant emission reductions, while unitary air-conditioning systems with refrigerants like R-290, R-32, R-452B, and R-466A exhibit promising results. Design guidelines emphasize optimizing heat exchangers and system components to enhance efficiency, safety, and compliance with regulatory standards. These findings underscore the importance of system efficiency and annual leakage as primary factors influencing emissions, with refrigerant manufacturing emissions playing a minor role.

LCCP assessments provide critical insights into the environmental impacts of HVAC systems. Factors such as weather data, power factors, manufacturing emissions, and material usage significantly influence LCCP calculations. The findings indicate that annual energy consumption and refrigerant leakage are primary contributors to LCCP, with regional variations highlighting the need for accurate local data. Low-GWP refrigerants like R-290, R-32, R-452B, and R-466A offer effective solutions for reducing emissions, with R-290 showing the lowest LCCP. Future work should focus on expanding evaluations to include new refrigerants, increasing regional comparisons, and studying the relationship between LCCP and cost.

Looking ahead to 2030, the HVAC&R industry must continue its efforts to evaluate promising refrigerants, conduct comprehensive safety research, and develop zero or near-zero GWP refrigerants. Regulatory trends indicate increasing adoption of low-GWP refrigerants, with ongoing updates to safety standards and policies facilitating this transition. Technological advancements, including innovative system designs and refrigerant blends, will drive future progress. Optimization techniques will enhance component performance and system efficiency, contributing to the industry's sustainability goals. Continued research, regulatory support, and technological innovation are essential to achieving significant environmental benefits and meeting future sustainability objectives.

The HVAC&R industry's transition to low-GWP refrigerants is imperative for reducing environmental impact and complying with regulatory mandates. This report highlights the current state of technologies, regulatory landscape, design optimizations, and future research directions. Continued efforts in research, safety assessments, and technological innovations are crucial to achieving a sustainable and environmentally friendly future for the HVAC&R industry. By embracing these advancements and adhering to evolving regulations, the industry can significantly mitigate its contribution to global warming and pave the way for a greener, more sustainable future.