2021 Progress Annual Report

HPT Annex 54:
Heat Pump Systems with Low-GWP Refrigerants

TASKS 1, 2 AND 3

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Executive Summary

This report provides a comprehensive, most up-to-date review of current research and the development of systems and their optimizations using low-GWP refrigerants for heat pump applications. It also includes a state-of-the-art review of life cycle evaluations of such systems using low-GWP refrigerants. It summarizes the collective efforts by researchers, engineers, and regulation committees across the industry. The report is comprised of the following chapters.

Chapter 1 provides two representative studies from teams in the United States. The first part summarizes the activities related to establishing a detailed database of alternative lower GWP refrigerants for various HVAC&R applications, including thermodynamic and transport properties. The goal is to identify the most suitable candidates for replacing traditional refrigerants based on thermodynamic performance and environmental benefits. It also considers the extent of potential system adjustments required to accommodate the new alternative refrigerants. The second part provides a comprehensive investigation of unitary air conditioners' life cycle climate performance. The review focused on life cycle climate performance methodologies, impacts of parameter and methodology selections, and a few representative case studies.

Chapter 2 provides a comprehensive update on research and development activities in Italy. Three research entities report their progress on low GWP refrigerants and their applications in various heat pump systems. The chapter covers condensation and boiling heat transfer coefficients measurements, and multiple experimental-oriented projects on long-term evaluations of novel heat pump systems using low GWP refrigerants.

Chapter 3 mainly describes the work in Japan, the first step of a two-step process on life cycle climate performance evaluation of heat pump-type air conditioners with next-generation refrigerants. It also presents an overview of a project to establish a new concept and hypothesis for life cycle climate performance evaluation, in which field data related to air conditioners is adopted.

Chapter 4 presents a study carried out by the France teams on finned tube heat exchangers using low GWP refrigerants. The study assessed the heat transfer performance during evaporation and condensation of R410A, R454B, and R32 in a finned tube heat exchanger. A 30-kW experimental setup was built to assess the heat exchanger performance with these three refrigerants. The simulations show that the same design of the finned tube heat exchanger can be used for R410A and R454B, but a design optimization is necessary with R32.

Chapter 5 presents a high-level summary of R&D progress across multiple institutions in Germany. Part one is a summary of the most recent large-scale heat pump monitoring project, a review of ongoing heat pump projects based on an analysis of the enArgus database, and a survey on heat pumps and their refrigerants as part of the market incentive program coordinated at the Federal Office for Economic Affairs and Export Control. In part two, within the last five years, the activities for heat pump research have changed. The activities cover more fundamental research up to the application of deployable heat pump demonstrators for white goods (e.g., dishwashers), mobile systems for electric-driven buses, or large capacity heat pumps systems for multi-family houses.

This report aims at providing a much-needed review and updates on component R&D using low-GWP refrigerants for heat pump applications. We hope it can be a good reference for researchers, engineers, and policymakers across the HVAC industry. We greatly appreciate the contributions of the authors for each chapter. The report would not exist without their valuable efforts.
Errata

Section 5.1 (and herein subsection 5.1.4) for the update on the German Task 1 activities contains wrongly associated data. The data presented are related to 2020 sales data on eligible heat pumps and not to sales data from 2019. Figures and texts are updated accordingly.
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1 Country Report: United States

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1.1 Thermodynamic Cycle Comparison of Alternative Refrigerants by Energetics/Oak Ridge National Laboratory

1.1.1 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 their thermodynamic and transport properties. The goal is to identify the most suitable candidates for replacement of traditional refrigerants based on 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.

1.1.2 Methods and 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-, medium-, and high-temperature refrigeration. Air-to-air, air-to-water, water-to-air, and water-to-water configurations were evaluated when relevant. Table 1 provides a summary of the refrigerants and alternatives included in the study. The thermodynamic properties for 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 2.

Table 1. Traditional refrigerants and the evaluated alternatives

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<th>Baseline Refrigerant</th>
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<td>R-123</td>
<td>R-1233zd(E) R-1224yd(Z)</td>
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<td></td>
<td>R-452A R-454A R-457A R-449C</td>
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<tr>
<td>R-404A</td>
<td>R-448A R-449A</td>
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<td>R-516A R-515B R-513A</td>
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<td>R-450A R-444A R-1234ze(E) R-1234yf</td>
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<td>R-410A</td>
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<td>R-452B R-463A R-468C</td>
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<td>R-245fa</td>
<td>R-1336mzz(Z) R-1233zd(E) R-1234ze(E)</td>
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Table 2. Assumptions for thermodynamic analysis

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<tr>
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<th>Tcond</th>
<th>Tevap</th>
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<td>-10</td>
<td>10</td>
<td>5</td>
<td>40</td>
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<td>-25</td>
<td>10</td>
<td>5</td>
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1.1.3 Thermodynamic Analysis Results

Analyses were broken into the categories by refrigerant, application, and medium, as indicated in Table 2. The categories 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)

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
1.1.3.1 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).

![GWP vs. Boiling Point for R-123 and evaluated alternatives, including ASHRAE 34 classification](image)

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

The P vs. -1/T graph (Figure 2) 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.

![Pressure vs. -1/T for R-123 and alternatives](image)

Figure 2. Pressure vs. -1/T for R-123 and alternatives
Figure 3. 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 2). 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 shown in the P-h diagram (Figure 4) show that the operating pressures of the evaporator and condenser are higher for the alternatives than those required for R-123. This may enable the use of a smaller compressor. Since the higher operating pressures require the use of thicker tube walls but allow the use of smaller compressors, the chiller cost of the alternatives may be similar to that of the baseline refrigerant. The refrigerant cycles also demonstrate differences in capacity, but it is important to consider the densities of these substances in order to examine capacity on a volumetric basis. The volumetric capacity can be observed in Figure 3, as well as the comparison in Figure 5.

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.
1.1.3.2 R-134a: General Comparison

All R-134a alternatives result in a significant reduction in GWP, as shown in Figure 6. 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 GWPsl 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 6. GWP vs. Boiling Point for R-134a and evaluated alternatives, including ASHRAE 34 classification

Based on the pressure vs. -1/T plot (Figure 7), 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 7. 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 8). These trends can be more closely examined based on medium type in the next section. Note that this analysis was conducted for only one of the condensing temperatures given in the assumption summary (Table 2). The general trend can be captured at any of the condensing temperatures.

1.1.3.3 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 9), 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) have lower pressures for both the evaporator and condenser, which can also be observed in Figure 7. These lower operating pressures could potentially enable 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 on the evaporator operating pressures of the alternatives, and the condenser operating pressures are lower by around 10% (Figure 10). The condenser pressures become higher for the mobile air conditioning application, while evaporator pressures slightly drop compared to air-to-water chillers (Figure 11).
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Figure 9. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for air-to-water chillers

Figure 10. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for water-to-water chillers
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 (Figures 12 and 13). 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 consider many other factors, including GWP, ASHRAE 34 classification, and other physical properties.

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 14).
Figure 12. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for air-to-water chillers.

Figure 13. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for water-to-water chillers.
1.1.3.4 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 15). The gap becomes smaller as pressure increases (Figures 16 and 17). 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 each of the cycles reduce by about 50 kJ/kg.

Figure 14. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for mobile air conditioning

Figure 15. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for low-temperature refrigeration
Figure 16. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-134a and alternatives for mid-temperature refrigeration

Figure 17. 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 (Figures 18 to 20). 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 baseline for low-temperature applications and around 74% of 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 volumetric capacities 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 18. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for low-temperature refrigeration

Figure 19. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for mid-temperature refrigeration
Figure 20. Relative volumetric capacity ($Q_v$) and coefficient of performance (COP) of R-134a alternatives compared to the baseline for high-temperature refrigeration

1.1.3.5 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 21). 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 21. GWP vs. Boiling Point for R-245fa and evaluated alternatives, including ASHRAE 34 classification

The P vs. -1/T graph (Figure 22) 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.
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 (Figures 21 and 23). 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 evaporator pressure would be slightly higher compared with R-245fa.

R-1233zd(E) and R-1234ze(Z) have COPs exceeding that of the baseline by about 6% and 5%, respectively (Figure 24). 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 24. Relative volumetric capacity (Q_v) and coefficient of performance (COP) of R-245fa alternatives compared to the baseline for high-temperature heat pumps](image)

**1.1.3.6 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 25). 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.

![Figure 25. GWP vs. Boiling Point for R-404A and evaluated alternatives, including ASHRAE 34 classification](image)
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 (Figures 28-30). Based on the P vs -1/T plot (Figure 26), R-457A is the most dissimilar from R-404A thermodynamically.

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 27. Figures 31-33 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.

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 (Figures 28-30).
Figure 27. 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 2). The general trend can still be captured regardless of the difference in system conditions.

Figure 28. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-404A and alternatives for low-temperature refrigeration
Figure 29. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-404A and alternatives for mid-temperature refrigeration

Figure 30. 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 (Figures 31-33). R-457B has a COP and volumetric capacity significantly higher than 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 (Figures 31-33).

Figure 31. Relative volumetric capacity (Qv) and coefficient of performance (COP) of R-404A alternatives compared to the baseline for low-temperature refrigeration.
1.1.3.7 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 34). However, operating pressures of system components remain relatively the same, as shown by the cycle diagrams in Figures 37-40.
Annex 54, Heat pump systems with low-GWP refrigerants

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

**Figure 35.** 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 35). 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 36, 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.

![Graph showing volumetric capacity vs. evaporating temperature](image)

**Figure 36. Volumetric capacity (calculated using the suction density condition) vs. evaporating temperature for R-410A and alternatives. Note that this analysis was conducted for only one of the condensing temperatures given in the assumption summary (Table 2). The general trend can be captured at any of the condensing temperatures.**

### 1.1.3.8 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 (Figures 37-40). 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 37. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-air heat pumps

Figure 38. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-water heat pumps
Figure 39. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-air heat pumps

Figure 40. 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 the evaluated refrigerants have COPs within 1% of the baseline COP for all four heat pump configurations (Figures 41-44). 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 34). 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 41. 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 42. Relative volumetric capacity ($Q_V$) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-water heat pumps
1.1.3.9 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 cost. 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 45. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-air air conditioning

Figure 46. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for air-to-water air conditioning
Figure 47. Pressure vs. Enthalpy (P-h) and refrigeration cycle plot for R-410A and alternatives for water-to-air air conditioning

Figure 48. 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 QV. 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 49.** Relative volumetric capacity (QV) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-air air conditioning.

**Figure 50.** Relative volumetric capacity (QV) and coefficient of performance (COP) of R-410A alternatives compared to the baseline for air-to-water air conditioning.
### 1.1.4 Summary and Discussion

In summary, across the different applications, relative QV follows a similar trend with small fluctuations. The performance ranking from best to worst on QV is: R-32, R-466A, R-410A, R-452B, R-454B, R-463A, R-468C, R-454A, R-454C (Figure 53). For each refrigerant category, there are viable lower-GWP alternatives with thermodynamic performances comparable to the baseline refrigerant, as summarized in Table 3.
### Figure 53. Summary of relative volumetric capacities of R-410A alternatives for all applications evaluated

### Table 3. Summary of alternative refrigerants that meet GWP, COP, and volumetric capacity requirements

<table>
<thead>
<tr>
<th>Traditional Refrigerant</th>
<th>Application</th>
<th>Short-Term Alternatives¹</th>
<th>Long-Term Alternatives²</th>
<th>Meets or Exceeds Baseline COP (99.9%+)</th>
<th>Acceptable Volumetric Capacity (≥ 90% of Baseline)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mobile Air Conditioning</td>
<td>Heat Pumps</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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.
### Heat Pumps

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Environments</th>
<th>Refrigerant Capabilities</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-410A</td>
<td>R-32</td>
<td>R-452B, R-454B, R-466A (mid- and high-temp only)</td>
</tr>
<tr>
<td></td>
<td>R-32, R-452B, R-454A, R-454C, R-466A (air-to-air and water-to-air only), R-468C</td>
<td>R-32, R-452B, R-454B, R-463A, R-466A, R-468C (air-to-air and water-to-air only)</td>
</tr>
</tbody>
</table>

### Air Conditioning

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Environments</th>
<th>Refrigerant Capabilities</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-410A</td>
<td>R-32, R-463A</td>
<td>R-452B, R-454B, R-466A, R-468C (excluding water-to-air), R-468C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-32, R-452B, R-454B, R-463A, R-466A, R-468C</td>
</tr>
</tbody>
</table>

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 refrigerator with a lower GWP may have a significantly lower volumetric capacity, requiring investment in a larger compressor. In another case, a refrigerator may perform very well on COP but poorly on volumetric capacity. Flammability can also vary between alternatives, and this may become a deciding factor despite performance differences.

#### 1.1.5 Conclusions

The data as presented throughout this report can be used to determine a “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 phase of this work, four pieces of candidate equipment will be selected, and they will be rigorously analyzed with the relevant traditional and alternative refrigerants.
1.2 Comprehensive Investigations on Life Cycle Climate Performance of Unitary Heat Pump Systems by the University of Maryland

1.2.1 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 Green House Gas (GHG) emissions from the direct refrigerant leakage, indirect fuel consumptions, and the embodied equipment emissions (Andersen et al., 2018). 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 4. Horie et al. (Horie et al., 2010) assessed the LCCP of the residential heat pump in Japan. Zhang et al. (Zhang et al., 2011) developed an LCCP tool for a residential heat pump for four U.S. cities. Li (G. Li, 2015a) evaluated the LCCP of various Packaged Air Conditioners (PAC) involving micro-channel heat exchangers for typical U.S. cities. Troch et al. (Troch, 2016) and Lee et al. (Lee et al., 2016) conducted an LCCP evaluation for the same heat pump system in five U.S. cities. Choi et al. (Choi et al., 2017) developed an LCCP model and evaluated it for South Korean weather conditions. Wu and Jiang (WU and JIANG, 2017) developed an LCCP calculation software to analyze different climate zones in China. Kim et al. (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 refrigerant 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 CO₂e 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 bring discrepancies in 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 an 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, 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 4. Recent LCCP evaluation research

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

1.2.2 Methodologies

1.2.2.1 LCCP Calculation Process

Troch et al. (Andersen 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 emissions and indirect emissions and is typically calculated in kg CO$_2$e unit, as shown in eq. 1:

$$LCCP = Direct\ emissions + Indirect\ emissions$$  

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:

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

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$_2$e/kg), $Adp.GWP$ means GWP of Atmospheric Degradation Product of the Refrigerant (kg CO$_2$e/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:

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

Where $AEC$ means Annual Energy Consumption (kWh); $E.M.$ means CO$_2$ produced/kWh (kg CO$_2$e/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$_2$e/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$_2$e/kg); $RFM$ means refrigerant manufacturing emission (kg CO$_2$e/kg); $RFD$ means refrigerant disposal emissions (kg CO$_2$e/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.
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.

### 1.2.2.2 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 54 (Gilmour and 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 5 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.
Table 5. Target Refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Safety Category</th>
<th>GWP</th>
<th>Drawback</th>
<th>Announced Year</th>
<th>Manufacturing Emissions (kg CO₂e/kg)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-410A (baseline)</td>
<td>A1</td>
<td>2,088</td>
<td>High GWP</td>
<td>Honeywell, 1991</td>
<td>10.7</td>
<td>(Goto et al., 2001; Wang et al., 2009)</td>
</tr>
<tr>
<td>R-466A (N41)</td>
<td>A1</td>
<td>730</td>
<td>High cost</td>
<td>Honeywell, 2018</td>
<td>-</td>
<td>(Devecioğlu &amp; Oruç, 2020; Honeywell Announces R410A Breakthrough, 2018; Honeywell's N41 – a Blast from the Past, 2018)</td>
</tr>
<tr>
<td>R-32</td>
<td>A2</td>
<td>675</td>
<td>Mildly flammable</td>
<td>-</td>
<td>7.2</td>
<td>(Mota-Babiloni et al., 2017; Pham &amp; Rajendran, 2012; Xu et al., 2013)</td>
</tr>
<tr>
<td>R-452B (DR-55)</td>
<td>A2</td>
<td>676</td>
<td>Mildly flammable</td>
<td>Ingersoll Rand, 2015</td>
<td>8.9</td>
<td>(Kedzierski &amp; Kang, 2016; S. A. Kujak et al., 2014)</td>
</tr>
<tr>
<td>R-290</td>
<td>A3</td>
<td>3</td>
<td>Flammable</td>
<td>-</td>
<td>0.05</td>
<td>(AHRTI-9007-02_Final_Report.Pdf, n.d.; Wu et al., 2012)</td>
</tr>
</tbody>
</table>

1.2.2.3 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.

1.2.2.3.1 System Performance

We developed models of a 10.55 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 55. 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 6 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 55. Cooling experiments validation results (Alabdulkarem et al., 2014)](image)

Table 6. Design compressor displacement volume and predicted charge level

<table>
<thead>
<tr>
<th>Capacity</th>
<th>Refrigerant</th>
<th>Compressor RPM</th>
<th>Compressor Displacement Volume (cm³)</th>
<th>Charge (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.5 kW</td>
<td>R-410A</td>
<td>4,700</td>
<td>34</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>R-290</td>
<td>7,300</td>
<td>43</td>
<td>1.2</td>
</tr>
<tr>
<td></td>
<td>R-32</td>
<td>4,000</td>
<td>34</td>
<td>3.5</td>
</tr>
<tr>
<td></td>
<td>R-452B</td>
<td>4,700</td>
<td>34</td>
<td>3.8</td>
</tr>
<tr>
<td></td>
<td>R-466A</td>
<td>4,800</td>
<td>34</td>
<td>4.2</td>
</tr>
<tr>
<td>12.3 kW</td>
<td>R-410A</td>
<td>4,700</td>
<td>47</td>
<td>4.1</td>
</tr>
<tr>
<td></td>
<td>R-290</td>
<td>7,300</td>
<td>61</td>
<td>1.8</td>
</tr>
<tr>
<td></td>
<td>R-32</td>
<td>4,000</td>
<td>47</td>
<td>3.6</td>
</tr>
<tr>
<td></td>
<td>R-452B</td>
<td>4,700</td>
<td>47</td>
<td>3.9</td>
</tr>
<tr>
<td></td>
<td>R-466A</td>
<td>4,800</td>
<td>34</td>
<td>4.3</td>
</tr>
</tbody>
</table>
Figure 56 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 56. Cooling COP comparisons for different refrigerants](image)

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.

1.2.2.3.2 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 is applicable 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 were 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 7. 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 (Wijeysundera, 2015) to estimate the cooling and heating load of the target room.

Table 7. Parameters for simulation

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

1.2.2.4 Grid Electricity Emission Factor and Material Embodied Carbon Coefficients

The emissions due to 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 8 shows the GEFs used in this study.

Table 8. GEF, Material usage, and ECCs

<table>
<thead>
<tr>
<th>Weight: 115 [kg]</th>
<th>GEF [kg CO$_2$e/kWh]</th>
<th>ECCs [kg CO$_2$e/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material (% usage)</td>
<td>Aluminum (12%)</td>
<td>Copper (19%)</td>
</tr>
<tr>
<td>Average around world</td>
<td>0.623</td>
<td>13.1</td>
</tr>
<tr>
<td>EU</td>
<td>0.277</td>
<td>6.58</td>
</tr>
<tr>
<td>Sweden (SE)</td>
<td>0.012</td>
<td>5.65</td>
</tr>
<tr>
<td>Switzerland (CH)</td>
<td>0.014</td>
<td>10.60</td>
</tr>
<tr>
<td>NA</td>
<td>0.425</td>
<td>11.90</td>
</tr>
<tr>
<td>US FL</td>
<td>0.467</td>
<td>14.60</td>
</tr>
<tr>
<td>US AZ</td>
<td>0.457</td>
<td>11.90</td>
</tr>
<tr>
<td>US GA</td>
<td>0.457</td>
<td>14.60</td>
</tr>
<tr>
<td>AS</td>
<td>0.492</td>
<td>14.60</td>
</tr>
<tr>
<td>JP</td>
<td>0.517</td>
<td></td>
</tr>
<tr>
<td>KR</td>
<td>0.517</td>
<td></td>
</tr>
<tr>
<td>CN</td>
<td>0.623</td>
<td>13.1</td>
</tr>
</tbody>
</table>

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. (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. (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 8, 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.

1.2.2.5 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 from 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. (de 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 58 shows the comparisons between the two data. Figure 57 (a) shows the daily temperature measured on January 15, 2019. Figure 57 (b) shows the daily temperature tested on July 15, 2019. The blue line is the temperature tested in the campus reading through LabView, 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 AC 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 58 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 58.
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 57. Comparison of Ambient Temperatures](image)

(a) January 15 Ambient Temperature  
(b) July 1st Ambient Temperature

Figure 57. Comparison of Ambient Temperatures

![Figure 58. Histogram of the year 2019 ambient air temperature in College Park, MD, U.S.](image)

Figure 58. Histogram of the year 2019 ambient air temperature in College Park, MD, U.S.

1.2.3 Results

1.2.3.1 Different Countries and Regions

Figure 59 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 1.2.3.3. 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 (G. Li, 2015b) concluded that the SEER rating had a far more significant impact on lowering CO$_2$e. Nevertheless, based on our study, this conclusion is only valid in the countries with a high GEF.
1.2.3.2 Different Refrigerants

Figure 59 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. (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. (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 CO₂e, 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 CO₂e per kg. We assumed the value to be 5, 10, and 20 kg of CO₂e per kg for assumptions 1, 2, and 3, respectively. The LCCP calculation results are shown in Table 9. 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 61, 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 60. Different refrigerant LCCP results for four selected cities](image)

Table 9. R-466A Refrigerant Manufacturing Process Emission Effect

<table>
<thead>
<tr>
<th>City</th>
<th>Assumption 1</th>
<th>Assumption 2</th>
<th>Assumption 3</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>RFM (kg CO₂e)</td>
<td>LCCP (kg CO₂e)</td>
<td>Percent</td>
</tr>
<tr>
<td>Beijing, CN</td>
<td>31</td>
<td>94164</td>
<td>0.03%</td>
</tr>
<tr>
<td>Shanghai, CN</td>
<td>31</td>
<td>69482</td>
<td>0.04%</td>
</tr>
<tr>
<td>Tokyo, JP</td>
<td>31</td>
<td>61524</td>
<td>0.05%</td>
</tr>
<tr>
<td>Kallax, SE</td>
<td>31</td>
<td>4246</td>
<td>0.73%</td>
</tr>
<tr>
<td>Basel, CH</td>
<td>31</td>
<td>4314</td>
<td>0.72%</td>
</tr>
<tr>
<td>London, UK</td>
<td>31</td>
<td>61524</td>
<td>0.05%</td>
</tr>
<tr>
<td>Atlanta, US</td>
<td>31</td>
<td>61518</td>
<td>0.05%</td>
</tr>
<tr>
<td>College Park, US</td>
<td>31</td>
<td>70087</td>
<td>0.04%</td>
</tr>
<tr>
<td>Miami, US</td>
<td>34</td>
<td>98865</td>
<td>0.03%</td>
</tr>
<tr>
<td>Phoenix, US</td>
<td>34</td>
<td>91276</td>
<td>0.04%</td>
</tr>
</tbody>
</table>

1.2.3.3 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 61 shows the results. From Figure 61, the UMCP campus ambient temperature was always higher than that from the College Park airport. This brings a higher emission in the summer but a lower emission in the winter. Figure 61 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 8% decrease for LCCP calculation. If onsite ambient data is not available, a correction on the ambient temperature is recommended.
Annex 54, Heat pump systems with low-GWP refrigerants

### 1.2.4 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.

### 1.2.5 Acknowledgment

We gratefully acknowledge the support of the Center for Environmental Energy Engineering (CEEE) at the University of Maryland.
1.2.6 Reference

AHRTI-9007-02_Final_Report.pdf (n.d.).
Annex 54, Heat pump systems with low-GWP refrigerants


Pham, H. M. and Rajendran, R. (2012). R32 And HFOs As Low-GWP Refrigerants For Air Conditioning. 11.
2 Country Report: Italy

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2.1 Summary

This report briefly summarizes the activities carried out during the 3rd year of Annex 54 in Italy. Due to the Covid-19 emergency, the research activities slightly slowed down. Despite this, advances in the use of low GWP refrigerants in heat pumps have been made, as detailed in the following sections.

NB: Since the Italian team consists of three different research groups, the document is organized in three sections, one for each research group.

2.2 Activities at National Research Council

2.2.1 Thermodynamic properties of low GWP refrigerants

As more extensively described in the last year's country report, the main part of the activities of ITC-CNR on Low-GWP refrigerants is devoted to the measurements of some thermodynamic properties to evaluate efficiency analysis of hydrofluoroolefins (HFOs). In particular, the laboratory of refrigerants and nanofluids is equipped with various instruments capable of measuring the main properties of refrigerants: saturated pressure of pure fluids, vapor liquid equilibria (VLE) of binary mixtures, compressed liquid density of both pure and mixed refrigerants, mutual solubility of refrigerants and lubricants, liquid, and vapor thermal conductivity for both pure and mixed refrigerants. Due to the restrictions induced by the COVID-19, no further measurements could be performed in the last year. However, at present, measurements of VLE for the binary mixture R32+R1234yf and for the thermal conductivity of R1234ze(E) are in progress: the results will be available in the next weeks and will be presented in the next report.

In 2018, the ITC CNR research group performed a thorough analysis of the literature to evaluate the amount of experimental data available for the main thermodynamic and transport properties of low GWP refrigerants. These data are essential to developing the accurate Equations of state (thermodynamic properties) and dedicated equations (transport properties) necessary to properly design the components of the heat pump systems and to evaluate their performance. It was highlighted that only few refrigerants (namely R1234yf, R1234ze(E), R1234ze(Z) and R1233zd(E)) were already sufficiently studied, while for all the other pure low GWP refrigerants there were scarce or null information. Even worse were the situation for refrigerant mixtures.

This year, an update of this review was performed to evaluate the progress obtained in the last three years. Table 1 synthetizes the results of the review. For a given fluid, green, yellow, and red areas identify a property for which more than 1 set of data, 1 set or no sets, respectively, are available.

With respect to 2018, for only two new fluids (R1336mzz(Z) and R1224yd(Z)) a sufficient amount of data is available for almost all the properties. For eight fluids (R1243zf, R1123, R1123a, R154mzy(E), R1225ye(Z), R1336mzz(E), RE356mzz and R1354myf(E)) there is a discreet or scarce amount of data, but several properties are not studied yet. For the other 5 fluids considered, no information at all is available in the literature. It is evident that still a consistent research activity is necessary to cover the lack of data for the pure low GWP refrigerants. As refer the mixtures, the situation is even worse. Recently, a review on the experimental data available for mixtures has been performed by Bell et al. (2021)1. They showed that most of the potential mixtures formed by

HFOs have not been studied at all at the moment and that only few data are available for a restricted amount of mixtures.

### Table 1 - Sets of data available in the open literature on the thermodynamic and transport properties of low GWP pure refrigerants.

<table>
<thead>
<tr>
<th>ASHRAE Designation</th>
<th>Thermodynamic Properties</th>
<th>Other Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Data</td>
<td>Critical point</td>
<td>Saturated pressure</td>
</tr>
<tr>
<td>Sets Data</td>
<td>Sets Data</td>
<td>Sets Data</td>
</tr>
<tr>
<td>R1234ze (E)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1234yf</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>R1233zd(E)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1234ze(Z)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R136mzz(Z)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1224yd(Z)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1243zf</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>R1123</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>R1132a</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1354mzy(E)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R1225ye(Z)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>R136mzz(E)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>RE356mzz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1354myf(E)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1132(E)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1132(Z)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1141</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1225yze(E)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1234yze(E)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### 2.2.2 Heat pumps

As described in the last country report, ITC-CNR has been recently involved in two Horizon2020 EU projects dedicated to the development of ground source heat pumps (GSHPs) in all Europe (CHEAP-GSHPs and GEO4CIVHIC projects). The last one is still in progress and its deadline has been postponed to October 2023. CNR-ITC is involved in a specific activity aimed at identifying low-GWP refrigerants as substitutes for R134a (low pressure refrigerants) and R410A (high-pressure refrigerants). To perform the selection of the most promising substitutes, a homemade software based on Matlab® environment and coupled with Refprop 10.0 for the calculation of the thermodynamic properties has been developed. For each fluid, an energetic and exergetic analysis of the performance (in particular COP and volumetric heating/cooling effect (VHE)) with several
heat pump cycles schemes has been performed, assuming as boundary conditions those typical for geothermal applications in the different regions of Europe. The first part of the analysis has been performed on “transition” fluids characterized by GWP lower than the current HFCs (GWP<800). Various potential fluids have been considered after a survey of the open literature. Table 2 and Table 3 reports the fluids considered and their basic characteristics.

**Table 2 - Characteristics of the potential substitutes for R410A (high pressure fluids).**

<table>
<thead>
<tr>
<th>Fluid</th>
<th>GWP</th>
<th>ASHRAE Safety Class</th>
<th>Composition (wt %)</th>
<th>T(_{\text{crit}}) (K)</th>
<th>P(_{\text{crit}}) (MPa)</th>
<th>T Glide (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R410A</td>
<td>2088</td>
<td>A1</td>
<td>R32/R125 (50/50)</td>
<td>343.32</td>
<td>4.770</td>
<td>0.05</td>
</tr>
<tr>
<td>R32</td>
<td>675</td>
<td>A2L</td>
<td>R32 (100)</td>
<td>351.55</td>
<td>5.816</td>
<td>-</td>
</tr>
<tr>
<td>R454B</td>
<td>466</td>
<td>A2L</td>
<td>R32/R1234yf (68/31.1)</td>
<td>350.15</td>
<td>5.041</td>
<td>1</td>
</tr>
<tr>
<td>R452B</td>
<td>698</td>
<td>A2L</td>
<td>R32/R125/R1234yf(67/7/26)</td>
<td>350.25</td>
<td>5.200</td>
<td>0.9</td>
</tr>
</tbody>
</table>

**Table 3 - Characteristics of the potential substitutes for R134a (low pressure fluids).**

<table>
<thead>
<tr>
<th>FLUID</th>
<th>GWP</th>
<th>NBP (ºC)</th>
<th>Safety class (ASHRAE)</th>
<th>Components</th>
<th>Compositio (wt%)</th>
<th>T glide (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>REFERENCE</td>
<td>R134a</td>
<td>1300</td>
<td>-26.1</td>
<td>A1</td>
<td>R32/R125</td>
<td>50/50</td>
</tr>
<tr>
<td>POTENTIAL ALTERNATIVES</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R513A</td>
<td>631</td>
<td>-29.2</td>
<td>A1</td>
<td>R1234yf/R134a</td>
<td>56.0/44.0</td>
<td>0</td>
</tr>
<tr>
<td>R515A</td>
<td>393</td>
<td>-18.0</td>
<td>A1</td>
<td>R1234ze(E)/R227ea</td>
<td>88/12</td>
<td>0</td>
</tr>
<tr>
<td>R515B</td>
<td>293</td>
<td>-19.1</td>
<td>A1</td>
<td>R1234ze(E)/R227ea</td>
<td>91.1/8.9</td>
<td>0</td>
</tr>
<tr>
<td>R516A</td>
<td>142</td>
<td>-29.4</td>
<td>A2L</td>
<td>R134a/R1234yf/R152a</td>
<td>8.5/77.5/14</td>
<td>0</td>
</tr>
</tbody>
</table>

The main results obtained till now can be synthetized as follows:

1. Within high pressure fluids, R454B resulted to be the most promising substitute for R410A in all the cycles analyzed;
2. Within the low-pressure fluids, the most efficient fluid resulted to be R516A (+10% COP with reference to R134a);
3. The main drawback of these fluids is that they are mildly flammable fluid (A2L ASHRAE classification) and this forced to take suitable safety measures, with potential higher costs.

The next step of the analysis will be the identification of very low GWP refrigerants (GWP<150) to meet the new regulations on F-gas. These fluids, probably mixtures rather than pure refrigerants, will have to guarantee an energy efficiency similar to that of traditional refrigerants and possibly a low or null flammability.
One of the activities of CNR ITC within the Geo4Civic project is the monitoring of two geothermal heat pumps installed in the demo site at the CNR Research Area in Padova. The refrigerant R454B is used in both heat pumps as working fluid. The heat pumps are identical but are endowed with two different control systems: one on-off, the other with inverter. The scope of the activity is testing the efficiency of the two heat pumps along one year of operation to evaluate the different energetic behavior of the two control systems by evaluating the heat exchange at the source (geothermal wells) and the sink (user) side by monitoring secondary fluids temperatures and flow rates at the corresponding heat exchangers. Moreover, the heat pump with inverter is endowed with sensors to measure pressures, temperatures, and the refrigerant mass flowrate at the characteristic points of the cycle. Figure 1 shows the arrangement of the sensors on the refrigerant and secondary fluids circuits.

The heat pumps and the monitoring system are at present under testing. The first-year acquisition program will start in the next weeks.

![Figure 1 - Scheme of the monitoring system for the heat pump with inverter.](image)

### 2.3 Activities at University of Padua

The research activity performed at the Department of Industrial Engineering (University of Padova) during 2021 has been focused on condensation heat transfer measurements with low-GWP fluids and on the experimental investigation of a solar assisted heat pump working with CO₂.

#### 2.3.1 In-tube condensation with low-GWP refrigerants

R1234ze(E) has emerged in the recent years as low Global Warming Potential substitute for R134a in heat pumps. As a drawback, R1234ze(E) is classified as a mildly flammable fluid (A2L
class, ANSI/ASHRAE classification) and, in the search for non-flammable alternatives to R134a, hydrofluorocarbon/hydrofluoroolefin binary mixtures have been considered.

Low-GWP refrigerant mixtures that fall in the A1 non-flammable category are frequently sought by the industry since they can be used as drop-in fluids in existing systems and, in the case of new installations, they do not require more stringent safety measures. Among R134a low-GWP substitutes that belong to A1 class, mixtures R450A (R1234ze(E)/R134a at 58.0/42.0% by mass composition) and R515B (R1234ze(E)/R227ea at 91.1/8.9% by mass) can be identified. Most of the studies available in the literature on R450A and R515B focus on the employment of these fluids as drop-in replacements in vapor compression refrigeration systems working with R134a. Less attention has been addressed to the condensation heat transfer coefficient of these fluids, especially inside minichannels. Furthermore, there is a lack of experimental studies on HFCs and HFOs mixtures providing two-phase heat transfer data together with flow pattern visualizations. These complementary data are fundamental for the development of new heat transfer correlations, particularly at low mass flux conditions (below 100 kg m⁻² s⁻¹) which can be encountered in vapor compression systems with inverter-driven compressors.

Condensation tests have been performed at the University of Padova with R1234ze(E) and with binary non-flammable mixtures R450A and R515B inside two channels with inner diameter equal to 3.38 mm and 0.96 mm. R515B is an azeotropic mixture (GWP 100-years = 299) whereas R450A is a near-azeotropic mixture with 0.6 K temperature glide at 40 °C dew temperature (GWP 100-years = 547).

The test rig used during the experimental campaign consists of a primary refrigerant loop (oil-free) and three auxiliary water circuits. In the refrigerant loop, the subcooled liquid exits from a tube-in-tube heat exchanger (post-condenser) which operates with distilled water as secondary fluid, and it enters a magnetic-driven gear pump. The refrigerant is then vaporized and superheated inside a tube-in-tube heat exchanger (evaporator). After the evaporator, the refrigerant enters the 0.96 mm diameter or the 3.4 mm diameter test section where it is partially or fully condensed. The 3.4 mm diameter test section consists of two tube-in-tube heat exchangers for heat transfer measurements separated by a 70 mm long borosilicate glass tube for flow pattern visualizations. Flow pattern visualizations have been recorded using a high-speed camera coupled with macro lens and a LED illumination system.

![Figure 2 - Two-phase flow pattern visualizations of R1234ze(E) at 40 °C saturation temperature inside the 3.4 mm inner diameter horizontal tube. The selected frames correspond to mass velocity \( G = 100 \text{ kg m}^{-2} \text{ s}^{-1} \) and five different values of vapour quality.](image)

Figure 2 shows flow patterns recorded during condensation of R1234ze(E) at mass velocity equal to 100 kg m⁻² s⁻¹ and different vapour quality. Stratified wavy flow can be observed for a wide range of vapour qualities \( x \). From \( x = 0.75 \) to \( x = 0.20 \), the film thickness at the bottom progressively
increases and the interfacial waves become large enough to reach the upper part of the tube. It can be noticed that, with the decreasing vapour quality, the liquid-vapour interface between two large amplitude waves becomes smooth. Considering the whole range of mass velocities and all the three fluids investigated (R1234ze(E), R450A, R515B) four regimes were detected in the 3.38 mm diameter channel: annular, stratified-wavy, stratified-smooth and slug.

Heat transfer coefficients have been measured at 40 °C saturation temperature and mass velocity from 40 kg m$^{-2}$ s$^{-1}$ to 600 kg m$^{-2}$ s$^{-1}$. At the same mass velocity, vapor quality and channel diameter, R1234ze(E), R450A and R515B display similar values of heat transfer coefficient. For each fluid, at the same conditions of vapor quality and mass velocity, the heat transfer coefficient is found to increase when reducing the channel diameter. The prediction accuracy of condensation heat transfer models has been assessed against the experimental results.

Adiabatic two-phase frictional pressure drops have been measured inside the 0.96 mm minichannel at mass velocity equal to 200 and 400 kg m$^{-2}$ s$^{-1}$. At given mass velocity and vapor quality, R1234ze(E) and R515B present the same pressure gradient, whereas R450A shows a lower pressure drop.

The comparison among different refrigerants during condensation should also consider the pressure drop of the fluid, which affects the saturation temperature and thus the mean effective driving temperature difference in the condenser. To allow the comparison among R134a and the drop-in candidates R1234ze(E), R515B and R450A, a performance evaluation criterion has been adopted.

### 2.3.2 Solar assisted heat pump working with CO$_2$

Solar assisted heat pumps (SAHPs) exploit solar energy as the low-temperature thermal source. There are two types of SAHPs: indirect solar assisted heat pumps (IDX-SAHP) where a secondary fluid is heated up by solar collectors and then it is sent to the evaporator, and direct solar assisted heat pumps (DX-SAHPs), where the solar collectors act as the evaporator. However, a solar evaporator is not able to absorb enough energy in the case of low or absent solar irradiance and thus a dual source heat pump working both with solar and air source can be used to guarantee the operation of the heat pump regardless of the presence of solar irradiance. Furthermore, the use of hybrid photovoltaic-thermal (PV-T) solar collectors in DX-SAHPs, allows the system to achieve higher photovoltaic (P.V.) conversion efficiency due to the cooling of the cells.

With the recent restrictions to the use of high GWP refrigerants, the use of environmentally friendly or natural fluids, such as CO$_2$, is increasingly growing. However, in the literature there are few experimental studies on DX-SAHPs working with CO$_2$ as the refrigerant, and there are no experimental works on DX-SAHPs working with PV-T solar collectors and CO$_2$ as operative fluid.

Starting from this background, a novel direct expansion solar assisted heat pump prototype working with CO$_2$ has been installed at the Department of Industrial Engineering. The study is realized in the framework of the Solair-HP CSEA project. The prototype has a nominal heating capacity equal to 5 kW at the maximum compressor speed. The dual source heat pump can work with air or solar energy as low temperature source at the evaporator. The solar collector is a hybrid photovoltaic module coupled with a sheet-and-tube heat exchanger that works as the evaporator.

The heat pump has been tested in real environmental conditions to investigate its performance when working in air mode with the finned coil evaporator and when working in solar mode with the PV-T evaporator. The PV-T evaporator has been studied to determine its thermal efficiency curve and a model based on the experimental characterization of the heat pump has been developed.
2.4 Activities at Polytechnic of Milan

During 2021, the research activities carried out at Polytechnic of Milan went on slowly since a general revision of the experimental set-up was made. As a result, the analysis of low-GWP refrigerants was limited to R515B and the comparison with R134a was made in some selected conditions. The experimental set-up and the experimental methodology used to assess the performance of HFO described in 2020 country report were used to assess the performance of that refrigerant. For the sake of clarity, the layout the experimental set-up, the main characteristics of its components and the main characteristics of the instrumentation are reported in Figure 4, Table 4 and Table 5, respectively. More information is available in 2020 country report.

Table 4 - Main characteristics of the refrigerant loop

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Swept volume @ 50 Hz</td>
<td>13.15 m³/h</td>
</tr>
<tr>
<td></td>
<td>Shaft rotational frequency</td>
<td>30 Hz - 87 Hz</td>
</tr>
</tbody>
</table>
The test rig is used to compare the performance of R515B to those achieved with R134a and R1234ze(E) in a drop-in application. The testing conditions are reported in Table 6 and consist of 5 tests at constant evaporator inlet and outlet temperatures and variable condenser inlet and outlet temperatures.

Table 5 - Measurement instrumentation range and accuracy.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Instrument</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant mass flow rate</td>
<td>Coriolis mass flow meter</td>
<td>0 kg/h - 300 kg/h</td>
<td>±0.15% r.v.</td>
</tr>
<tr>
<td>Refrigerant pressure (low side)</td>
<td>Pressure transducer</td>
<td>0 kPa - 700 kPa</td>
<td>±0.3% f.s.</td>
</tr>
<tr>
<td>Refrigerant pressure (high side)</td>
<td>Pressure transducer</td>
<td>0 kPa - 4000 kPa</td>
<td>±0.3% f.s.</td>
</tr>
<tr>
<td>Refrigerant temperature</td>
<td>RTD Pt 100</td>
<td>243.15 K - 373.15 K</td>
<td>±0.1K</td>
</tr>
<tr>
<td>Compressor power</td>
<td>Power transducer</td>
<td>0 W - 4000 W</td>
<td>±0.2% f.s.</td>
</tr>
<tr>
<td>Water mass flow rate</td>
<td>Vortex flow meter</td>
<td>0.21 m³/h - 3 m³/h</td>
<td>±2% r.v.</td>
</tr>
<tr>
<td>Water temperature</td>
<td>RTD Pt 100</td>
<td>263.15 K - 353.15 K</td>
<td>±0.1K</td>
</tr>
</tbody>
</table>

The heating capacity and the COP of the heat pump are shown in Figure 5 and Figure 6 respectively. Starting from the heating capacity, it is possible to state that the use of R515B leads to a reduction of the heat pump heating capacity. The capacity reduction is quite large since with this refrigerant a capacity in the range 78%-80% is achieved. Conversely, the trend of the COP is opposite since a slight increase, in the range 103%-105%, is found.
Figure 5 - Heat pump heating capacity as a function of condenser outlet temperature for the three refrigerants considered.

Figure 6 - Heat pump COP as a function of condenser outlet temperature for the three refrigerants considered.

2.5 REFERENCES


3 Country Report: Japan

Japan’s progress in the year 2021 on LCCP Evaluation for Air-to-Air Heat Pumps using Next-Generation Refrigerants
(Report 1: First Step) Residential Air Conditioners

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3.1 Summary

In Japan, a two-step process is applied to the LCCP evaluation of heat pump-type air conditioners with next-generation refrigerants. This report mainly describes the first step of the process through to the LCCP evaluation methods, together with the concept of the study utilizing field data and hypotheses.

In particular, a representative model is selected as an example of general split-type air conditioners, and it is considered in two ways, one of which is only as evaluation for a system drop-in of candidate refrigerants replacement while the other is in terms of system optimization, both using the candidate refrigerants to be examined R290, R32, R454C (and R22, R410A). Based on such systems, this report explains a calculation method using the performance simulation that is adopted as a standard tool by the Japan Refrigeration and Air Conditioning Industry Association (hereinafter referred to as “JRAIA”).

The report also presents an overview of a project to establish a new concept and hypothesis for LCCP evaluation in which field data related to air conditioners is adopted.

3.2 Introduction

The main issues of recent urgent environmental efforts to address global warming in relation to air conditioners (hereinafter referred to as “AC”) are the Kigali Amendment to the Montreal Protocol in globally, the F-Gas Regulation in Europe, and the Act on Rational Use and Proper Management of Fluorocarbons in Japan.

The Kigali Amendment is a regulation aimed at gradually reducing the production and consumption amounts of refrigerants used in CO2 equivalents. This regulation is a global warming countermeasure to be promoted worldwide by focusing on a transition to lower GWP refrigerants. For actual global warming countermeasures, it is important not only to reduce the GWP of the refrigerant, but also to improve the performance of equipment by reducing the amount of greenhouse gas emissions derived from power consumption.

Figure 1 gives an overview of the forecast of demand for residential AC cooling by 2050 by country/region based on the Future of Cooling report published by the International Energy Agency (hereinafter referred to as “IEA”) [1]. As can be seen, the chart indicates that the world’s demand for residential AC cooling will expand rapidly by 2050. The use of AC in the United States and Japan will increase at a gradual pace. On the other hand, due to growing demand in India, Indonesia, Brazil, China, and EU countries, the world’s AC demand is projected to rise considerably – by more than three times – by 2050.

It is anticipated that such an increase in AC demand will not only cause refrigerants to have a direct impact on global warming but also possibly give rise to an increase in the indirect impact on global warming due to the power consumption of AC equipment. Therefore, in addition to the direct impact of refrigerants, the energy efficiency and power consumption of equipment will become the focus of even greater attention in the years to come.
For this reason, we believe the LCCP evaluation to be studied in Task 3 is an important evaluation for selecting the most suitable refrigerants because it takes into account the transition to lower GWP refrigerants and the environmental impact of power consumption. To make the evaluation more realistic, based on the concept of S+3E (Safety, Environment Performance, Energy Efficiency, Economic Feasibility) advocated by JRAIA, it is desirable to conduct a comprehensive evaluation from multiple perspectives, including safety, cost, sustainability, and infrastructure development, in addition to environmental assessment through the LCCP evaluation.

The study conducted by JRAIA in December 2008 [2] is introduced as a previous case study relating to the LCCP evaluation. The LCCP evaluation in this case was based on a simplified simulation, with climate and other conditions set in accordance with Japanese Industrial Standards (hereinafter referred to as “JIS”).

There is also a case study concerning IEA-related LCCP evaluation, which was presented by the University of Maryland in the United States (hereinafter referred to as “UMD”) [3]. In this paper, mainstream residential ACs in the United States are evaluated; therefore, it is necessary to conduct an evaluation for the split-type heat pump ACs that are in conventional residential use in Japan and Asia.

Accordingly, in Japan, a new LCCP evaluation is carried out in two steps for heat pump-type ACs that use next-generation refrigerants. The first step (this report) examines the LCCP evaluation methods, and the second step to be implemented in the future will mainly describe a new evaluation applying performance simulation and the concept and hypothesis of the study utilizing market data.

In the evaluation in this report, a representative model of typical split-type ACs is selected, and refrigerants R290, R32, R454C, R22, and R410A are examined. Since the indirect impact of power consumption varies significantly depending on the market and factors such as climate condition and lifestyle, the first step evaluates this under the standard conditions in Japan. The performance evaluation method verified jointly by JRAIA and Waseda University is used. As the second step, evaluation will be conducted based on the possibility of the application of actual market data that varies according to local climate conditions, and Report 2 will explain the established concept and hypothesis about this approach.
3.3 Concept of LCCP Evaluation

This chapter explains the concept of the LCCP evaluation. Basically, the calculations of LCCP are carried out in accordance with the guidelines published by the International Institute of Refrigeration (hereinafter referred to as “IIR”) [4]. The components to be evaluated for LCCP are shown in Figure 2.

This chapter describes how to calculate the amount of refrigerant charge and annual energy consumption.

![LCCP Components](image)

3.3.1 Concept of Candidate Refrigerants

This section explains the concept of candidate refrigerants for examination in the LCCP evaluation in this project.

The candidate refrigerants include R410A, a pseudo-azeotropic refrigerant mixture of an HFC refrigerant, and R32, an HFC single refrigerant, both of which are used in current AC units. In addition, R454C was selected as a zeotropic mixture of HFO and HFC refrigerant with relatively large temperature glide, which is attracting attention as a low-GWP refrigerant. We also selected the natural refrigerant R290, which has a lower operating pressure than R410A and R32, and the HCFC refrigerant R22, which is still adopted in many current ACs in emerging nations and is an important refrigerant for comparison with R290.

Table 1 shows the properties of each refrigerant [5] and Table 2 shows the theoretical COP calculated based on the thermodynamic properties of the refrigerants.

As described above, five refrigerants were selected for the study from the perspective of comprehensively covering the properties of the next-generation refrigerants in relation to system performance: R410A, R32, R454C, R290, and R22.
3.3.2 Examined AC and Performance Simulation

The AC to be examined is a split-type heat pump AC. Widely distributed ACs are diverse according to the manufacturer and development year; therefore, in this project, we decided to examine a residential AC for which JRAIA defined the standard specifications.

More specifically, the selected AC is equivalent to a high-end unit, and an analysis model (standard model) with a rated cooling capacity of 4 kW was created for the examination. An overview of the AC’s refrigerant circuits is shown in Figure 3.

The performance simulation was conducted using the simulation software Energy Flow +M, which was developed by Professor Saito’s Laboratory at Waseda University and is used by JRAIA as a standard tool. Figure 4 shows the refrigerant circuit diagram of the standard model constructed with this performance simulation software.

### Table 1 Comparison of Refrigerant Properties

<table>
<thead>
<tr>
<th>Composition</th>
<th>R410A</th>
<th>R32</th>
<th>R454C</th>
<th>R290</th>
<th>R22</th>
</tr>
</thead>
<tbody>
<tr>
<td>R32/R125 (50/50 wt%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>GWP</td>
<td>2090</td>
<td>675</td>
<td>148</td>
<td>3</td>
<td>1810</td>
</tr>
<tr>
<td>Safety Label</td>
<td>A1</td>
<td>A2L</td>
<td>A2L</td>
<td>A3</td>
<td>A1</td>
</tr>
</tbody>
</table>

### Table 2 Comparison of Theoretical COP

<table>
<thead>
<tr>
<th>Evaporating Pressure MPa</th>
<th>R410A</th>
<th>R32</th>
<th>R454C</th>
<th>R290</th>
<th>R22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing Pressure MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature glide °C</td>
<td>0.1</td>
<td>0.0</td>
<td>5.6</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Suction Temperature °C</td>
<td>15.1</td>
<td>15.0</td>
<td>18.5</td>
<td>15.0</td>
<td>15.0</td>
</tr>
<tr>
<td>Discharge Temperature °C</td>
<td>71.6</td>
<td>84.3</td>
<td>59.4</td>
<td>58.5</td>
<td>71.7</td>
</tr>
<tr>
<td>Theoretical COP</td>
<td>5.49</td>
<td>5.63</td>
<td>6.02</td>
<td>5.88</td>
<td>5.94</td>
</tr>
<tr>
<td>Volume Capacity kJ/m³</td>
<td>6,404</td>
<td>7,045</td>
<td>3,700</td>
<td>3,737</td>
<td>4,456</td>
</tr>
<tr>
<td>Evaporation temperature Cooling10/Heating −5°C,</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Suction superheat 5K, Sub-cooling 10K</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Saturation temperature of zeotropic refrigerants is midpoint temperature of two-phase region under constant pressure.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Condensing temperature Cooling50/Heating35°C,*

Evaporation temperature Cooling10/Heating0°C,

Suction superheat 5K, Sub-cooling 10K
To examine this standard model with greater accuracy, a comparative verification was made between the performance simulation results and the actual equipment test results for the case where the R410A was used for operation. Table 3 shows a comparison with the actual equipment test results. To implement more advanced performance simulation, each element device consisting of the refrigeration cycle was calibrated so that both the capacity and power input in Table 3 have an accuracy of ±10%.
3.3.3 Performance Simulation Conditions

This section describes the calculation conditions for performance simulation shown in Table 4. The performance test conditions for ACs have been evaluated by means of JIS C 9612 (2013). Its performance evaluation method is carried out through a relative comparison under the predetermined conditions for the balance point (condensation and evaporation temperatures, suction superheating degree, and subcooling degree), which is the operating status point.

AC performance is evaluated by calculating an Annual Performance Factor (hereinafter referred to as "APF"). To calculate the APF, operation modes are set for each of cooling and heating, and the capacity is set for each of the test conditions and a Coefficient of Performance (hereinafter referred to as "COP") is calculated. AC performance is evaluated based on a relative comparison using the APF.

These evaluations are conducted for each case of using a single refrigerant and using an azeotropic (including pseudo-azeotropic) refrigerant mixture, and for the latter performance is evaluated using a technique called the “cycle midpoint protocol (boiling point/dew point)” as specified in JIS B 8623 (2019), which defines condensation and evaporation temperatures.

In addition, the expansion valve opening degree is adjusted so that the suction superheating degree (= compressor suction gas temperature – saturation temperature in compressor suction gas) is 5°C, and performance is evaluated using the same methods described above. Regarding the subcooling degree, the amount of refrigerant charge is adjusted so that the maximum COP is achieved for each refrigerant.

### Table 3 Calibration results of Standard model

<table>
<thead>
<tr>
<th></th>
<th>Cooling</th>
<th></th>
<th>Heating</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actual</td>
<td>Calculated</td>
<td>Actual</td>
<td>Calculated</td>
</tr>
<tr>
<td>Capacity</td>
<td>%</td>
<td>100</td>
<td>95</td>
<td>100</td>
</tr>
<tr>
<td>Power consumption</td>
<td>%</td>
<td>100</td>
<td>111</td>
<td>100</td>
</tr>
</tbody>
</table>

### Table 4 Calculation conditions

<table>
<thead>
<tr>
<th>TEST</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Entering</td>
<td>Cooling</td>
<td>Heating</td>
</tr>
<tr>
<td>of Indoor unit</td>
<td>Dry Bulb[°C]</td>
<td>27.0</td>
</tr>
<tr>
<td></td>
<td>Wet Bulb[°C]</td>
<td>19.0</td>
</tr>
<tr>
<td>Air Entering</td>
<td>Cooling</td>
<td>Heating</td>
</tr>
<tr>
<td>of Outdoor unit</td>
<td>Dry Bulb[°C]</td>
<td>35.0</td>
</tr>
<tr>
<td></td>
<td>Wet Bulb[°C]</td>
<td>24.0</td>
</tr>
<tr>
<td>Cycle Point</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Capacity</td>
<td>Same at each operation point</td>
<td></td>
</tr>
<tr>
<td>SH</td>
<td>5.0 K at Suction Temperature</td>
<td></td>
</tr>
<tr>
<td>SC</td>
<td>At Optimum Performance</td>
<td></td>
</tr>
</tbody>
</table>
3.3.4 Calculation of LCCP

Basically, LCCP is calculated in accordance with the guidelines published by the IIR. The calculation methods for the amount of refrigerant charge, annual energy consumption, and other items are arranged and implemented as follows.

(1) Equipment: JRAIA AC standard model (H/P split type) equivalent to high-end unit with a rated cooling capacity of 4 kW

(2) Tool: JRAIA standard tool Energy Flow +M (Prof. Saito’s Lab at Waseda University)

(3) Comparison: Optimization is performed for each refrigerant to calculate the amount of refrigerant charge and the annual energy consumption.

(4) Selection of refrigerant types: Five refrigerants (R410A, R32, R454C, R290, and R22) were selected based on the above-mentioned concept.

(5) Calculation conditions: As described above, the power consumption in a refrigeration cycle with the same capacity is calculated under the conditions shown in Table 4.

3.4 LCCP Evaluation Conditions and Specifications

This chapter describes the evaluation conditions and specifications for LCCP.

3.4.1 Definitions of LCCP Equations

The definitions of the LCCP equations are stipulated as the LCCP evaluation conditions. LCCP is calculated by obtaining the sum of the direct and indirect emissions according to the method proposed by Dr. Hwang of UMD [3]. The method of calculating the direct emissions is shown in Equation 1. Indirect emissions are calculated using Equation 2. The following is a brief summary of the LCCP calculation.

\[
LCCP = \text{Direct Emissions} + \text{Indirect Emissions}
\]

\[
\text{Direct Emissions} = C \times (L \times ALR + EOL) \times (GWP + Adp. GWP)
\]  ... Equation 1

\[
\text{Indirect Emissions} = L \times AEC \times EM + \sum (m \times MM) + \sum (mr \times RM)
+ C \times (1 + L \times ALR) \times RFM
+ C \times (1 - EOL) \times RFD
\]  ... Equation 2

The symbols in the evaluation equations are shown in Table 5.
3.4.2 Influential Factors of LCCP

This section describes influential factors when calculating LCCP.

Table 6 summarizes the calculation items used in Equation 1 for the direct emissions and in Equation 2 for the indirect emissions in the LCCP calculation. Table 6 indicates that LCCP involves various influential factors. Therefore, when evaluating LCCP, it is necessary to clarify its influential factors and parameterize them for the study. There is particular concern that the evaluation may significantly differ depending on the country or region.

For example, regarding energy conversion, since the power generation systems vary by country or region, the data that comes under Equation 1 for the direct emissions and Equation 2 for the indirect emissions are deemed to be influential factor parameters for energy conversion. By replacing these influential factor parameters as appropriate for each country or region, it is possible to estimate valid LCCP.

### Table 5 Symbol Description

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C )</td>
<td>kg Refrigerant Charge</td>
</tr>
<tr>
<td>( L )</td>
<td>yr Average Lifetime of Equipment</td>
</tr>
<tr>
<td>( ALR )</td>
<td>% of Ref. Charge Annual Leakage Rate</td>
</tr>
<tr>
<td>( EOL )</td>
<td>% of Ref. Charge End of Life Refrigerant Leakage</td>
</tr>
<tr>
<td>( GWP )</td>
<td>kgCO(_2)e/kg Global Warming Potential</td>
</tr>
<tr>
<td>( Adp. GWP )</td>
<td>kgCO(_2)e/kg GWP of Atmospheric Degradation Product of the Refrigerant</td>
</tr>
<tr>
<td>( AEC )</td>
<td>kWh Annual Energy Consumption</td>
</tr>
<tr>
<td>( EM )</td>
<td>kgCO(_2)e/kWh CO2e Produced / kWh</td>
</tr>
<tr>
<td>( m )</td>
<td>kg Mass of Unit</td>
</tr>
<tr>
<td>( MM )</td>
<td>kgCO(_2)e/kg CO2e Produced / Material</td>
</tr>
<tr>
<td>( mr )</td>
<td>kg Mass of Recycled Material</td>
</tr>
<tr>
<td>( RM )</td>
<td>kgCO(_2)e/kg CO2e Produced / Recycled Material</td>
</tr>
<tr>
<td>( RFM )</td>
<td>kgCO(_2)e/kg Refrigerant Manufacturing Emissions</td>
</tr>
<tr>
<td>( RFD )</td>
<td>kgCO(_2)e/kg Refrigerant Disposal Emissions</td>
</tr>
</tbody>
</table>
3.4.3 Evaluation Applying Performance Simulation

To predict the performance of AC equipment, which will have a major impact on the LCCP evaluation, newly constructed performance simulation with improved prediction accuracy is used in the study.

Figure 4 shows the schematic configuration of the standard model used for performance simulation in this study. The standard model was created based on the specifications of a common commercially available heat pump AC.

In this study, to consider the design concept for actual ACs, the performance is predicted assuming drop-in (hereinafter referred to as “DI”) and soft optimization (hereinafter referred to as “SO”) in the performance simulation. Specifications optimized with greater awareness of AC product capabilities (e.g., size equivalence and energy efficiency equivalence) are also examined. The DI evaluation in this study refers to performance evaluation where the same AC equipment is used but only the refrigerant is replaced. However, regarding equipment mounted with an inverter compressor and electronic expansion valve, the compressor frequency is changed (change of refrigerant mass flow rate) with the same AC capacity, and the expansion valve opening degree is changed with the same suction superheating degree. In addition, the SO evaluation refers to performance evaluation with minor modifications added on the AC’s hardware, such as changing the diameter of the connection pipe so that almost the same pressure drop occurs in the circuit even when different refrigerants are applied. Table 7 shows the concept of the assumed specification changes. The refrigerant used for the base model is R410A.

As shown in Table 7, compared with the base AC (a), in (b) the charged refrigerant (R410A) was changed, the refrigerant amount was adjusted to equalize the subcooling degree, and the compressor frequency was also changed to equalize the capacity before comparison. In (c), for the connection pipes of the liquid side and gas side that connect the indoor and outdoor units, the pipe diameter was optimized so that the pressure drop is equal to that of the current refrigerant. In (d), while maintaining the size of the heat exchanger, the path was changed to achieve the maximum efficiency. In (e), the size of the heat exchanger was changed to equalize the energy efficiency.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>kg</td>
</tr>
<tr>
<td>$L$</td>
<td>yr</td>
</tr>
<tr>
<td>$ALR$</td>
<td>% of Ref. Charge</td>
</tr>
<tr>
<td>$EOL$</td>
<td>% of Ref. Charge</td>
</tr>
<tr>
<td>$GWP$</td>
<td>KgCO$_2$e/kg</td>
</tr>
<tr>
<td>$AEC$</td>
<td>kWh</td>
</tr>
<tr>
<td>$EM$</td>
<td>KgCO$_2$e/kWh</td>
</tr>
</tbody>
</table>
Based on these studies, the values for the refrigerant charge amount, $C$, and the annual energy consumption, $AEC$, which are factors affecting the LCCP evaluation, are calculated, and a comparative assessment is made using the obtained values.

### Table 7 Specification of refrigeration cycle

<table>
<thead>
<tr>
<th></th>
<th>(a)</th>
<th>(b)</th>
<th>(c)</th>
<th>(d)</th>
<th>(e)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Outdoor unit</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Column</td>
<td>35</td>
<td>←</td>
<td>←</td>
<td>←</td>
<td>Optimize</td>
</tr>
<tr>
<td>Cooling path</td>
<td>6-3-1</td>
<td>←</td>
<td>←</td>
<td>←</td>
<td>Optimize</td>
</tr>
<tr>
<td><strong>Indoor Unit</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Column</td>
<td>21</td>
<td>←</td>
<td>←</td>
<td>←</td>
<td>Optimize</td>
</tr>
<tr>
<td>Cooling path</td>
<td>1-3</td>
<td>←</td>
<td>←</td>
<td>←</td>
<td>Optimize</td>
</tr>
<tr>
<td><strong>Connection pipes</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Liquid</td>
<td>Φ6.35</td>
<td>←</td>
<td>Optimize (*1)</td>
<td>Same as (c)</td>
<td>Same as (c)</td>
</tr>
<tr>
<td>Gas</td>
<td>Φ9.52</td>
<td>←</td>
<td>Optimize (*1)</td>
<td>Same as (c)</td>
<td>Same as (c)</td>
</tr>
<tr>
<td><strong>Refrigerant charge [kg]</strong></td>
<td>1.1</td>
<td>COP</td>
<td>Maximum (*2)</td>
<td>COP Maximum (*2)</td>
<td>COP Maximum (*2)</td>
</tr>
<tr>
<td><strong>Capacity vs base model [%]</strong></td>
<td>100</td>
<td>←</td>
<td>←</td>
<td>←</td>
<td>←</td>
</tr>
<tr>
<td><strong>Expansion valve</strong></td>
<td>As it is</td>
<td>Same as Base model SH</td>
<td>Same as Base model SH</td>
<td>Same as Base model SH</td>
<td>Same as Base model SH</td>
</tr>
</tbody>
</table>

※Note 1: Same as temperature-equivalent pressure drops of base model
※Note 2: Proposal under consideration

### 3.5 Application of LCCP Evaluation and Consideration

With regard to the LCCP evaluation, this chapter formulates a hypothesis for the study with a view to the influential factors and the study of regions both in Japan and abroad and describes a consideration of these.

#### 3.5.1 Key Points of the Project in the LCCP Evaluation

Firstly, we will explain the key points of the project in the LCCP evaluation.

The LCCP evaluation is basically examined by utilizing field data in addition to the past study by JRAIA [2] and the paper by UMD [3].
The results of evaluation based on the actual operating conditions will be reported next time in the final report.

3.5.2 Application of the Project’s LCCP Evaluation

The concept of the LCCP evaluation in this project is shown in Figure 5, and the following explains how to apply the evaluation and other matters. The final report of the project will classify and study the regional characteristics of ambient air temperatures and other influential factors in Japanese regions that affect the LCCP evaluation. Therefore, we will study the LCCP evaluation through this task, having in mind that other countries will be able to make broad and general speculation.

(1) Impact of equipment specifications

As explained in Chapter 4, the LCCP values in this project are calculated for the standard model based on a high-end AC unit with a cooling capacity of 4 kW that is commercially available in Japan.

However, there is a wide variety of ACs on the Japanese market, and they are even more diversified in other countries. In our view, for such diverse equipment, it is possible to calculate the annual power consumption, namely the indirect emissions in the LCCP evaluation, by modifying the specifications in the refrigeration cycle simulation shown in Figure 4.

(2) Influence of regional characteristics

In addition to the modification of the equipment specifications mentioned in (1), the annual power consumption, namely the indirect emissions in the LCCP evaluation, also requires modifications of the estimation parameters according to environmental characteristics including the temperatures in the region, users’ usage conditions, differences in power generation systems, and other factors.

For reference, Figure 6 shows a comparison of the changes of monthly average temperatures around the world in 2020. Climates vary depending on the assumed regional environment, which results in differences in the ambient air temperatures and humidity, operating hours, and required
Annex 54, Heat pump systems with low-GWP refrigerants

load. These differences strongly affect the annual power consumption during the use of an AC – the indirect emissions of LCCP. For example, in a region where ACs are required to operate throughout the year, annual power consumption is projected to increase throughout the life of the equipment. Regarding the load on the equipment, the higher the ambient air temperature during the cooling mode, the higher the condensing pressure in the refrigeration cycle, increasing the load on the compressor. The heat load entering the building from outdoors also increases, and the room temperature tends to rise; therefore, it is necessary to select an AC with a higher capacity.

Figure 7 shows the composition of power generation systems in five countries in 2016. Power resources are classified into fossil fuels, such as petroleum, natural gas, and coal, and non-fossil fuels, such as hydraulic power, other renewable energies, and nuclear power, and they are compared in percentage terms. As indicated in Table 6, according to the resource of power generation means, LCCP is affected by E.M., which is the amount of CO₂ [kgCO₂] generated per unit power consumption [kWh]. The E.M. varies greatly depending on the major power generation systems in each country. For example, in countries where the major systems are renewable energy power generation, such as hydraulic power, and nuclear power generation, the power consumed by ACs will be unlikely to lead to CO₂ emissions, and the indirect emissions will be negligible. On the other hand, in countries where power generation is mainly derived from fossil fuels such as

Figure 6 Example of Monthly average temperatures around the world

Moreover, larger heat exchangers are required as the load increases, leading to the problem of an increase in the amount of refrigerant used. Therefore, it is anticipated that there will be demand for highly efficient refrigerants suitable for large capacity and other influences.
petroleum and natural gas, CO₂ emissions from the use of AC becomes sensitive, and the indirect emissions of LCCP are strongly affected by the equipment efficiency and air-conditioning load. Concerning the power generation systems, changes in the composition due to changes in power resource demand, shift to renewable energies and other factors may also need to be considered in future predictions.

In our view, by studying the basic values required for the LCCP calculation shown in Table 6, it is possible to apply the LCCP evaluation to the influential factors described above.

(3) Impact of the properties of various refrigerants

This section explains the concept concerning the LCCP evaluation methods for diverse candidate refrigerants.

As mentioned in Chapter 4 and this chapter, in this project, by utilizing new performance simulation with high accuracy and market data, the LCCP evaluation is expected to be closer to actual market conditions than the previous evaluation was.

The LCCP calculation for various refrigerants in this project is as described in Chapter 4. The method is to examine the factors that affect the LCCP evaluation of various refrigerants through SO or by optimizing the refrigerant circuit for each refrigerant.

For example, the values for the refrigerant charge amount, C, and annual energy consumption, AEC, which are considered major influential factors, can be calculated with high levels of accuracy. In terms of the calculation conditions, it is expected that LCCP for each refrigerant can be compared by reflecting market data and regional characteristics, such as climate and refrigerant recovery rate, which vary in different countries and regions.

3.6 Conclusions

In Japan, LCCP evaluation is studied in a two-step process, and the concept (hypothesis) of the first step described in this report is summarized in the following bullet points:
- The concept (hypothesis) of the LCCP evaluation in this report was established by focusing on the consistency of the criteria of evaluation indicators.
- In the LCCP evaluation, a performance simulation evaluation will be carried out using JRAIA’s standard tool (performance simulation program developed by Waseda University) to optimize the system in relation to candidate refrigerants.
- Performance evaluation will be carried out through a relative comparison for each refrigerant under the same capacity.
- A residential heat pump mini-split AC with a capacity of 4.0 kW was selected as the standard model of AC equipment to be examined, and calculation will be performed.
- For the standard model, mutual correction will be made between the actual equipment test data and the performance simulation data.
- The calculation conditions of LCCP were clarified to share the same data recognition.

In addition, in the second step, the LCCP evaluation will be studied by utilizing the market data on ACs.

Influential factors of the LCCP evaluation were clarified and parameterized so that the evaluation can be applied overseas.

Finally, the basic concept on the positioning of the LCCP evaluation in the selection of refrigerants is explained as follows.

We believe that the LCCP evaluation to be studied in Task 3 is one of the important criteria for selecting optimal refrigerants. We also feel that in order to make the evaluation more realistic, it is desirable to conduct a comprehensive evaluation from multiple perspectives, including safety, cost, sustainability, and infrastructure development, in addition to the environmental assessment through the LCCP evaluation.

### 3.7 Future Plans

Regarding issues and responses to be addressed in the second step in the future, we plan to verify the concept (hypothesis) of the LCCP evaluation described in this report.

Specifically, the study will be conducted based on the results of the performance simulation to optimize ACs. In addition, the utilization of market data will also be examined.

In the second step, we plan to study the LCCP evaluation not only for Japanese regions with a temperate climate but also, by utilizing market data, for other areas in the world such as India, where ACs are used more often.
3.8 Terminology

**Life Cycle Climate Performance (LCCP):** An index to evaluate the global warming impact of a product throughout its life, from manufacture to disposal. Based on the TEWI, the value is calculated by adding energy consumption (indirect emissions) when manufacturing the gas to be used and the leakage of the gas (direct emissions).

**Global Warming Potential (GWP):** Indicates the degrees of the global warming impacts of gases released in the atmosphere. Based on carbon dioxide (1.0), the same gas weight and same period (100 years) are assumed to allow relative comparisons of the impact of each gas. When the HFC refrigerant used for ACs is considered with the same gas weight, its GWP is generally hundreds to thousands of times greater than that of carbon dioxide; therefore, its significant impact on global warming is regarded as a problem.

**Optimization:** In performance simulation, compared with the performance prediction made with conventional DI and SO, optimized performance prediction considers the design concept of an actual product; optimization is carried out by adjusting the component parts (piping, heat exchanger, etc.) of the refrigeration circuit so that their specifications conform to the refrigerant characteristics.

**Performance simulation:** A method to estimate the power consumption of the target equipment under various operating conditions using Energy Flow +M (Prof. Saito’s Lab at Waseda University), which is a standard tool of JRAIA. Used to calculate annual power consumption.

**Coefficient of Performance (COP):** A factor used as a measure of the energy consumption efficiency of cooling equipment, etc. The value represents cooling/heating capacity per kW of power consumption.

**Total Equivalent Warming Impact (TEWI):** A method to evaluate global warming impacts in comprehensive consideration of refrigerant leakage during equipment use, emissions into the atmosphere at the time of disposal, and the amount of carbon dioxide generated from fossil fuel usage due to operating power consumption. TEWI is expressed by the following equation.

\[
TEWI = \text{Direct CO}_2 \text{ emission equivalent } + \text{Indirect CO}_2 \text{ emission equivalent}
\]

Direct \ CO\textsubscript{2} \text{ emission equivalent} = GWP \times L \times N + GWP \times M \times (1-\alpha)
Indirect \ CO\textsubscript{2} \text{ emission equivalent} = N \times E \times \beta

- **GWP:** Global warming potential per kg on the basis of CO\textsubscript{2}, 100-year integration period (kg-CO\textsubscript{2}e/kg)
- **L:** Annual amount of leakage from equipment (kg/year)
- **N:** Service life of equipment (years)
- **M:** Amount of charge to equipment (kg)
- **\alpha:** Recovery rate at equipment disposal
- **E:** Annual energy consumption of equipment (kWh/year)
- **\beta:** CO\textsubscript{2} emissions required for 1 kWh of power generation (kg-CO\textsubscript{2}e/kWh)

**CO\textsubscript{2} emissions:** Total of the amount of carbon dioxide generated from fossil fuel usage due to power consumption and the equivalent amount of carbon dioxide using GWP to the degree of the global warming impact of the gas released into the atmosphere.
3.9 References

[7] The data from webpage “Energy circumstances of each country” The Figure made by Japan Agency for Natural Resources and Energy and the original data are from “IEA World Energy Balances” https://www.enecho.meti.go.jp/about/whitepaper/2019html/1-2-3.html

3.10 Acknowledgments

This project is conducted by the following members and observers of JRAIA LCCP Evaluation Study W.G. We would like to express our deepest gratitude for their cooperation.

The Chief is Dr. Shigeharu Taira, Daikin Industries, Ltd., and the Sub-Chief is Mr. Seishi Iitaka, Panasonic Corporation.

Members are as follows: Mr. Itaru Nagata, Sharp Corporation; Mr. Tomoyuki Haikawa and Mr. Katsunori Murata, Daikin Industries, Ltd.; Mr. Hiroichi Yamaguchi and Mr. Kohei Maruko, Toshiba Carrier Corporation; Mr. Ryoichi Takafuji and Mr. Takashi Inoue, Johnson Controls-Hitachi Air Conditioning; Mr. Shunji Itakura, Fujitsu General Limited; Mr. Takanori Nakamura and Mr. Keisuke Mitoma, Mitsubishi Heavy Industries Thermal Systems, Ltd.; Dr. Koji Yamashita and Mr. Yasuhide Hayamaru, Mitsubishi Electric Corporation.

Observers are Professor Kiyoshi Saito and Mr. Yoichi Miyaoka, Chief Researcher, Waseda University; Mr. Takeshi Sakai and Mr. Kazuhiro Hasegawa, JRAIA.
4 Country Report: France

Prepared by:
Pierre PARDO*1

1CETIAT, Centre Technique des Industries Aérauliques et Thermiques, HVAC systems department, Villeurbanne, France

*Corresponding author: pierre.pardo@cetiat.fr
4.1 Summary

This report presents a study carried out by the CETIAT on finned tube heat exchanger using low GWP refrigerants. This work has been already presented during the International Congress of Refrigeration 2019 (Pierre PARDO, Nicolas GAILLARD, Michèle MONDOT, Experimental and numerical study of heat transfer in evaporation and condensation with R410A, R32 and R454B in a finned tube heat exchanger, Manuscript ID: 1439 DOI: 10.18462/iir.icr.2019.1439).

The progressive phase-out of HFCs and their replacement by the 4th generation of refrigerants require identifying the performance of these alternative refrigerants in heat exchangers. The objective of this study was to assess the heat transfer performance during evaporation and condensation of R410A, R454B and R32 in a finned tube heat exchanger. A 30-kW experimental setup was built to assess the heat exchanger performance with these three refrigerants. For each refrigerant, 6 tests were carried out in evaporation and 5 tests in condensation. During the tests, energy balance between the thermal capacities measured on the air and refrigerant sides was found to be less than 5%, which confirmed a good accuracy of measurements. A good agreement (+/-2.5%) between experimental data and simulations allowed using EVAP-COND software for comparison of the performance of a same heat exchanger design with the 3 refrigerants. The simulations show that the same design of finned tube heat exchanger can be used for R410A and R454B, but a design optimization will be necessary with R32.

Keywords: Low GWP Refrigerants, Finned Tube Heat Exchanger, R410A, R32, R454B

4.2 INTRODUCTION

New refrigerants with low GWP are now offered by the chemical industry. Some of these refrigerants will be the solutions of tomorrow and HVAC manufacturers have to be ready to use them. Studies conducted on this transition essentially dealt with the thermal performances of low GWP refrigerants in existing equipment with drop-in tests or soft optimization tests (Amrane, 2015; Wang and Amrane, 2016; Pardo and Mondot, 2018). Some also looked at the compatibility between low GWP refrigerants and materials (Majurin et al., 2014). Studies on heat transfer of low GWP refrigerants in evaporation and in condensation have been published (Azzolin et al., 2016; Del Col et al., 2010; Diani et al., 2014; Diani et al., 2015; Jige et al., 2016; Jige et al., 2017; Longo et al., 2018a; Longo et al., 2018b). Most of these works focused on the heat transfer coefficient in evaporation or in condensation in smooth tubes, microchannels and microfin tubes. The evaluation of the heat transfer coefficient is essential for the heat exchanger design.

The objectives of this work are to assess experimentally the thermal performance, capacity and pressure drop, in condensation and in evaporation, of R32 and R454B, as alternatives to R410A, in a finned tube heat exchanger and to use a reliable software tool for performance comparison of the refrigerants.

The paper is divided in three parts:

- presentation of the main refrigerant properties.
- experimental investigation: method and results.
- numerical investigation: validation of EVAP-COND software (Domanski and Yashar, 2016) and simulations in evaporation and condensation modes.
4.3 REFRIGERANT PROPERTIES

The choice of the alternative refrigerants to R410A is based on the result analysis of the AHRI Low GWP AREP program (Wang and Amrane, 2016) and on the experimental results of drop-in tests previously carried out in an air-to-water heat pump by CETIAT (Pardo and Mondot, 2018). Table 1 presents the main properties of the selected refrigerants. The data source is the software REFPROM V10 (Lemmon et al., 2018).

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Composition</th>
<th>GWP&lt;sub&gt;100&lt;/sub&gt;</th>
<th>Latent heat (kJ/kg) (at 0 °C)</th>
<th>Critical temperature (°C)</th>
<th>Normal boiling point (°C)</th>
<th>Glide (K)</th>
<th>Safety class</th>
</tr>
</thead>
<tbody>
<tr>
<td>R410A</td>
<td>R32/R125 (50/50%w)</td>
<td>2088</td>
<td>221.3</td>
<td>70.2</td>
<td>-51.6</td>
<td>0.1</td>
<td>A1</td>
</tr>
<tr>
<td>R32</td>
<td>R32 (100%w)</td>
<td>675</td>
<td>315.3</td>
<td>78.0</td>
<td>-52.0</td>
<td>0</td>
<td>A2L</td>
</tr>
<tr>
<td>R454B</td>
<td>R32/R1234yf (68.9/31.1%w)</td>
<td>466</td>
<td>260.9</td>
<td>78.1</td>
<td>-50.4</td>
<td>1.3</td>
<td>A2L</td>
</tr>
</tbody>
</table>

Alternative refrigerants have a lower GWP than R410A, -67% for R32 and -78% for R454B. R32 is a pure refrigerant and R454B is a mixture composed of R32 (~69%w) and R1234yf (~31%w). The phase change enthalpies of R454B and R32 are higher than that of R410A by +18% and +42%, respectively. The safety class of alternative refrigerants is A2L, which means they have a low flammability and are non-toxic. The saturation properties (pressure-temperature) are shown in Fig. 1. The three refrigerants have equivalent saturation properties. R454B remains slightly less volatile than R410A and R32.

![Figure 1: Saturation properties: Pressure-Temperature](image)

4.4 EXPERIMENTAL INVESTIGATION

4.4.1 Finned Tube Heat Exchanger Description

The main characteristics of the finned tube heat exchanger are presented in Fig. 2. The finned tube heat exchanger was designed to operate either as an evaporator or as a condenser in a reversible air-to-water heat pump for an optimal heating performance. During tests, two flow patterns were thus used (see Fig. 2d):

- Evaporation mode: counter-current.
- Condensation mode: co-current.
In evaporation mode, the refrigerant flows through a distributor at the upstream side of the heat exchanger, in which a diaphragm is placed. This element generates an additional pressure drop downstream of the expansion device. Then, the distributor is connected to eight capillary tubes which each feeds a circuit. At the outlet, the eight circuits are connected to a collector. In condensation mode, the refrigerant flows in the other way with a by-pass of the diaphragm through a non-return valve; the circuit thus generates less pressure drop than in evaporation mode (see Fig. 4).

### 4.4.2 Experimental Set-Up

A 30-kW thermodynamic loop was designed and built to test the finned tube heat exchanger. It is composed of an inverter driven compressor, an oil separator, a plate heat exchanger, and a manual expansion device. Pt100 contact probes (-20 to 100°C, +/-0.1°C) and pressure transducers (0-40 barg, +/-0.01 barg) are installed at the inlet and outlet of each component of the refrigeration circuit. In addition, a Coriolis mass flow meter (0-720 kg/h, +/-1%) in the liquid line, at the outlet of the condenser, allows measuring the refrigerant flow. This loop allows controlling the test conditions at the inlet of the finned tube heat exchanger on the refrigerant side. On the air side, the finned tube heat exchanger is installed between straight ducts of 2500 mm length, having the same rectangular section as the heat exchanger frontal surface (813 mm × 813 mm) and a 20 mm thick thermal insulation ($\lambda = 0.022$ W/m·K). The inlet air is the ambient air of a climatic room which is controlled in temperature and humidity by an air handling unit. The air volumetric flow rate is adjusted and measured thanks to a multi-nozzle chamber coupled to a fan (0-6000 m$^3$/h, +/-3.5%). Pt100 probes (-20 to 100°C, +/-0.1°C), mirror hygrometers (-20 to 60°C, +/-0.3°C) and Pitot probes (0-100 Pa, +/-1Pa) are placed upstream and downstream to measure, the air-dry bulb and the dew point temperatures and the pressure drop, respectively. The schematic diagram in Fig. 3 shows the installation and the instrumentation in evaporation mode.
8.8 Figure 3: Schematic diagram of the installation and the instrumentation in evaporation mode

In addition, the inlet and outlet of the finned tube heat exchanger were instrumented with pressure transducers and Pt100 contact probes. The fifth circuit from the top of the finned tube heat exchanger was instrumented with 3 pressure transducers (red squares) and 10 Pt100 contact probes (blue hexagons) (see Fig. 4).

4.4.3 Test Conditions
4.4.3.1 Evaporation Mode

The test conditions in evaporation mode are shown in Table 2.
Table 2: Test conditions in evaporation mode

<table>
<thead>
<tr>
<th>Run</th>
<th>Flow</th>
<th>Air</th>
<th>Air dry bulb temperature (wet bulb temperature) (°C)</th>
<th>Air frontal velocity (m.s⁻¹)</th>
<th>Refrigerant mass flow rate (kg.h⁻¹)</th>
<th>Refrigerant liquid temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Counter-current</td>
<td>Dry</td>
<td>20(10)</td>
<td>2.25</td>
<td>230</td>
<td>30</td>
</tr>
<tr>
<td>2</td>
<td>Counter-current</td>
<td>Dry</td>
<td>20(10)</td>
<td>2.25</td>
<td>270</td>
<td>30</td>
</tr>
<tr>
<td>3</td>
<td>Counter-current</td>
<td>Dry</td>
<td>20(10)</td>
<td>2.25</td>
<td>330</td>
<td>30</td>
</tr>
<tr>
<td>4</td>
<td>Counter-current</td>
<td>Wet</td>
<td>16(14)</td>
<td>2.25</td>
<td>270</td>
<td>30</td>
</tr>
<tr>
<td>5</td>
<td>Counter-current</td>
<td>Wet</td>
<td>16(14)</td>
<td>2.25</td>
<td>230</td>
<td>30</td>
</tr>
<tr>
<td>6</td>
<td>Counter-current</td>
<td>Wet</td>
<td>16(14)</td>
<td>2.25</td>
<td>330</td>
<td>30</td>
</tr>
</tbody>
</table>

Two air dry and wet bulb conditions are defined. For each, the refrigerant flow is set at three different values. To obtain an air frontal velocity of 2.25 m·s⁻¹, the inlet volumetric air flow rate is adjusted close to 5350 m³·h⁻¹ (+/-50 m³·h⁻¹). 2.25 m·s⁻¹ is the maximal frontal air velocity reachable by the experimental set-up, due to the fan downstream of the multi-nozzle chamber.

4.4.3.2 Condensation Mode

The test conditions in condensation mode are shown in Table 3.

Table 3: Test conditions in condensation mode

<table>
<thead>
<tr>
<th>Run</th>
<th>Flow</th>
<th>Air</th>
<th>Air dry bulb temperature (wet bulb temperature) (°C)</th>
<th>Air frontal velocity (m·s⁻¹)</th>
<th>Refrigerant mass flow rate (kg·h⁻¹)</th>
<th>Refrigerant inlet condenser temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Co-current</td>
<td>-</td>
<td>30(-)</td>
<td>2.15</td>
<td>230</td>
<td>80</td>
</tr>
<tr>
<td>2</td>
<td>Co-current</td>
<td>-</td>
<td>30(-)</td>
<td>2.15</td>
<td>270</td>
<td>80</td>
</tr>
<tr>
<td>3</td>
<td>Co-current</td>
<td>-</td>
<td>30(-)</td>
<td>2.15</td>
<td>330</td>
<td>80</td>
</tr>
<tr>
<td>4</td>
<td>Co-current</td>
<td>-</td>
<td>30(-)</td>
<td>2.15</td>
<td>360</td>
<td>80</td>
</tr>
</tbody>
</table>

One air condition is tested with four refrigerant flow rates. To obtain an air frontal velocity of 2.15 m/s, the inlet volumetric air flow rate is adjusted close to 5115 m³·h⁻¹ (+/-50 m³·h⁻¹). The frontal air velocity is lower than in evaporation because the air specific volume at the inlet of the heat exchanger is slightly higher: 0.9 against 0.85 (at 16°C) m³_wet_air·kg_dry_air⁻¹. Thus, 2.15 m·s⁻¹ is the maximal reachable frontal air velocity.

4.4.4 Experimental Results

In this section, the evaporating or condensing temperatures, the thermal capacities and the pressure drops measured during the tests are presented.

4.4.4.1 Evaporation Mode

During the evaporation tests, the heat balance between the thermal capacities measured on the air and refrigerant sides is comprised between:

- 0.4% and 2.5% in dry air condition;
- 3.4% and 9.3% in wet air condition.

Three tests with wet air (among a total of nine) have a heat balance between 6.1% and 9.3%. There is only one dew point temperature measurement in the downstream duct against five for the
dry temperature (mean value from 5 Pt100 probes). It can thus be considered that the dew point temperature measurement probably generates these heat balance deviations. Nevertheless, the measurements can be considered as reliable enough in the context of this study.

The evaporating temperatures, the superheating (SH), the cooling capacity and the pressure drop (P4 – Outlet P; see Fig. 4) are presented in Table 4 for the three refrigerants.

### Table 4: Main results measured on the refrigerant side in evaporation mode

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Run</th>
<th>Evaporating temperature (°C)</th>
<th>SH (K)</th>
<th>Cooling capacity (kW)</th>
<th>Pressure drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Bubble point</td>
<td>Dew point</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R410A</td>
<td>1</td>
<td>2.2</td>
<td>1.4</td>
<td>16.6</td>
<td>12.2</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1.1</td>
<td>-0.3</td>
<td>17.6</td>
<td>14.3</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>-0.7</td>
<td>-2.8</td>
<td>18.5</td>
<td>17.5</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>3.4</td>
<td>2.6</td>
<td>11.7</td>
<td>11.9</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>0.4</td>
<td>-1.0</td>
<td>15.1</td>
<td>14.1</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>-0.5</td>
<td>-2.7</td>
<td>16.0</td>
<td>17.3</td>
</tr>
<tr>
<td>R454B</td>
<td>1</td>
<td>1.8</td>
<td>0.4</td>
<td>16.7</td>
<td>14.6</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>0.5</td>
<td>-1.3</td>
<td>17.3</td>
<td>17.0</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>-0.3</td>
<td>-3.3</td>
<td>15.6</td>
<td>20.8</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>0.4</td>
<td>-1.1</td>
<td>15.2</td>
<td>14.3</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>0.2</td>
<td>-1.8</td>
<td>15.2</td>
<td>17.1</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>0.3</td>
<td>-2.7</td>
<td>14.6</td>
<td>20.6</td>
</tr>
<tr>
<td>R32</td>
<td>1</td>
<td>2.2</td>
<td>1.0</td>
<td>12.7</td>
<td>17.4</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>-3.7</td>
<td>-4.7</td>
<td>20.9</td>
<td>21.1</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>-3.5</td>
<td>-7.1</td>
<td>12.5</td>
<td>25.1</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>2.2</td>
<td>1.0</td>
<td>11.0</td>
<td>17.5</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>0.6</td>
<td>-1.4</td>
<td>12.4</td>
<td>20.6</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>-1.7</td>
<td>-5.1</td>
<td>14.0</td>
<td>25.2</td>
</tr>
</tbody>
</table>

It should be noted that the evaporating temperatures of the three refrigerants are not the same. The differences can reach up to 5 K (see runs 2, 3 and 4). This is due to the difficulty in adjusting this parameter during the test. Some of the differences in cooling capacity are due to better heat transfer coefficient values and some due to large temperature differences between air and refrigerant. Similarly on pressure drop, most of the increase in pressure drop for R32 and R454B seem to be due to lower evaporation temperature (and pressure) so lower density vapour and higher vapour velocities for the same mass flow. This means that we cannot directly conclude on the benefit of using one refrigerant in comparison to another.

#### 4.4.4.2 Condensation Mode

During the condensation tests, the heat balance between the thermal capacities measured on the air and refrigerant sides is comprised between 0.5% and 2.1%. The measurements are reliable.

The condensing temperatures, the subcooling (SC), the heating capacity and the pressure drop (Outlet P – P4; see Fig. 4) are presented in Table 5 for the three refrigerants.
### Table 5: Main results measured on the refrigerant side in condensation mode

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Run</th>
<th>Condensing temperature (°C)</th>
<th>SC (K)</th>
<th>Heating capacity (kW)</th>
<th>Pressure drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Dew point</td>
<td>Bubble point</td>
<td></td>
<td></td>
</tr>
<tr>
<td>R410A</td>
<td>1</td>
<td>46.6</td>
<td>46.3</td>
<td>6.2</td>
<td>13.31</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>48.3</td>
<td>48.1</td>
<td>6.6</td>
<td>15.29</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>50.2</td>
<td>49.9</td>
<td>4.8</td>
<td>17.8</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>51.1</td>
<td>50.7</td>
<td>4.0</td>
<td>19.25</td>
</tr>
<tr>
<td>R454B</td>
<td>1</td>
<td>50.3</td>
<td>48.5</td>
<td>7.3</td>
<td>15.29</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>52.1</td>
<td>50.3</td>
<td>7.7</td>
<td>17.34</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>53.8</td>
<td>52.0</td>
<td>6.2</td>
<td>20.39</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>55.0</td>
<td>53.2</td>
<td>5.9</td>
<td>21.88</td>
</tr>
<tr>
<td>R32</td>
<td>1</td>
<td>49.4</td>
<td>49.3</td>
<td>4.4</td>
<td>17.78</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>51.5</td>
<td>51.3</td>
<td>4.3</td>
<td>20.10</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>54.0</td>
<td>53.8</td>
<td>3.3</td>
<td>23.44</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>56.3</td>
<td>56.1</td>
<td>5.7</td>
<td>25.45</td>
</tr>
</tbody>
</table>

As previously, the condensing temperatures of the three refrigerants do not have the same value. The differences can reach up to 5 K. We cannot directly conclude on the benefit of using one refrigerant in comparison to another.

### 4.5 NUMERICAL INVESTIGATION

#### 4.5.1 Software Validation

The objective of this part is to compare the experimental results with the results of the EVAP-COND software. The geometrical parameters of the finned tube heat exchanger and the inlet operating conditions have been used as input data in the software. The 30 experimental tests were simulated and compared. The calculated versus experimental capacities of R410A, R454B and R32, in evaporation and condensation modes, are plotted in Fig. 5.

![Graphs showing calculated versus experimental capacities](image)

**Figure 5:** Calculated capacity versus experimental capacity of R410A, R454B and R32, for both modes: a) Evaporation mode; b) Condensation mode
According to Fig. 5, the comparison shows that the EVAP-COND software is a reliable tool to simulate the thermal capacity of finned tube heat exchangers. The maximal deviation observed is less than 2.5%. As a reminder, the fifth circuit from the top of the finned tube heat exchanger was instrumented with 10 Pt100 contact probes and 3 pressure transducers. The comparisons between the simulated and the experimental temperature and pressure evolutions along the 5th circuit of the run 1, in evaporation and in condensation, are shown in Fig. 6 and Fig. 7, respectively. The horizontal axis refers to the inlet and outlet of each of the 12 tubes of the circuit.

**Figure 6: Calculated and experimental temperatures and pressures along the 5th circuit in evaporation:** a) Refrigerant temperature; b) Refrigerant pressure

**Figure 7: Calculated and experimental temperatures and pressures along the 5th circuit in condensation:** a) Refrigerant temperature; b) Refrigerant pressure

The temperature and pressure evolutions of the three refrigerants in the simulated circuit are close to those observed during experimental tests. The same conclusions are drawn from all other runs of the tests (not presented in detail in this paper). We can conclude that EVAP-COND software is a reliable tool to design the finned tube heat exchanger; however, its robustness has not been fully validated in this study. Since most of the HFC alternatives have a high temperature glide, a comparison between experimental results obtained with a higher temperature glide refrigerant (>5 K) and the EVAP-COND simulations could be of interest.

### 4.5.2 Numerical Analysis

EVAP-COND software is then used to simulate the performance in evaporation mode and in condensation mode of the same heat exchanger design with the different refrigerants. For that purpose, the refrigerant mass flow rate is not fixed as it was for the experimental approach.

In evaporation mode, the SC1 condition of the NF EN 328 standard (see Table 6) is used with an air frontal velocity of 2.25 m·s⁻¹. The main results are presented in Table 7.
Table 6: SC1 condition of the NF EN 328 standard

<table>
<thead>
<tr>
<th>Air dry bulb temperature (°C)</th>
<th>Dew point temperature (°C)</th>
<th>Evaporating temperature (Dew point) (°C)</th>
<th>Superheating (K)</th>
<th>Refrigerant liquid temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>&lt; -2</td>
<td>0</td>
<td>6.5</td>
<td>30</td>
</tr>
</tbody>
</table>

Table 7: Main simulation results: Evaporation mode – SC1

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Evaporating temperature (°C)</th>
<th>Refrigerant mass flow rate (kg.h(^{-1}))</th>
<th>Cooling capacity (kW)</th>
<th>Pressure drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R410A</td>
<td>0.2</td>
<td>195</td>
<td>9.7</td>
<td>8.7</td>
</tr>
<tr>
<td>R32</td>
<td>0.1</td>
<td>125 (-36%)</td>
<td>9.3 (-5%)</td>
<td>5.2 (-40%)</td>
</tr>
<tr>
<td>R454B</td>
<td>-0.7</td>
<td>170 (-13%)</td>
<td>10.3 (+5%)</td>
<td>8.7 (0%)</td>
</tr>
</tbody>
</table>

In comparison to R410A, R454B shows a higher cooling capacity (+5%) and an equivalent pressure drop. In comparison to R410A, R32 shows a lower cooling capacity (-5%), but also a lower mass flow rate (-36%) and a lower pressure drop (-40%).

In condensation mode, the SC1 condition of the NF EN 327 standard (see Table 8) is used with an air frontal velocity of 2.15 m·s\(^{-1}\). The main results are presented in Table 9.

Table 8: SC1 condition of the NF EN 327 standard

<table>
<thead>
<tr>
<th>Air dry bulb temperature (°C)</th>
<th>Condensing temperature (Bubble point) (°C)</th>
<th>Subcooling (K)</th>
<th>Superheating (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>40</td>
<td>&lt; 3</td>
<td>40</td>
</tr>
</tbody>
</table>

Table 9: Main simulation results: Condensation mode – SC1

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Condensing temperature (°C)</th>
<th>Refrigerant mass flow rate (kg.h(^{-1}))</th>
<th>Heating capacity (kW)</th>
<th>Pressure drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R410A</td>
<td>39.9</td>
<td>212</td>
<td>12.9</td>
<td>2.4</td>
</tr>
<tr>
<td>R32</td>
<td>40.0</td>
<td>151 (-29%)</td>
<td>12.7 (-1%)</td>
<td>1.8 (-25%)</td>
</tr>
<tr>
<td>R454B</td>
<td>39.9</td>
<td>186 (-12%)</td>
<td>13.3 (+3%)</td>
<td>2.4 (0%)</td>
</tr>
</tbody>
</table>

In comparison to R410A, R454B shows a slightly higher heating capacity (+3%) and an equivalent pressure drop. In comparison to R410A, R32 shows an equivalent heating capacity (-1%), a lower mass flow rate (-29%) and a lower pressure drop (-25%).

For the replacement of R410A, the same finned tube heat exchanger could be designed with R454B, and a design optimization will be necessary with R32.

4.6 CONCLUSIONS

In the first part of this study, the thermal performances of R410A, R32 and R454B were measured in a finned tube heat exchanger, in condensation and in evaporation. 30 tests were carried out. In the second part of the study, the experimental results were used to validate the simulation results from the EVAP-COND software. The differences between experimental and calculated capacities are less than 2.5% for all tests. The temperature and pressure evolutions of the three refrigerants in the simulated circuit are also close to those observed during experimental tests. These
comparisons for a pure, a quasi-azeotrope and small temperature glide (<2 K) refrigerants, allow considering EVAP-COND software as a reliable tool for the simulation of finned tube heat exchangers. The simulations of a same heat exchanger design with the 3 refrigerants show that the same design of finned tube heat exchanger can be used for R410A and R454B, but a design optimization will be necessary with R32.

4.7 REFERENCES


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5 Country Report: Germany

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5.1 Task 1: Review of state-of-the-art technologies

5.1.1 INTRODUCTION

This is the continuation of the German contribution of the Annex 54 annual report 2020\(^1\) included a summary on the most recent large-scale heat pump monitoring project “WP Smart im Bestand”\(^2\) (FKZ 03ET1272A) at Fraunhofer ISE, a review of ongoing heat pump projects in Germany based on an analysis of the enArgus database (www.enargus.de) and a survey on heat pumps and their refrigerants as part of the market incentive program (MAP) coordinated at the Federal Office for Economic Affairs and Export Control (BAFA).

Since the monitoring project has ended and the enArgus database can easily be accessed by anyone to get an overview on most recent public, innovative heat pump projects only the analysis of state-of-the-art heat pumps on the German and EU market is analyzed, updated and extended in this report.

5.1.2 STATE-OF-THE-ART ON CURRENTLY MARKET-AVAILABLE AIR-TO-WATER HEAT PUMPS

This year the top-down approach to analyze market-available heat pumps to identify current components and system designs using already low-GWP refrigerants was put into the focus, extended, compared to last year’s activities, and comprises now

1. An update of BAFA heat pump models correlated with their used refrigerants. This time the dataset doesn’t focus only on the ten largest manufacturers. This year the coverage of identified refrigerant type was 83% out of about 2650 listed air-to-water heat pumps. The used version for the BAFA list was from 14\(^{th}\) December 2020 as the last version offered including COP data,
2. The analysis of sold heat pumps (only models included in the incentive scheme) within 2020 in terms of the used refrigerants with a coverage of >99% identified refrigerants of all sold heat pumps, and
3. The analysis of all medium temperature air-to-water heat pumps listed as part of the HP keymark database\(^3\) which provides similar if not even better data on market available systems due to the large number of dead bodies included in the BAFA list of subsidized heat pump models.\(^4\)

5.1.3 ANALYSIS OF THE BAFA DATABASE

The combination of different data sources is continued to correlate the refrigerant types as well as the COPs within the BAFA dataset. The BAFA dataset used this time is the last version available including COP figures and is from 14th December 2020. Only air-to-water heat pumps were


\(^3\) See here: [https://www.heatpumpkeymark.com/](https://www.heatpumpkeymark.com/) (last visit: 15\(^{th}\) December 2021)

\(^4\) For comparison have a look to the most recent BAFA “list” here: [https://www.bafa.de/SharedDocs/Downloads/DE/Energie/ee_waermepumpen_anlagenliste_bis_2020.pdf?__blob=publicationFile&v=1](https://www.bafa.de/SharedDocs/Downloads/DE/Energie/ee_waermepumpen_anlagenliste_bis_2020.pdf?__blob=publicationFile&v=1) (last visit 15\(^{th}\) December 2021)
Annex 54, Heat pump systems with low-GWP refrigerants

analyzed. BAFA data included up to four different COPs for air-to-water heat pumps. It is assumed that the largest share of the listed COP figures is based on testing heat pumps according to DIN EN 14511 and DIN EN 14825. Contrary to the analysis in the Annex 54 annual report in 2020 this analysis was not limited to largest manufacturers but aimed for maximizing the amount of heat pump models for which the refrigerant type could be identified, see Figure 1 for the graphical representation of this analysis. There were 13 different refrigerants identified. A relatively large share of 744 units is still not identified in terms of the used refrigerant types since online data on the heat pump specifications were investigated. The error bars represent a scatter range of the sum of available heat pumps for a specific refrigerant. Of course, the more heat pump models are offered and the longer a refrigerant is used in market-available systems the broader these error bars become. This is due to a change in the technical readiness of such systems (old systems are also included in these figures) and quality differences between manufacturers and their models.

Peaks and Minima represent individual models defining these limits. These limits could be surprisingly small or large. If possible, the dataset was analyzed to avoid such broadening effects of the scatter ranges due to typos in the BAFA list entries.

Figure 54: Refrigerant-specific graphical representation of the COP values included in the BAFA tables in 12/2020 for air-to-water heat pumps.

To simplify the comparison of the future-proof low-GWP refrigerants included in the BAFA database Figure 2 and Figure 3 show a comparison limited to the low-GWP refrigerants.
Figure 55: Refrigerant-specific graphical representation of the COP values included in the BAFA list in December 2020 for air-to-water heat pumps, filtered to focus only on low-GWP refrigerants. The GWP-threshold was set to the GWP of R32.

Figure 56: Share of low-GWP refrigerant (neglecting the large amount of not identified refrigerants, see NN in Figure 1). The GWP-threshold was set to the GWP of R32.
5.1.4 ANALYSIS OF SOLD UNITS OF THE GERMAN MARKET INCENTIVE PROGRAM

The market incentive program (MAP) and the BAFA lists are published for more than ten years now. Due to the unreviewed character of this list, it contains a share of dead bodies for which heat pump specifications can still be found, but which are not offered on the market anymore. Furthermore, sales figures are not equally distributed over all listed heat pump models. For this reason, the share of refrigerants sold within the market incentive program differs a lot.

More than 25,000 heat pumps were correlated to their refrigerants and a coverage of about 99% was reached for the refrigerant identification for the annual sales figures of the MAP for 2020. A Comparison of the shares of refrigerant types is shown in Figure 4.

![Comparison of the shares of refrigerant types. The graph comprises all refrigerant types for air-to-water heat pump units sold in 2020 as part of the market incentive program.](image)

5.1.5 ANALYSIS OF THE HP KEYMARK DATABASE

To complete the picture on relevant figures about current market-available air-to-water heat pump models the HP Keymark database was analyzed. HP Keymark is a certification mark. The official Keymark website defines the HP Keymark as cited below:

"The Heat Pump KEYMARK is a voluntary, independent European certification mark (ISO type 5 certification) for all heat pumps, combination heat pumps and hot water heaters (as covered by Ecodesign, EU Regulation 813/2013 and 814/2013). It is based on independent, third-party testing and demonstrates compliance with product requirements as set in the Heat Pump KEYMARK scheme rules and with efficiency requirements as set by Ecodesign Lot 1 and Lot 2. The Heat Pump KEYMARK scheme is owned by the European Committee for standardization (CEN). The certificates are granted by independent Certification Bodies to products fulfilling all requirements of the scheme."
Once again, the analysis was limited to air-to-water heat pumps. The analysis was further limited to the medium temperature applications which includes the data for EN 14511-2 testing at the ambient temperature 7°C as well as the two supply temperatures of 35°C (low temperature – LT) and 55°C (medium temperature – MT), respectively, see Figure 5. The bar plots show the COP at both operating conditions.

The HP Keymark database is contrary to the BAFA list not very old and thus it promises to comprise fewer dead bodies as the BAFA database on heat pump models. The HP Keymark website was last visited on 28th June 2021 to generate the database and the figures. Similar – as shown for the BAFA data – scatter ranges were included in the bar plots. As before, the scatter range becomes larger the larger the number of included models is and the longer a refrigerant is established as being used in heat pumps. Peaks and Minima represent individual models defining these limits. These could be surprisingly small or large. If possible, the dataset was analyzed to avoid such broadening effects of the scatter ranges due to typos in the Keymark database entries.

![Figure 58: Refrigerant-specific graphical representation of the COP values for air-to-water heat pumps included in the HP Keymark database.](image)

**5.1.6 CONCLUSIONS**

Independent from the investigated databases from all available heat pump types in these databases only air-to-water heat pump types were analyzed due to their large market relevance.

The analysis of market-available heat pumps allows very diverse but also very relevant analysis to understand roll-out processes for newly developed heat pumps based on low-GWP refrigerants but also to understand what efficiencies can be expected for these refrigerants.

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It can be concluded that a lot of heat pump models that use future-proof low-GWP refrigerants are already available on the market. In terms of efficiency, heat pumps with low-GWP refrigerants are similar and differences are in at the maximum in the first decimal place.

**5.1.7 OUTLOOK**

To get optimized energy and emission saving strategies it makes sense to investigate the details of these differences. Optimal emission savings could be reached by choosing specific refrigerants. Care must be taken in such a case to include especially most efficient and most modern heat pumps. Fraunhofer ISE will continue investigating the datasets in 2022 to support Task 3 activities for identifying LCCP reduction potentials.

In addition, a more in-depth analysis of all heat pump models, sorted by the refrigerants used, will be carried out to understand whether key figures such as a "technical readiness" or potentials for further improvements can be derived.
5.2 Task 2: Case studies and design guidelines for optimization of components and systems

5.2.1 Summary

The work on heat pump research in Germany in 2021 is diverse and heterogenous in terms of involved institutions (universities, public research bodies, and companies) as well as topics. Within the last five years the activities for heat pump research have changed. The activities cover more fundamental research up to the application of deployable heat pump demonstrators for white goods (e.g. dishwashers), mobile systems for electric-driven buses or large capacity heat pumps systems for multi-family houses.

5.2.2 INTRODUCTION

Main activities in Germany will be introduced as separate cases and can be associated with the following research topics (if there is any working fluid as part of the research focus and not part of the title already it will be added in brackets). There is no relationship in sequence between the list of authors on the first page of the German Task 2 report and the activities listed and described in this section and in the following section. Authorship and origin of the work can be traced back by following the abbreviations mentioned for each institution.

- System design principles
  - Towards an integrated design of heat pump systems: Application of process intensification using two-stage optimization (RWTH)
- Piping design
  - Investigation of Pressure Drop and Mass Flow Distribution in Refrigerant Pipes (TUHH, R1234yf)
- Precise charge optimization
  - Automated charge optimization toolchain for LC150 project (ISE, R290)
- Working fluid research
  - Theoretical Assessment of Binary Mixtures as Working Fluids in Heat Pump Cycles (RWTH, binary mixtures of R290, R1270, R600, R600a, R170 and R744 are in the focus)
  - Predictive Screening of Working Fluid Mixtures to Increase the Energy Efficiency of Refrigeration Systems and Heat Pumps (TUD, binary mixtures with focus on R290, RE170, other refrigerants are involved)
- System research
  - Networks with lowered temperature as providers of control power (THI, R513A)
- Mobile systems
  - Heating and air conditioning of battery electric city buses by reversible R744 heat pump modules (TUB)
- Stationary systems
  - Hybrid Heat Pump+ research (THI, R454B)
  - LowEx-concepts for supplying heat to existing multifamily buildings – New refrigerants for high-temperature heat pumps (ISE, R454C/R455A)
- Self-driven processes
- Component-testing research
  - Optimal Designed Experiments for Reliable Model Calibration of a fixed-speed Scroll Compressor with R410A and R32 (RWTH)

- Standard research
  - Domestic heat pumps with natural refrigerants – development of requirements for climate-friendly and energy-efficient appliances for the Blue Angel (HEAT)

- Safety research
  - Main findings of the OpenFOAM CFD simulation study on a monoblock propane heat pump in ten different installation location scenarios (ISE, R290)

- Absorption research with low-GWP refrigerants
  - Efficiency increase of a NH₃/H₂O absorption chiller – Experimental investigation of a plant concept with plate desorber (IGTE)

5.2.3 CASE STUDIES

5.2.3.1 Research on System Design Principles: Towards an integrated design of heat pump systems: Application of process intensification using two-stage optimization (RWTH)

For a sustainable building stock, air-source heat pump systems are identified as a key technology in this work. The authors conclude that there is no integrated strategy to optimally design heat pump systems. The authors use process intensification to consider the heat pump system design and operation simultaneously.

Based on a systematic application of the regime of process intensification, the complexity of the optimization problem is reduced applying rigorous analysis and modeling prior to optimization. Main conclusions found by the authors are that compared to (typical) HPS design guidelines, the optimal solutions improve costs by up to 36.4% and CO2-equivalent emissions by up to 51.7%. Furthermore, R290 and vapor injection are selected as optimal fluid and flowsheet due to broader envelope and higher efficiency compared to all other options (case a). This theoretical framework was not yet validated/tested for its practicality.

5.2.3.2 Piping design: Investigation of Pressure Drop & Mass Flow Distribution in Refrigerant Pipes (TUHH)

With the commencement of EU Directive 2006/40/EC, the use of refrigerants with a global warming potential greater than 150 has no longer been permitted in new passenger cars since January 1, 2017. As a result, the widespread refrigerant R134a has been substituted with the low-GWP alternative R1234yf in automotive air conditioning systems. Since the two refrigerants have very similar thermophysical properties, there is no need to redesign the refrigeration cycle. However, the question arises how the substitution affects the individual system components in detail. In this work, therefore, a comprehensive investigation of the single-phase and two-phase pressure drop of refrigerant lines is carried out. In addition to serially flowed lines, another focus of this work is the investigation of the hydraulic flow behavior of evaporator tubes connected in parallel. For the experimental determination of the pressure drops and for the validation of the models used, a test facility with the refrigerant R1234yf is set up, which allows the measurement of different line types.
under the respective usual operating conditions. For the numerical investigation of the geometries, both one-dimensional system simulations and three-dimensional field simulations are carried out.

The investigation is based on simple geometries such as a straight pipe, a 90° bend, an S-bend and a pipe helix. It is found that the pressure drop of these components in single-phase flows can be calculated very accurately using known correlations from the literature. If the Mach number is greater than 0.18, there is a significant influence of gas compressibility. In the case of two-phase flows, a large number of literature correlations exist with widely varying accuracy. More complex refrigerant lines from a production car are also investigated. These consist of a combination of straight pipes, bends, cross-sectional variations, and flexible hoses. CFD simulations highlight the strong influence of the tube-to-hose transition pieces on the total pressure drop of the line. Their share can be as high as 85% for lines with two hoses. If the transition pieces are considered separately in the modeling, one-dimensional models are suitable for predicting the pressure drop of the refrigerant line with an error of less than 10%. A system simulation of the entire refrigeration cycle further shows that the cumulative pressure drop of the lines during operation can be as high as 1 bar. This leads to an overestimation of the efficiency by up to 4.9% if the lines are neglected. The investigation of the hydraulic interaction for two evaporator tubes connected in parallel shows that the heat flow ratio has the strongest influence on the mass flow distribution. In this case, an increase in the heat flow ratio by 10% points approximately causes a reduction in the mass flow ratio by 10% points as well. To avoid a cooling failure, the control of the total mass flow, the reduction of the inlet subcooling or the installation of upstream resistances are suitable. It is shown that the mass flow ratio alone is not sufficient to evaluate the cooling, which is why gas superheat must also be considered in the case of complete evaporation.

5.2.3.3 Precise charge optimization: Automated charge optimization toolchain for LC150 project (ISE)

At Fraunhofer ISE the LC150 project is ongoing in which industrial partners – representing 60% of the German heat pump market – are involved to develop the database for propane-based heat pump appliances for indoor applications. The core activity is a fully automated cross-evaluation, a team from the institute is testing various components of heat pumps on the large scale, in which dozens of component combinations under different operating parameters are investigated. The year 2021 was fully focused on the deployment on the automated testbenches as well as optimization of procedural tasks to start the experimental testing campaigns of up to 80 different refrigerant cycles.

The main objectives are to further reduce the volume of required refrigerant, to identify methodological correlations and to obtain data for the simulation of heat pump design. The measurement campaign collects an abundance of parameter variations 24/7 over one year, thus generating a unique database. The tests run in parallel on three identical test stands 24 hours a day for one year, with between 30 and 150 operating points being measured per prototype and the measured values recorded by 26 sensors. The automated testing technology was developed in close cooperation with the company EP Ehrler Prüftechnik Engineering GmbH.
The automated charging equipment, shown in Figure 6, allowed in first tests the following reproducible charging and discharging routines, see Table 1.

Table 4: Reached charge process specifications

<table>
<thead>
<tr>
<th>Action</th>
<th>dP-Sensor (mass measurement due to liquid column)</th>
<th>Balance (gravimetric mass measurement)</th>
<th>Coriolis (mass measurement by mass flow)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charging process for 100g refrigerant</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Start at $T_{\text{sat}}$ -7°C</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>V7 closed</td>
<td>101,3</td>
<td>100,7</td>
<td>101,6</td>
</tr>
<tr>
<td>Equilibration time 30 min</td>
<td>100,3</td>
<td>100,6</td>
<td>101,6</td>
</tr>
<tr>
<td>Equilibration time 90 min</td>
<td>100,4</td>
<td>101</td>
<td>101,6</td>
</tr>
</tbody>
</table>

The needed accuracy is reached after about 30-60 min equilibration time, depending on ambient conditions as well as the handled charge amounts. The operation of this testbench also works at very small refrigerant charge changes. Due to the fully automated testbench and the physical consequences to work with cold traps and not-so-small refrigerant hoses the testing time is relatively long time but acceptable in the 24/7 operation mode. This automated charge modification will be realized at three cycles consecutively but in the same testing campaigns. First results are expected within 2022.
5.2.3.4 Working fluid research: Theoretical Assessment of Binary Mixtures as Working Fluids in Heat Pump Cycles (RWTH)

In this work the authors replace conventional refrigerants by zeotropic mixtures. Based on the assumption that zeotropic mixtures have the potential of reducing exergy losses within both heat exchangers a mixture's improvement potential is investigated. The mixture of interest is propane/isobutane. For the cycle evaluation, two types of compressor modeling approaches are used to study the effects of the mixture composition on the compressor. The first model uses a constant isentropic efficiency while a semi-physical compressor model is used in the second approach that evaluates the isentropic efficiency for each fluid individually. Both approaches lead to strong deviations. The optimal mixture composition for the first model regarding the cycle efficiency is located at 20 mol-% propane and 80 mol-% isobutane, while the second model results in an optimal composition of 90 mol-% propane and 10 mol-% isobutane. For both studies, efficiency improvements of 4 % to 8 % are possible compared to the existing mixture R436A. The high deviations result from the shifts in isentropic efficiency. Thereby, propane shows the highest and isobutane shows the lowest isentropic efficiencies using the semi-physical compressor model. The authors conclude that the assumption of a constant isentropic efficiency is inaccurate when variations in mixture compositions are studied. Thus, precise models not only for the heat exchangers but also for the compressor are necessary when working fluids are compared.

5.2.3.5 Working fluid research: Predictive Screening of Working Fluid Mixtures to Increase the Energy Efficiency of Refrigeration Systems and Heat Pumps (TUD)

Mickoleit et al. (2020) and Stöckel et al. (2021) have presented the application of a new predictive mixture model to identify optimal working fluid mixtures for different cycle architectures, see Jäger et al. (2018a, 2018b). The screening of the working fluids is carried out for specific applications in the corresponding refrigerant circuits with the goals of a GWP below 150 and high efficiency. Figure 7 shows preliminary results for a binary mixture screening applied to a heat pump in a dishwasher. The corresponding process schematic is shown in Figure 8. The current investigation focuses on validating the thermophysical properties predicted in the mixture model. In a next step, the theoretically most suitable working fluid is tested in an adapted heat pump dishwasher to validate the system-level performance enhancement predictions in addition to the validation of the refrigerant property predictions made by the model.

A commercial dishwasher was equipped with a basic refrigeration cycle to replace the electrical heating element with a refrigerant condenser. A single speed compressor, an air-cooled evaporator and an electronic expansion device complete the cycle. The screening for a low-temperature glide mixture resulted in a composition of R290 and RE170 of a molar ratio of 87.1% to 12.9%. This blend has a GWP of 2.99. Especially at high condensing temperatures, the mixture with RE170 theoretically reaches higher COP than pure propane.

Within the scope of the project, the test-setup with additional measuring devices serves the validation of the predicted mixture behaviour and aids in understanding the implementation of a heat pump in yet another household appliance.
Figure 7: Exemplary results (Coefficient of Performance over the volumetric heating capacity) of a binary mixture screening for a dishwasher using a heat pump for supplying water at three different temperature levels.

Figure 8: P&I Diagram of the adapted heat pump dishwasher

5.2.3.6 System research: Networks with lowered temperature as providers of control power (THI)

As part of the research project EnEff: Wärme – Networks with lowered temperature as providers of control power (NATAR, FKZ 03ET1425), the use of an alternative refrigerant for decentralized heat pumps was investigated as part of a work package. As part of the project, a heating network in Dollnstein (Bavaria) with seasonal temperature lowering was investigated. The network flow temperature of approx. 45 °C prevailing during the summer months is not sufficient to cover the
domestic hot water demand or to meet the associated hygienic requirements. For this reason, in addition to a classic heat exchanger for winter operation, additional decentralized heat pumps with a power range (depending on the consumer) of between 1.9 kW and 3.1 kW are integrated into the consumers' house transfer stations, which raise the heat provided by the grid to the useful temperature level. The decentralized heat pumps are the Oskar-Max product from the manufacturer ratiotherm Heizung + Solartechnik GmbH & Co KG.

Currently, the heat pumps are operated with the refrigerant R134a. To reduce the global warming potential for refrigerant use, tests were carried out by the manufacturer ratiotherm Heizung + Solartechnik GmbH & Co. KG after comparing various refrigerant alternatives with the refrigerant R513A.

The following reasons speak for this choice:

- Reduction of GWP by 50% compared to R134a,
- Identical safety class as R134a,
- No/hardly any constructive changes necessary,
- Expectation of comparable efficiency.

For the testing of the refrigerant R513A, the heat pump was not changed constructively, the compressor capacity thus continues to range from 25-100 %. For a representative measurement, the source temperature was varied in the range from 10-30 °C and the sink temperature from 35-70 °C. The tests showed that the use of the alternative refrigerant R513A in the heat pump reduces the heating capacity by an average of 1%. In addition, the COP is reduced by 2.2% on average. Based on these findings, it can be concluded that the performance of the heat pump is comparable when using an alternative refrigerant, but the GWP can be significantly reduced compared to R134a.

When using the refrigerant R513A, the expansion valve of the heat pump offers optimization potential. Since the volumetric energy density of the refrigerant is higher, the valve position changes by an average of 15.6 %. Due to the non-linear valve characteristics, a widely closed valve has a detrimental effect on the control behavior. By adjusting the valve, the efficiency of heat pump operation can be improved. Due to the promising results achieved by using the refrigerant R513A and thus reducing the greenhouse potential, further development of the heat pump by ratiotherm is planned in the medium term.

5.2.3.7 Mobile systems: Heating and air conditioning of battery electric city buses by reversible R744 heat pump modules (TUB)

City buses exhibit relatively high energy demands for heating and air conditioning the passenger cabin compared to other types of vehicles such as cars or trucks. This is due to the large external area and the frequent door openings. Heating a current battery electric city bus on cold days with ambient temperatures below 0 °C will reduce the driving range by approximately half. For this application a heat pump can increase the driving range significantly. The currently used refrigerants R134a and R1234yf have limited heating efficiency and are also topics of controversy due to their negative environmental impact. With the natural refrigerant R744 (carbon dioxide, CO2) these issues are resolved, see Peteranderl et al. (2018) and Peteranderl (2020).

A modular heat pump and air conditioning system makes it possible to use passenger car components in a city bus, see Peteranderl et al. (2018). The number of modules can be adapted to customer requirements and the climatic conditions determined for the locally limited areas where
city buses are operational. Economies of scale will be achieved by using components from the passenger car industry, see Peteranderl et al. (2018), Peteranderl (2020) and VW (2020).

In the present work of Peteranderl (2020) independent reversible R744 heat pump modules for city buses based on passenger car components are scientifically investigated with regard to the ideal number of modules and energy efficiency. So far, no scientific studies on modular concepts in vehicle HVAC systems are known. In addition, the heating and cooling demands of the cabin in realistic use cases are analyzed and their minimum is deduced for battery electric city buses. This is used to evaluate the R744 modules.

With the example of a prototype R744 module the functionality is verified, and measurements are taken to validate a simulation model. The simulation model of the R744 prototype module shows good correlation to the measurement data. Further optimisation steps are implemented in the model and, based on this, the maximum performance is determined and an energy-efficient control strategy is derived.

Figure 9: Influence of heating technology on the range of a battery-electric city bus

The simulation studies have shown that in heating mode the active number of modules is reduced at low energy demands to ensure energy-efficient operation, while in cooling mode all modules have to be activated. In the reference cities of Munich, Stockholm, Ankara and Dubai, three modules for heating and cooling a characteristic city bus cabin are required for a typical year of operation. Two modules are needed in Granada and four modules in Moscow. The driving range of a battery electric city bus can be improved in heating mode at -15 °C ambient temperature by up to 52% with three R744 modules compared to an R134a heat pump with electric heater. The comparison of annual energy consumption in the abovementioned moderately cold to cold reference cities shows that 25% of the electric current for heating and air conditioning as well as corresponding ratios of greenhouse gas emissions.
Table 5: Comparison of a reference R134a cycle with the investigated R744 cycle performance for different climates

<table>
<thead>
<tr>
<th>Reference city, climate</th>
<th>Annual savings by a modular R744 mobile air conditioner / heat pump compared to a R134 system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moskau, very cold</td>
<td>-30 %</td>
</tr>
<tr>
<td>Stockholm, cold</td>
<td>-21 %</td>
</tr>
<tr>
<td>Munich, gemäßigt</td>
<td>-19 %</td>
</tr>
<tr>
<td>Ankara, continental</td>
<td>-15 %</td>
</tr>
<tr>
<td>Granada, warm</td>
<td>- 4 %</td>
</tr>
<tr>
<td>Dubai, very warm</td>
<td>+ 27 %</td>
</tr>
</tbody>
</table>

The presented R744 heat pump modules based on passenger car components represent a potential key technology for increasing the efficiency and market share of battery electric city buses for a large part of the world’s urban climate regions.

5.2.3.8 Stationary systems: Hybrid Heat Pump+ research (THI)

In the current Hybrid Heat Pump+ research project (FKZ: 16KN056420), a new refrigeration circuit connection is being developed that enables flexible and energy-optimized utilization of two different heat sources. By combining, for example, an air-source heat pump with a ground-source heat pump, the individual advantages of the respective heat sources can be utilized in a common refrigeration circuit. This is charged with refrigerant R454B, while components of the heat pump are already largely designed for R32 (the main component of R454B). The COPs achieved in the air source operating mode are like those of an air source heat pump charged with the common refrigerant R410A. In addition to the A2L classification according to ISO 817 (low toxicity and flammability) of the refrigerant R454B, this finding indicates a technically uncomplicated retrofitting with a – compared to conventional refrigerants – significantly low GWP of 467.

Based on the findings it can be concluded that the use of alternative refrigerants promises comparable performance with a simultaneous significant reduction in greenhouse potential.

5.2.3.9 Stationary systems: LowEx-concepts for supplying heat to existing multifamily buildings – New refrigerants for high-temperature heat pumps (ISE)

The main task of the New Refrigerants for High Temperature Heat Pumps (NK4HTWP) project was the development of a demonstrator of a heat pump heating system for multi-family houses in existing buildings based on new refrigerants with a sustainable GWP of less than 150. The targeted efficiencies for the heating and hot water operating modes with the operating limit of up to -20°C were met in full, with hot water preparation meeting the requirements of DVGW meets the requirements of DVGW W 551. The plant efficiencies met the project's target of a COP (Coefficient of Performance) of 2.0 for A-7/W55 and reached 2.13, whereby none of the models of comparable size had been offered at the start of the project.

The cost-effectiveness of fan coils was evaluated as an alternative to operating floor heating systems with low flow temperatures. The results show that the annuity of the total costs for systems with fan coils at a supply temperature of temperature of 35 °C compared to a radiator system with a supply temperature of 55 °C is not advantageous.
The vibrations of the refrigeration circuit as well as noise emissions could be reduced by extensive work on the ventilation ducting and with the aid of resonance analyses and the consideration of transfer paths over the pipes could be identified and reduced.

The construction and operation of a demonstrator was realized in the laboratory and prepared for a field test. The special safety requirements for a new A2L refrigerant were taken into account. Further achievements are shown in Table 3.

### Table 6: Survey on objectives and achievements of the NK4HTWP project

<table>
<thead>
<tr>
<th>Target:</th>
<th>Target value: What quantitative improvement should be achieved?</th>
<th>Methodology: How should the objective be achieved?</th>
</tr>
</thead>
<tbody>
<tr>
<td>Environmental influences</td>
<td>Improvement of the CO2 balance;</td>
<td>Application of a refrigerant with a GWP &lt; 150. Achieved, COP at A-7 W/35 &gt; 2.0. Achieved.</td>
</tr>
<tr>
<td>Costs</td>
<td>Lower costs of the evaporator compared to comparative systems</td>
<td>Cost optimization for components. Achieved.</td>
</tr>
<tr>
<td>Acoustics</td>
<td>35db(A) in the neighbourhood taking into account tonality allowances. Not yet tested, see report.</td>
<td>Structural dynamic and acoustic investigation of the plant. Achieved.</td>
</tr>
<tr>
<td>Minimization of the sink temperature in heating mode</td>
<td>Permanent reduction of the vertical temperature by approx. 2K. Achieved.</td>
<td>Design and use of a fan coil unit. Achieved.</td>
</tr>
</tbody>
</table>

#### 5.2.3.10 Self-driven processes: Experimental parameter studies on a two-phase loop thermosyphon cooling system with R1233zd(E) and R1224yd(Z) (TUHH)

Two-phase loop thermosyphon (TPLT) is a promising technology looking at highly effective electronics cooling. Due to strong coupling between the internal and external parameters, in this study experimental tests in steady-state are carried out using R1233zd(E) and R1224yd(Z) as a working fluid to investigate the respective influences and resulting design requirements. The relationship between the governing thermal and flow equations is presented to facilitate the interpretation of the test results. The study shows a stable flow and cooling performance over a wide range of heat loads and re-cooling temperatures. The refrigerant charge is identified as one of the main influencing factors, with an optimum being between excessive subcooling and beginning dry-out. Both tested refrigerants lead to basically similar results, showing minor differences regarding thermal performance and system stability.

#### 5.2.3.11 Component-testing research: Optimal Designed Experiments for Reliable Model Calibration of a fixed-speed Scroll Compressor with R410A and R32 (RWTH)

A compressor performance depends on the fluid with its inlet and outlet conditions, an appropriate oil, and the mechanic compression principle. Based on that, the performance is expressed by the isentropic and volumetric efficiency. To describe a compressor in detail and fluid independent is challenging. Therefore, many theoretical studies have been conducted, while in comparison the authors expressed that experimental studies are rare. To overcome this imbalance, this study focuses a detailed experimental investigation of a 4 kW Copeland scroll compressor with R410A and R32 on a fully automated compressor test stand. Within this paper, the authors present a
method for in-depth uncertainty analysis, which ensures the usability and comparability of their experimental results to other experiments. Results are interacting with an optimal experimental design procedure. This allows to reduce experimental effort up to 70% compared to full factorial experimental designs. To prove the method in a compressor and two fluids, the authors apply it to a fixed-speed scroll compressor with R410A and R32. The test stand can conduct all experiments automatically and part load behaviour of both refrigerants can be summarized in a model according to recent Literature with an overall uncertainty below 8% for R410A. For R32, the method fails. Utilizing this method, the authors aim at open-source experiments in order to accelerate the solution of complex research questions.

5.2.3.12 Standard / Certification research including lifecycle emissions: Domestic heat pumps with natural refrigerants – development of requirements for climate-friendly and energy-efficient appliances for the Blue Angel (HEAT)

Energy efficient heat pumps, driven by electricity from renewable energies, will be key to achieve carbon neutrality in the building sector. Heat pump sales are increasing strongly. In 2020, 1.62 million units were sold across Europe. 120,000 heat pumps for domestic heating were sold in Germany, where they are already being installed in more than half of all newly built houses. Given the huge potential of heat pumps for decarbonizing the building sector as well as the growing market, the German Environment Agency is planning to launch a new Blue Angel ecolabel for heat pumps. The Blue Angel is a well-known, voluntary ecolabel, which aims to help consumers identify the most environmentally friendly products in a certain product group.

Possible criteria for heat pumps are currently being developed by the German consultancy HEAT GmbH as part of the project "Domestic heat pumps with natural refrigerants – development of requirements for climate-friendly and energy-efficient appliances for the Blue Angel". The use of natural refrigerants will be among the criteria. Natural refrigerants are considered the most environmentally friendly option because they have no or very low global warming potential and no harmful atmospheric degradation products. Heat pumps using the natural refrigerant R290 are already well-established in the market and a growing market share is likely. The emissions caused by energy consumption of heat pumps during their lifetime represent the most significant environmental impact as long as the energy is not generated entirely from renewable sources, therefore, high energy efficiency will be the second, crucial criteria.

An interim project report will provide the basis for the development of concrete criteria proposals. This report will be published in early 2022. It examines the market and legislative situation, the various available heat pump types on the market with natural refrigerants and further environmental impacts such as noise emissions.

5.2.3.13 Safety research: Main findings of the OpenFOAM CFD simulation study on a monoblock propane heat pump in ten different installation location scenarios (ISE)

Based on the simulation setup and the different simulated installation locations presented within the last Annex 54 annual report major findings are presented. In general, for a risk assessment of externally installed heat pumps with the refrigerant propane, it is helpful to be able to estimate the concentration of propane in the vicinity of the heat pump in the event of a leak. The dispersion of propane depends on various factors. Based on flow simulations, the following dependencies of propane dispersion were investigated numerically:

- Location of the heat pump
- Wind speed
- Wind direction
- Leakage mass flow

The simulation results show the dispersion of propane depending on the concentration in the air and in relation to the lower flammability limit (LFL) as a 3-dimensional cloud around the heat pump. Maximum dispersion of propane can thus be visualized and calculated. The simulation results are checked for plausibility but were not validated with experimental data. Deeper understanding of propane dispersion of specific heat pumps with better knowledge of leakage locations, leakage mass flows and heat pump geometry can only be revealed by individually adapted simulations and measurements. The main and general findings from the general simulation are shown in Figure 10.

**At medium leakage rates spread 100% LFL predominantly below 1m.**

- The lower ignition limit spreads in a large part of the variation with a distance to the center of the heat pump out under 1m.
- Exception is an installation of the heat pump in a recess

**Rear propane discharge dominant**

- The major portion of the propane cloud flows out the back of the heat pump at the evaporator and then exits to the front outside the housing
- Dominant emergence via a hole located in the ground is not evident

**Frontal wind speeds less critical**

- A frontal wind flow leads predominantly to a smaller dispersion dimension of the propane cloud than (house) parallel flows or flows with the heat pump in the lee of the building.

**Propagation LFL at high leakage up to 3m**

- High leakage rate of 5 g/s propane and low wind resulted in the largest spread of LFLs of up to 3.5m
- High leakage rates of this magnitude correspond to a "rupture leak". In total, 1.2 kg flows out in 4 minutes.

**High wind speed leads to stronger dilution**

- the 100% LFL is mostly inside or only a few centimeters outside the heat pump casing at wind speeds of 5 m/s

*Figure 10: main and general findings of a CFD simulation study of propane dispersion on a monoblock heat pump*

A further but less generic analysis specific for each of the ten chosen installation locations is possible but not included here.
5.2.3.14 Absorption research with low-GWP refrigerants: Efficiency increase of a NH$_3$/H$_2$O absorption chiller – Experimental investigation of a plant concept with plate desorber (IGTE)

Absorption chillers and heat pumps use almost exclusively heat as drive energy and are operated with climate-friendly “zero-GWP” refrigerants. At the Institute for Building Energetics, Thermotechnology and Energy Storage (IGTE), experimental investigations are currently being carried out to increase the efficiency of NH$_3$/H$_2$O absorption systems.

A desorber concept was developed with a plate heat exchanger through which the rich solution flows directly. A plate dephlegmator with downstream condensate separator is used to increase the ammonia mass fraction in the refrigerant. Compared with the widely used desorbers in vessel design with integrated rectification column, the desorber concept with plate heat exchanger is characterized by a very low filling quantity and low manufacturing costs.

Proof of the high efficiency of the absorption system with plate desorber has already been provided for operation as a heat pump as part of the project “Optimization of Absorption Heat Pumps for Use in the Heat Grid 4.0”.

In this paper, newly determined performance data of the direct-flow plate desorber are compared with literature data of a tank desorber in the application case of an absorption chiller. The performance data obtained in the literature are achieved or even exceeded by the new desorber concept.

In the plant concept with plate desorber, no rectification is provided to increase the ammonia content in the vapor, for which only a dephlegmator is used. One challenge in optimizing the plate dephlegmator used to increase the ammonia mass fraction in the refrigerant is the different volume flows of the two fluid streams, refrigerant vapor and rich solution. Thermographic images are used to analyze the temperature distribution in the plate dephlegmator. To increase the transferred heat flux, a concept for an asymmetric plate dephlegmator is proposed on the one hand, and the orientation of the plate dephlegmator is discussed on the other hand.

5.2.4 REFERENCES

References to Section 2.1


References to Section 2.3

Fraunhofer ISE LC150 project: https://www.ise.fraunhofer.de/en/research-projects/lc-150.html

References to Section 2.4

References to Section 2.1

References to Section 2.5

References to Section 2.7

References to Section 2.9
NK4HTWP Project: https://www.ise.fraunhofer.de/de/forschungsprojekte/nk4htwp.html
References to Section 2.10

References to Section 2.14

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