Progress Annual Report

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

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

Task 2: Case studies and design guidelines for optimization of components and systems

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Executive Summary

This report provides a comprehensive, most up-to-date review of current research, and the development of component and their optimizations using low-GWP refrigerants for the heat pump applications. 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 a comprehensive review of R&D progress on components using low-GWP refrigerants for residential applications. The review particularly focused on heat exchangers and compressors. Furthermore, it presents a study on circuitry optimization of tube-fin heat exchangers.

Chapter 2 provides a comprehensive update on recent research in Germany. The chapter presents several projects on heat pumps using low-GWP refrigerants (mainly R290 and R454C). Projects cover novel component investigations, design guidelines and field testing.

Chapter 3 briefly summarizes the activities carried out during the 2nd year of Annex 54 in Italy. It consists of activities from academic and industry institutions, focusing on the development of novel components and systems for low-GWP refrigerants.

Chapter 4 provides the application of low-GWP refrigerants for residential heat pumps, air-conditioners and heat pump water heaters. A total of 10 alternative refrigerants with low-GWP were evaluated with not less than 130 performance tests. These experimental results will be useful for the HVAC community for facilitating the selection of the most promising candidates for replacement of R410A, R134a and R407C in residential heat pumps.

Chapter 5 presents topics foreseen in the reports for 2019, the review of the state of the art technologies and 2020, case studies and design guidelines for the optimization of components and systems, mainly in Sweden.

This report aims at providing a much needed review and update on component R&D using low-GWP refrigerants for heat pump applications. We hope it can be a good reference for researchers, engineers and policy makers across the HVAC industry. We greatly appreciate contributions of authors for each chapter. The report would not exist without their valuable efforts.
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1 Component Research Review and Heat Exchanger Optimization for Residential Air-conditioning Systems with Low-GWP Refrigerants

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1.1 Component Research Review

1.1.1 Introduction

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

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

- Followings are studies with a primary focus on testing and analyzing potential alternative refrigerants. Heredia-Aricapa et al. (2020) [1] reviewed the most up to date work on HFC, HFO, HC, and R744 as being replacement refrigerants for R134a, R404, and R410A. Their work revealed that R32 appears to be one of the leading replacements candidates but has some drawbacks that some mixture can overcome like high discharge temperature and finale energy performance. Harby (2017) [2] presented their findings from analyzing studies regarding hydrocarbons as refrigerants in refrigeration, air conditioning, heat pumps, and automobile air conditioning. They determined that hydrocarbons are not only an eco-friendly option, but they also provide the ability to reduce energy consumption and possess the necessary properties to be an excellent drop-in replacement for halogenated refrigerants. Additionally, they recommended that future research should be focused on HFC/HF mixtures, low charge large capacity, and equipment reliabilities. Mota-Babiloni et al. (2017) [3] summarized recent investigations that resulted from F-regulations on refrigerants and had a primary focus on lower GWP synthetic alternative refrigerants. When considering residential air conditioning systems, they noted that are no significant advantages to using mixtures over R32 as a replacement for R410A. Pabon et al (2020) [4] reviewed the most relevant work of the last decade on the potential of R1234yf. In the end, they found most studies endorsed the use of R1234yf as a replacement for R134a. However, they added that optimization on the circuit would be necessary in order to achieve the optimal operating conditions. Kasaeian et al. (2018) [5] discussed recent theoretical and experimental studies surrounding the application of low-GWP, such as hydrocarbons, hydrofluorocarbons, R744 (carbon dioxide), hydrofluoroolefin, and nano refrigerants. The authors found that nano refrigerants offer many advantages like their thermodynamic performance and effects on lubricants. Despite they highlighted the fact that there needs to be much more research concerning their potential for being a replacement. Bolaji and Huan (2013) [6] examined research on the possibilities of utilizing natural refrigerants in the areas of refrigeration and air conditioning. Through their research, the authors found that natural refrigerants are the most favorable for a long-time replacement in both refrigeration and air-conditioning systems.

- Here is one of the very few reviews on heat exchanger optimization. Tancabel et al (2018) [7] conducted a literature review on shape and topology optimization for the design of Air-to-Refrigerant Heat Exchangers. Through their research, they concluded that there is a need for more studies on Heat exchanger optimization using low-GWP refrigerants.
<table>
<thead>
<tr>
<th>Author</th>
<th>Focus</th>
<th>Highlights</th>
</tr>
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<tbody>
<tr>
<td>Bolaji and Huan (2013) [6]</td>
<td>Natural refrigerants as environmentally friendly alternative refrigerants</td>
<td>Natural refrigerants are the prospects for long time replacements</td>
</tr>
<tr>
<td>Harby (2017) [2]</td>
<td>Hydrocarbons as a replacement in refrigeration, AC and automobile AC</td>
<td>Comprehensive review of latest hydrocarbon research efforts</td>
</tr>
<tr>
<td>Abas et al. (2018) [8]</td>
<td>Natural and synthetic refrigerants.</td>
<td>Utilizing Refrigerant Qualitative Parametric (RQP) quantification model in the refrigerant choice decision</td>
</tr>
<tr>
<td>Kasaeian et al. (2018) [5]</td>
<td>Evaluate the potential of hydrocarbons, hydrofluorocarbons, R744 (carbon dioxide), hydrofluoro olefin, and nano refrigerants as environmentally friendly refrigerants.</td>
<td>Nano refrigerants have many advantages like their thermodynamic performance and lubricant interaction</td>
</tr>
<tr>
<td>Pabon et al. (2020) [4]</td>
<td>R1234yf as working fluid in compression systems</td>
<td>R1234yf endorsed as possible replacement</td>
</tr>
<tr>
<td>Heredia-Aricapa et al. (2020) [1]</td>
<td>Replacing R134a, R404A and R410A with HFC/HFO/HC/R744 refrigerant mixtures.</td>
<td>Mixtures have ability to overcome the drawbacks that R32 possess</td>
</tr>
</tbody>
</table>

Overall much of the effort on summarizing recent work regarding low-GWP refrigerants has been concerned with the ability for them to be replacements to current refrigerants. As a result, there are many gaps in information provided by the reviews. First many of the existing reviews are not specifically dedicated to the application of low-GWP refrigerants in residential applications. Secondly, much of the studies examined only discuss alternative refrigerants as drop in or performance in terms of the entire system, only a very limited amount discussed specifics down to a component level. Lastly, there are hardly any reviews devoted solely to component optimization with new refrigerants. For this reason, the objective of this report is to present a thorough review of all the recent progress and developments in the areas concerning component optimization of low-GWP refrigerants in residential air conditioning systems. This report is intended to serve as the most current update on the R410A phase-down for engineers and researchers. The review is structured into the following three sections: heat exchangers, compressors, and expansion valves/other components. For the heat exchanger portion the two main categories are tube-fin and microchannel. In those two-sub section we further categorized the sources as either a simulation or experiment study. For the compressor the section was only divided between simulation and experimental studies. As a result of the limited number of expansion valve/other component studies these have just been listed under the category for other.
1.1.2 Review of Component Optimization Progress

1.1.3 Heat Exchangers

1.1.3.1 Tube-fin Heat Exchangers

Tube-fin heat exchangers are one of the most common types of heat exchangers investigated in residential air conditioning systems, mostly because of its simplistic design and long commercial usage history. However, tube-fin heat exchangers do have their disadvantages. To begin with, because of the design of tube-fin heat exchangers, they require much material. Additionally, compared to other heat exchanger designs, tube-fin heat exchangers require a higher refrigerant charge. Researchers and engineers have been working on ways to overcome these drawbacks by exploring alternative refrigerants to R410A and the benefits they provide. Overall, R32, R290 and HFO blends are thought to be the best replacement candidates when using tube-fin heat exchangers. Lastly, researchers and engineers have been exploring the possibility of implementing small diameter copper tubes into the heat exchanger designs and using simulation software like ISHED (Intelligent System for Heat Exchanger Design) to optimize the design of the tube-fin Heat exchangers for specific refrigerants. Table 1-2 provides the review summary of recent tube-fin heat exchanger studies.

1.1.3.1.1 Simulation

Domanski and Yashar (2007) [9] described the outcomes of applying ISHED (intelligent system heat exchanger design) simulator to optimize tube-fin condensers. The authors explained that ISHED has the capabilities to produce optimized circuitry architecture. Their results showed that R32 performed the best, while R600a performed the worst. Additionally, the ratio of condenser capacity between the two was calculated to be 1.18.

Shabtay et al. (2014) [14] presents simulation studies on how small diameter copper tube heat exchangers can be applied to low-GWP alternative refrigerants. The authors concluded that all four alternatives R290, R744, HFO blends and R32 all require less charge and provide lower costs. It was found that using R32 reduce that tube sizes by 30% compared to R410A.

Kamada et al. (2016) [15] conducted simulations to optimize heat exchangers in heat pumps for R32 and HFO-mixtures. In particular, they focused on how to overcome the temperature glide to achieve a better performance by optimizing the number of passes. They concluded, that HFO blends, like R32/R125/R1234yf, that have smaller temperature glide, will have a reduced ratio of improvement of heat transfer by optimizing the number of passes.

Besher et al. (2016) [16] examined three different optimization simulations on heat exchangers. The author focused on using genetic algorithms to optimize the small diameter tubes in the heat exchangers in order to optimize the designs of heat exchangers for the use of both R410A and R32. They concluded that It is possible to reduce the heat exchangers cost by 60% without having to reduce the heat exchangers performance. Additionally, the authors noted that the systems refrigerant charge can be decreased by 35% by using the ASHP system.

Cho and Domanski (2016) [17] examined the performance of twelve R22 and R410A low-GWP alternatives after optimizing the tube-fin heat exchangers using first-principle based simulation models and ISHED, Intelligent System for Heat Exchanger design. They reported that all the alternative refrigerants benefited from the optimization of tube-fin heat exchangers. At the same time, R744 and R717 saw the highest performance improvements with their capacities increasing by at least 13% and 10% respectively.

Li et al. (2019) [19] presented a novel integer permutation-based Genetic Algorithm (IPGA) for optimizing Tube-fin heat exchanger circuitry. Through their simulations they found that a 2.4–14.6% increase in heat exchange capacity is resulted from applying IPGA to an A-shaped indoor unit. Additionally, they noted in comparison to other optimization methods IPGA results in higher capacity, lower pressure drop and better manufacturability.
Table 1-2: Review of Recent Tube-Fin Studies

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

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

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

In et al. (2014) [11] performed experimental studies on tube heat exchangers using two alternative refrigerants to R410A, R32 and L-41b (HFO blend). When analyzing the HFO blend it was discovered that its performance was much worse than that of R410A. The authors justified the reason for this as being a result of the HFO blend having a higher viscosity, which causes a larger pressure drop and a lower mass flow rate. It was suggested that partial optimization on the tube size and the number of passes in the heat exchanger would increase the performance for the HFO blend.

Cheng et al. (2014) [12] explored R32 and R290 as possible alternatives to R410A in small tube heat exchangers. They noted that R290 had the smallest refrigerant charge, half of the charge required for R22, R32 and R410A. Additionally, the authors described how R290 could be the best alternative refrigerant for small tube heat exchangers in air conditioners.

Ren et al. (2014) [13] investigated experimentally the effect of a suction line heat exchanger on a R290 Air Conditioner with small diameter copper tube. Their experiment data showed that the suction line heat exchanger improved the cooling capacity by 5.3% and reduced the refrigerant charge by 6%.

1.1.3.2 Microchannel Heat Exchangers

Over the past two decades’ researchers and engineers have started to explore the possibility of utilizing microchannel heat exchangers with R410A alternative refrigerants. They have concluded that R32, R744, and HFO blends contain the best characteristics to achieve the best results when using microchannel heat exchangers. Additionally, their work has shown that microchannel heat exchangers provide two major advantages. The first advantage being that microchannel heat exchangers reduce the required refrigerant charge and secondly, they have the ability to increase the cooling capacity. Table 1-3 provides the review summary of recent microchannel heat exchanger studies.

1.1.3.2.1 Simulation

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

Yun et al. (2007) [21] investigated microchannel heat exchanger designs for R744 air-conditioning systems. It was determined that the microchannel heat exchangers performance can be enhanced by increasing the two-phase region, modifying the fin spacing, and varying the flow rate of the refrigerant to the slab.

Shen et al. (2016) [25] presented that microchannel heat exchangers are capable of reducing the refrigerant charge up to 55% for some of the refrigerants, with R290 requiring the least amount of charge. They were also able to conclude it would be possible to use R290 in a small capacity room air-conditioner with only microchannel heat exchangers.
### Table 1-3: Review of Recent Microchannel Studies

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

López-Belchí and Illán-Gómez (2017) [27] investigated the use of R32 as a replacement for R410A using a microchannel condenser. They calculated through numerical analysis that if R32 were directly used a drop-in replacement could either have a positive or negative effect on the refrigerant charge and energy efficiency. However, it was concluded that if the system was modified to run with R32 then the refrigerant charge will decrease, and the energy efficiency would increase.

Zanetti et al. (2018) [28] discussed the importance of using a microchannel heat exchanger with low-GWP refrigerant like R32 to reduce refrigerant charge. In addition, they showed through a
numerical simulation that by using a microchannel heat exchanger as a condenser reduces the refrigerant charge by about 30% compared to a tube-fin heat exchanger.

1.1.3.2 Experiment

Pham and Rajendran (2012) [22] presented data comparing R32 and HFO blends to R410A through various drop in test using different heat exchangers. They reported that in their experiment it was found when using a microchannel heat exchanger design the refrigerant charge is reduced by 30~40% compared to using a tube-fin heat exchanger.

Fuentes and Hrnjak (2012) [23] displayed experimental results of potential charge reduction for several refrigerants using a microchannel evaporator. Their experiment was designed to compare the refrigerant charge vs the hydraulic diameter of the evaporator for 1kW refrigeration system causing 2% difference from ideal COP due to evaporator pressure drop. They concluded that R744 showed the highest potential for possible refrigerant reduction.

Tian et al. (2015) [24] explored swapping a tube-fin heat exchanger with a micro channel heat exchanger and compared them through experiment testing using R32/R290. They found by using the microchannel heat exchanger the power consumption was reduced by .4%. Additionally, the microchannel heat exchanger increased the cooling capacity by 6.4%.

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

1.1.3.3 Other Designs

1.1.3.3.1 Simulation

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

1.1.4 Compressors

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

1.1.4.1 Simulation

Table 1-4 provides the review summary of recent simulation studies on compressors. Sethi and Motta (2016) [30] discussed simulations that show the benefits of using low-GWP alternative over R410A in stationary air conditioning systems. They found that both the alternative refrigerants researched, R447B and R452B, had discharge temperatures within 10C and 5C of r410a respectively. As a result, discharge temperature mitigation is not required for both R447B and R452B.
Table 1-4: Review of Recent Simulation studies on Compressors

<table>
<thead>
<tr>
<th>Author</th>
<th>Type</th>
<th>Focus</th>
<th>Highlights</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pham and Monnier (2016) [31]</td>
<td>Scroll</td>
<td>Explored possible low-GWP refrigerant solutions to R410A.</td>
<td>Recommended reducing discharge temperature by optimizing compressor internal design, employing an injection system, improve oil properties.</td>
</tr>
<tr>
<td>Mota-Babiloni et al. (2017)</td>
<td>Scroll, Rotary</td>
<td>R32 as a possible alternative for R410A.</td>
<td>R32 resulted in too high of a discharge temperature as a direct replacement.</td>
</tr>
<tr>
<td>Berkah Fajar et al. (2020) [33]</td>
<td>Reciprocating</td>
<td>Analyzed the energy and exergy of a small vapor compression system with R410A and R290.</td>
<td>R290 cut the compressor power consumption by 35.7% compared to R410A.</td>
</tr>
</tbody>
</table>

Pham and Monnier (2016) [31] examined low-GWP replacement refrigerants for R410A. From their simulations they concluded that the replacements all have high compression heat leading to a high discharge temperature. They listed possible solution to reduce the discharge temperature as 1) optimizing compressor internal design, features and materials of construction 2) employing compressor vapor injection (VI) or liquid injection (LI) cycle, or 3) improve the oil to enable higher maximum allowable discharge temperature.

Mota-Babiloni et al. (2017) [32] examined the possibility of using R32 as a possible alternative to R410A. In their research they found that the compressor discharge temperature for a direct replacement using R32 was too high. It was suggested that modifying the system to utilize a liquid, vapor or two-phase refrigerant injection system in order to decrease the compressor discharge temperature. Additionally, they noted that using a R32 mixture could also decrease the compressor discharge temperature.

Berkah Fajar et al. [33] analyzed the energy and exergy of a small vapor compression system with R410A and R290. Their simulations showed that the power consumption of the compressor with R290 compared to R410A decreased by 35.7%.

1.1.4.2 Experiment

Table 1-5 provides the review summary of recent experimental studies on compressors. Pham and Rajendran (2012) [22] focused on potential R410A low-GWP replacements for air conditioning and heat pump systems. One of their experiments compared the compressor discharge temperature of R410A with an optimized compressor and R32 with a non-optimized compressor. The results showed that R32’s discharge temperature was about 22% higher than that of R410A. They suggested the following optimizations for the R32 compressor to improve the discharge temperature: optimize compressor internal design, use compressor vapor injection, improve the oil.

Barve and Cremaschi (2012) [34] conducted an experimental comparison in energy performance of refrigerants R32 and R1234yf in a R410A heat pump split system for residential applications. Their results for the efficiency of the compressor showed the R32 was generally higher than R410A and R1234yf was lower than R410A.
Karnaz (2014) [35] investigated how low-GWP refrigerant perform with current POE lubricants. Their experiment showed that some refrigerants like R-32 and R-1234ze will require lubricant optimization on the current POE lubricants.

### Table 1-5: Review of Recent Experimental Studies on Compressor

<table>
<thead>
<tr>
<th>Author</th>
<th>Type</th>
<th>Focus</th>
<th>Highlights</th>
</tr>
</thead>
<tbody>
<tr>
<td>Barve and Cremaschi (2012)</td>
<td>Hermetic reciprocating</td>
<td>Experimental comparison in energy performance of refrigerants R32 and R1234yf to R410A.</td>
<td>R32 had higher compressors efficiency than R410A.</td>
</tr>
<tr>
<td>In et al. (2014) [37]</td>
<td>Swing</td>
<td>Partial optimization of a compressor using low-GWP refrigerants.</td>
<td>R32 system’s compressor discharge temperature was 20-25% larger than the R410A system.</td>
</tr>
<tr>
<td>Schultz (2014) [38]</td>
<td>Scroll</td>
<td>Low-GWP refrigerant tests in heat pump designed for R410A.</td>
<td>Cycle and hardware modifications need to reduce R32’s discharge temperature.</td>
</tr>
<tr>
<td>Yuuichi Yamamoto et al. (2015) [39]</td>
<td>Swing, Rotary</td>
<td>Compared an optimized swing compressor with R32 to a compressor with R410A.</td>
<td>COP for R32 was 101.2% of that of R410A.</td>
</tr>
</tbody>
</table>

Hessell et al. (2014) [36] discussed the challenges of finding lubricants for low-GWP refrigerants like R32 and blends. Through experimental test they found that advanced POEs (APOE) to be the best option. For the reasons of APOEs having the ability to be made to have acceptable miscibility in R32, while maintain the same performance as prior lubricants with R410A.

Schultz (2014) [38] conducted numerous tests on a heat pump design for R410A using low-GWP refrigerants. They found R32 offered potential benefits in its ability to increase capacity and efficiency, however R32 was shown to have high discharge temperature when using a scroll compressor designed for R410A. The authors noted that cycle and hardware modification would be required to improve the operating conditions for R32.

In et al. (2014) [37] detailed experimental results from the partial optimization of a compressor using low-GWP refrigerants. They found that the R32 system’s compressor discharge temperature was 20-25% larger than the R410A system.

Yamamoto et al. (2015) [39] described an experiment that compared an optimized swing compressor with R32 to a compressor with R410A. The compressor was optimized by creating spaces above and below the compression chamber to reduce the suction overheating losses. Additionally, the discharge port diameters were optimized in order to benefit from the reduction in compression loss. In the end it was found that the COP for R32 was 101.2% of that of R410A.
1.1.5 Other Components

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

1.1.5.1 Valves

Barve and Cremaschi (2012) [34] conducted experiments and discussed results of using an optimized expansion valve with R32 and R1234yf. Through their experiments it was found that by optimizing the expansion valve the capacity for R1234yf was improved by up to 10%.

1.1.6 Conclusions

This report presents information surrounding the current status of component optimization of low-GWP refrigerants in residential air conditioning systems. This is an extension of facts and evidence that intended to help the decision on a possible solution to R410A.

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

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

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

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

1.2.1 Introduction

Tube-fin heat exchangers (TFHXs) are prominent components in air conditioning and heat pump systems. This type of heat exchanger consists of a bundle of tubes with fin sheets. The performance of TFHXs is greatly affected by a large number of structural parameters (tube diameter, tube length, fin type, fin thickness, etc.) and therefore, conducting optimization on such parameters can significantly improve its performance (Huang et al., 2015). In addition to those parameters, the configuration of tube connections, i.e. refrigerant circuitry, determines the refrigerant flow path, and also significantly impacts the heat exchanger performance (Chwalowski et al., 1989; Wang et al., 1999; Bigot et al., 2000; Liang et al., 2001; Ding et al., 2011; Ye and Lee, 2012; Joppolo et al., 2015). For an existing TFHX design with a given geometry, performing optimization on refrigerant circuitry is more convenient and cost-effective than varying other structural parameters. For example, changing the circuitry is a matter of changing U-bend length and orientation, whereas changing tube-spacing can require new fin dies, which is costly and has much longer lead time.

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

1.2.2 Integer permutation based Genetic Algorithm

1.2.2.1 Representation of HX circuitry

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

![Diagram](image)

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

### 1.2.2.2 Selection, Crossover and Mutations

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

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

1.2.3 Results

In order to address the above mentioned concerns, different combinations of constraints are introduced in the optimization problem. Figure 1-2 shows the optimal solutions from three constrained optimization runs and Table 1-6 presents a detailed analysis of those optimal designs. In Table 1-6, ‘U-bends L1’ and ‘U-bends L2’ are the number of U-bends which span 1 tube row and 2 tube rows, respectively. ‘U-bends ≥L3’ is the number of long U-bends which span more than 2 tube rows. Case (c) in Table 1-6 shows an optimization case, in which manufacturing constraints, i.e., inlets and outlets on the same side, preventing long U-bends and preventing partial overlap U-bends crossovers, are enforced. Figure 1-2 (c) shows its optimal solution. It results in 13.8% capacity improvement. The optimal solution has 2 circuits, which still results in 12 times higher pressure drop than the baseline. Nevertheless, the circuitry can be readily manufactured. Case (d) in Table 1-6 is another constrained optimization run, in which only refrigerant pressure drop constraint is applied. This constraint restricts the designs to have equal or less refrigerant pressure drop than the baseline. Figure 1-2 (d) shows its optimal solution which has 4 circuits, the same as the baseline, a 4% increase of capacity than the baseline, and slightly lower pressure drop than the baseline. It is worthwhile to mention that some existing methods (Domanski et al., 2004; Ploskas et al., 2017) do not demonstrate pressure drop constraints, although various manufacturability constraints were investigated. Case (e) in Table 1-6 is a highly constrained optimization run, in which both manufacturability constraints and the pressure drop constraint are applied. Figure 1-2 (e) shows its optimal solution which achieves 2.4% capacity increase and 1% less pressure drop than the baseline. The circuitry has good manufacturability without any long U-bends or U-bend crossovers. Compared with other cases, the optimal solution from case (e) has desired thermal, hydraulic and manufacturability performance. The optimal evaporators in case (d) and (e) provide similar amount of latent cooling as baseline, since all three evaporators have similar sensible heat ratio (SHR). Since GA is a stochastic optimization approach, different runs have the possibility to converge to different solutions. In order to access the robustness of proposed optimization approach, the aforementioned optimization case study with four different constraint combinations (case (b) to case (e) in Table 1-6) are repeated 15 times per each case. The small variations of results between multiple optimization runs for each case indicates that the proposed approach has good robustness.

![Figure 1-2: Optimal circuits under different constraints](image)

(a) baseline; (b) unconstrained; (c) with mfg. constraints; (d) with refrigerant DP constraint; (e) with mfg. and refrigerant DP constraints.
Annex 54, Heat pump systems with low-GWP refrigerants

### Table 1-6: Evaporator optimization results with different constraints

<table>
<thead>
<tr>
<th>Case Constraints</th>
<th>Baseline (a)</th>
<th>(b) No constraints</th>
<th>(c) Mig, constraints</th>
<th>(d) Refrigerant DP constraint</th>
<th>(e) Mig, constraints and DP constraint</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ref. DP [kPa]</td>
<td>11.8</td>
<td>972.5 [181x↑]</td>
<td>160.5 [12x↑]</td>
<td>11.4 [3.8%↑]</td>
<td>11.7 [10%↑]</td>
</tr>
<tr>
<td>SHR</td>
<td>79.6%</td>
<td>67.6%</td>
<td>72.5%</td>
<td>79.8%</td>
<td>80.6%</td>
</tr>
<tr>
<td>U-bends L1</td>
<td>82</td>
<td>61</td>
<td>51</td>
<td>17</td>
<td>62</td>
</tr>
<tr>
<td>U-bends L2</td>
<td>0</td>
<td>4</td>
<td>35</td>
<td>9</td>
<td>22</td>
</tr>
<tr>
<td>U-bends ≥ L3</td>
<td>2</td>
<td>2</td>
<td>0</td>
<td>58</td>
<td>0</td>
</tr>
<tr>
<td>Collinear U-bends</td>
<td>0</td>
<td>3</td>
<td>0</td>
<td>43</td>
<td>0</td>
</tr>
</tbody>
</table>

### 1.2.4 Conclusions

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

### 1.3 Acknowledgment

This work was supported by the Center for Environmental Energy Engineering (CEEE). The authors acknowledge the support of Zach Spencer in searching and summarizing part of the references. The authors also acknowledge the heat exchange optimization study conducted by Modeling and Optimization Consortium (MOC) at the CEEE. The full paper is published online (https://doi.org/10.1016/j.ijrefrig.2019.04.006).

### 1.4 Reference


Annex 54, Heat pump systems with low-GWP refrigerants


2 Review of State-of-the-art Technologies, Case Studies and Design Guidelines for Optimized Components and Systems in Germany

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2.1 Review of State-of-the-art Technologies

2.1.1 Introduction

To evaluate the state-of-the-art technologies, it was decided to report on three issues:

- Most recent findings of the field test for heat pumps based on the findings of the funded project “WP Smart im Bestand”, FKZ 03ET1272A.
- A review on the funded projects (until now) based on the enArgus database (www.enargus.de).
- BAFA results on the use of refrigerants back in the year 2016.

The second issue is not necessarily a well-reflecting figure on state of the art but it is a useful starting point to understand what kind of changes on refrigerants could occur – at least partially – on the German market within the next years.

2.1.2 Field tests for heat pumps

2.1.2.1 Introduction

In Germany large monitoring heat pump field tests are executed since 2006. In an uninterrupted series of monitoring projects since this time project with different manufacturers were realized with about 200 residential houses. In most recent projects that ended in 2019 a project dedicated to quantify (1) the most recent efficiency and (2) the potential to operate heat pumps in a “smart-mode” to interact as beneficial as possible with renewable electrical energy production.

2.1.2.2 Recent findings

In order to be able to reconstruct the efficiency values, the balance limits 3 and 5 are presented on the basis the, exemplarily for an outside air heat pump. With the balancing limit 3 (AZ\(^1\) 3) the thermal energy made available by the heat pump for space heating and TW-warming is measured directly after the heat pump, thus before any storage tanks (grey). As input the necessary electrical energy of compressor, control, heat source drive (fan or brine pump) and electrical heating element is considered. The legionella pump shown serves to mix the TW storage tank and is only very rarely part of the installation (green). At the AZ 5 the thermal energy is measured after any storage tanks (blue). The electrical energy consumers correspond to those of AZ 3 plus the storage charging pumps, the circulation pump and the pump for the fresh-water station.

![Figure 2-1: System limits for the calculation of seasonal performance](image)

\(^1\) AZ stands for German "Arbeitszahl", the seasonal performance figure.
Figure 2-2 (22 air-to-water heat pumps) and Figure 2-3 (9 ground-source heat pumps) show the annual working figures (AWP) of the AZ 3 (light grey) and AZ 5 (dark blue). The selection of systems is limited to those heat pumps to which both balance sheet limits apply. To the left of each SPF, the energy produced by the heat pump is shown, proportionally according to room heating (red) and domestic hot water heating (blue). Furthermore, the graphs show the flow and return flow temperatures measured before the storage tanks as well as average temperatures in room heating (orange rhombus) and drinking water heating mode (blue rhombus). The maximum drinking water tap temperatures are shown in the form of blue triangles. The systems are sorted in ascending order according to the percentage difference between SPF 3 and SPF 5.

For outdoor air heat pumps, the range of SPF values for the balancing limit AZ 3 is between 2.5 and 4.6, with an average value of 3.2. The SPF value range for systems with the balancing limit AZ 5 is between 2.1 and 3.9 with an average value of 2.8. Remarkable are the big differences between the JAZ of both balancing limits, which are between 3% and 27% (average value: 13%). Although the additional electrical energy consumers (see above) also play a role in AZ 5, the main influence is caused by storage losses. The main drivers are thus the average temperatures in the space heating buffer storage tank (BS, in German PS) and in the domestic hot water (DHW, in German TWS) storage tank as well as the energy shares between the two operating modes. Since the storage tank temperatures were not measured, the storage tank charging temperatures are used here in a simplified way. Except a single ground-source heat pump investigated within the project (not considered here), the average heating circuit temperatures are always below the average DHW charging temperatures. Thus, both an increase in the difference between DHW and BS charging temperatures and an increase in the energetic share for DHW heating should contribute to an increase in the difference between SPF 3 and SPF 5. This trend can be traced using the quantities shown in Figure 2-2. For ground-source heat pumps, the SPF value range for the balancing limit AZ 3 (for AZ 5 in brackets) is between 3.3 (2.9) and 4.6 (3.8) with an average value of 3.9 (3.3). The deviation between the two SPF values is between 9% and 24%, with a mean difference being 15%. The correlations between these differences and the temperature differences (for space heating or domestic hot water heating) as well as the energy shares for domestic hot water heating visible for the outdoor air-to-water heat pumps cannot be observed for the ground-source heat pumps. Only the three highest energy shares for domestic hot water heating can be assigned to the heat pumps with the three highest JAZ differences.

**Figure 2-2:** Annual SPF of the balancing limits AZ 3 and AZ 5 as well as temperature and energy values of 22 outdoor air-to-water heat pumps sorted by the percentage difference of both annual performance factors
Figure 2-3: Annual SPF of the balancing limits AZ 3 and AZ 5 as well as temperature and energy values of 9 ground source heat pumps sorted according to the percentage difference of both annual performance factors

In the Figure 2-4 different types of heat pump system configurations are clustered to present the share of CO\textsubscript{2} are represented with the employment of this heat pump instead of a gas device. A differentiation according to heat source types is not (yet) made here. The first group (left) contains all heat pumps that can be evaluated for the balance limit 3. With the exception of one system (~5%), CO\textsubscript{2} emissions of 25% to 60% would be avoided. The average value is 42.6%. The second group also refers to the balance limit 3, but only considers the room heating mode. It is remarkable in this group that exactly one heat pump has a better YES in room heating mode than in domestic hot water heating mode. This shifts the already negative CO\textsubscript{2} avoidance of this one system to -9%, while the CO\textsubscript{2} avoidance of all other systems increases and averages 47.1%. Including (3rd group) the plants which were still operated "smart" during the evaluation period and thus usually had higher heating circuit temperatures, the values of CO\textsubscript{2} avoidance shift slightly downwards. In the 4th group only, those plants are considered which can be evaluated for a balance limit 5. This means that the system is comprehensively evaluated including storage losses and all electrical energy consumers (apart from heating circuit pumps). Assuming this balance limit, the values for CO\textsubscript{2} avoidance are between 9% and 52%, with an average value of 33.3%.

Figure 2-4: CO\textsubscript{2} avoidance potential of the examined heat pumps, differentiated according to balancing limits AZ 3 or AZ 5, the operating mode (space heating and domestic hot water heating), and additionally a “smart operated”-mode when replacing a gas boiler
2.1.2.3 Refrigerants in heat pumps that have been tested in the field test

While more and more units with alternative refrigerants, such as R290 (propane), came onto the market towards the end of the project term, the traditional refrigerants dominate in the systems investigated here. As Figure 2-5 shows, more than half of the heat pumps are operated with R410A (GWP: 2,088) and about a quarter with R404A (GWP: 3,922). Furthermore, the refrigerants R407C (GWP: 1,744) and R134A (GWP: 1,430) are also present. Only one plant uses the alternative refrigerant R290 (propane, GWP: 3). For one system the used refrigerant type was unknown (in German: unbekannt).

![Figure 2-5: Share of used refrigerants](image)

2.1.3 Review of funded projects

By analysing the BMWi project database EnArgus\(^1\) it is possible to gather information on the energy-related on all projects (running or terminated) on any field of interested. The database was filtered for the keywords “heat pump” (in German: Wärmepumpe) and “refrigerant” (in German: Kältemittel). Only project that were started with January 2015 were investigated. Since then the number of projects related to the keyword “heat pump” were 241 running projects. The number of projects was too large to figure out which project actively develops new refrigerant circuits since not all project reports were not yet or not at all available.

But the result group for “heat pump” had a large intersection with another result group when searching for “refrigerant”. The number of projects related to the term “refrigerant” was 19 projects. From Figure 2-6 it becomes clear that the focus on the new development of cycles often uses so-called natural refrigerants. It is avoided to state here that new developments are dominated on natural refrigerants since there were more than 240 projects heat pump projects. But it is probable that the major share of projects serves this goal. As expected, many projects focus on vapor compression systems as the investigated thermodynamic cycle, see Figure 2-7.

\(^1\) [www.enargus.de](http://www.enargus.de)
Figure 2-6: Range of refrigerants in energy-related projects in Germany related to the new development of thermodynamic cycles

Figure 2-7: Range of different thermodynamic cycles being investigated in energy-related projects in Germany
Annex 54, Heat pump systems with low-GWP refrigerants

From the application type that could be identified also heat pumps play the major role for innovative systems that are investigated in Germany, see Figure 2-8.

2.1.4 Analysis of BAFA data

As a third part of this report on the state-of-the-art technologies the BAFA data on heat pumps for what a financial incentive is given to support renewable thermal energy production for heat supply in residential houses. The list of supported heat pump models is publicly\(^1\) available. It was tried to gather information on the most important manufacturers, their overall number of air-to-water heat pump models using a frequency inverter that are supported and the share of refrigerants being used.

The following list of manufacturers and their portfolio for June 2016 as well as for June 2020 was investigated: ait-deutschland, Bosch Thermotechnik, Daikin, Fujitsu, Glen Dimplex, Heliotherm, Mitsubishi Electrics, Ochsner, Panasonic, Remko, Samsung, Stiebel Eltron, Thermia, Vaillant, Viessmann, Waterkotte, Weishaupt, Wolf. Each of the two corresponding lists of subsidized heat pumps was downloaded at the same time/date.

To identify what systems were only available in 2016 all heat pumps being listed in the 2020 table were negated. Overall, 2731 models could be identified as being supported in 2016 compared 4502 models being supported in 2020. In 2016 from about 402 models the refrigerant could be identified based on the list on manufacturers and the information that could be gathered. In 2020 the same number could be gathered for 4502 models. The search was not supported by any manufacturer and was based on the research on a manufacturer’s website.

Below in Figure 2-9 the share of refrigerants is Figure 2-10 for both years are displayed.

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\(^1\) [https://www.bafa.de/SharedDocs/Downloads/DE/Energie/beg_waermepumpen_anlagenliste.html](https://www.bafa.de/SharedDocs/Downloads/DE/Energie/beg_waermepumpen_anlagenliste.html), last visit at 1\(^\text{st}\) February 2021.
Annex 54, Heat pump systems with low-GWP refrigerants

2.1.5 References


Pavel Makhnatch, Hatef Madani, Rahmatollah Khodabandeh, KTH Royal Institute of Technology, Department of Energy Technology - Sweden – 12th IEA Heat Pump Conference Rotterdam 2017


2.2 Case Studies and Design Guidelines for Optimized Components and Systems

2.2.1 Summary

This task report tries to focus on design guidelines related to Annex 54 activities that and case studies that are either part of it or so close in its topics (related to low-GWP R&D activities – as with the work of TU Braunschweig, TU Dresden, and RWTH Aachen) that it was included. Safety research activities are not well defined as activity within the legal text and thus it was interpreted as being part of the design guidelines that could be developed as part of Task 2.

The German task report does not claim in any sense to clarify comprehensively about all activities in Germany. To some extent these findings include component level activities (TU Dresden), which are closer to Task 1 topics and not only case studies addressing Task 2 topics. However, the case studies are focused on different type of domestic heat pumps (Fraunhofer ISE) as well as laundry dryers (TU Braunschweig, TU Dresden) and – in one case – a refrigerator (TU Dresden). The topic about multi-objective decision-making tools as used by RWTH is a bit exceptional and has – in its presented form – no experimental approach. Nevertheless, it could be categorized as a design guideline to select the bank of feasible refrigerants as discussed in Task 1.

2.2.2 System developments and refrigerants put into focus

2.2.2.1 Background and Introduction

This section contributes to understand better the market situation for heat pumps in Germany. It continues the survey of Germany’s Task 1 report. In a development context two groups of refrigerants are in the focus of manufacturers:

- A2L (mainly R454C, R452B and R32) as well as
- A3 (almost only R290, on a small scale R744, too).

A small share of R290 systems is already – even on a long-term scale – market-available, these are

- small exhaust air heat pumps for passive houses or similar buildings with small heat loads, or solely small types dedicated for domestic hot water production,
- ground source heat pumps as well as
- outdoor air-to-water heat pumps.

2.2.2.2 Outlook

Research is ongoing for both refrigerant groups and all these three types of domestic heat pump systems. Of course, the activities are more diversified for A2L as for A3 refrigerants. The main objective is to improve and enlarge the availability of safe, acoustically optimized, energy-efficient and robust outdoor monobloc air-to-water heat pumps. One decisive step is the ongoing development process on appliances that would allow low charge indoor mounted heat pumps currently applied in several projects.

For R&D development projects the situation is not that much different to activities for A3 appliances. An ample range of heat pumps operated with F-Gases is already available or under development, too. For this reason, no itemized list is placed here on the types what F-Gases were used. The trend in R&D activities does not only comprise the
introduction of refrigerants for single family houses. Since a few years it comprises, too, R&D activities new large-capacity heat pump models for multi-family dwellings. In Germany the share for multi-family houses and rented flats is high and it is expected that decommissioning of old equipment and substitution (partially with heat pumps) will increase in the near future.

2.2.3 Design guidelines related to safety for heat pumps operated with A3 refrigerants

2.2.3.1 Background and Introduction

The national project comprises as part of the Annex 54 work CFD simulations to uncover safety issues for 10 different installation scenarios adjacent to residential houses. This topic is not precisely described within Task 2 of the legal text but several national partners and participants within this Annex 54 have referred to the safety issues that are related to A3 refrigerants.

This work is conducted by Fraunhofer ISE and is ordered by the Federal Ministry of Economic Affairs and Energy. The work is still ongoing and was mainly realized by a Master Thesis [1] and a student specialized in CFD simulation. Results are expected to be finished for all cases until mid of 2021. The results will be incorporated into the final report of Annex 54.

The standard IEC 60335-2-40 provides safety requirements for flammable refrigerants, which helps to reduce the risk of using them. As the heat pump unit could be in a partially closed environment (garage, sink in the ground, etc.) and provided the probability of a leak occurs it could leak into the surroundings with unacceptable high residual times that could lead to gas concentrations larger than 20% LFL and smaller clouds with such isopleths. It is required to know the orientation and dimension of such a cloud so that preventive measurements like sensors could be established or installation instruction adapted. The aim of these simulation campaigns is to give guidance for experimentally realized tests according to IEC EN 60335-2-40. Clouds and isopleths at different concentration profiles of interest are simulated or in a post-processing step calculated.

The direct way to know the position of the gas cloud quantitatively is through experiment. But direct physical testing requires a heavy initial investment and requires time to setup. Despite that, the resolution of data measured is limited, and only certain testing conditions can be considered. So, the known alternative to experimental testing is computational fluid dynamics, which provides numerical data with high resolution, this high-resolution data provides an opportunity to understand and interpret the results better and optimally could simplify any experimental procedure.

The work in [2] and [3] describes the importance of numerical approach to calculate spread of flammable refrigerant in closed spaces. The challenge with numerical simulation is finding a way to evaluate the results of the simulation. The typical way to evaluate simulation results is to compare them with experimental results [4] and [5]. If experimental results are not provided, then a mesh dependency study is performed. And also, as the simulation of the gas leak is time-dependent, studying properties of the cloud at a specific point of time is not enough and does not extol the nature of its complete transient behavior. So, the leakage has to be studied with respect to time, though out the simulation time.
To perform gas dispersion simulations, there are many commercial software programs available in the community. And most of the companies are using commercial packages like Flacs (which is especially designed for safety analysis in chemical plants or offshore oil platforms etc.), Fluent, and CFX. Along with them, there exist few open-source packages like Open-FOAM (OF) and Fire dynamics simulator. Commercial packages are user friendly but have limitations to develop your own solvers and to edit existing code to opt to your simulations.

In the shadow, OF is emerging slowly, mainly in the field of research. Although OF is not user-friendly, it provides full access to the solvers and provides an opportunity to develop your own solvers, boundary conditions, and whatnot. In the present research, a solver called rhoReactingBuoyantFoam is slightly modified to suit the current simulation case and named as rhoDyMbuoyantFoam. For preparing the geometry, open-source software called Salome is used. Salome provides a platform for pre- and post-processing for numerical simulation. On further, for post-processing results from OF, an open-source package called Paraview is used.

### 2.2.3.2 Setup of the simulations

Ten cases were defined in cooperation with manufacturers to identify neuralgic situations that could occur through installation locations and/or distances between neighbor properties or borders to publish space that could surround a house completely or partially, see Table 2-8. Each of these cases got its own mesh convergence study in case that its reusage led to physically incorrect setups. Starting/boundary conditions were defined for each case, but these parameters are quite similar between each other. For this reason, Table 2-7 only provides an insight in these parameters. Each case is simulated eight times resulting into 80 simulations each with isopleth calculations for 25%, 50%, 75%, and 100% with results into 320 3-dimensional isopleth surfaces. These clouds will be reusable for validation measures and to study specific issues if there are any from manufacturers’ side.

**Table 2-7: Exemplary presentation of case-specific starting or boundary conditions.**

<table>
<thead>
<tr>
<th>S.No</th>
<th>case</th>
<th>$m$ (g/s)</th>
<th>Wind velocity (m/s)</th>
<th>Wind direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>case 1</td>
<td>0.6</td>
<td>0.5</td>
<td>xmin</td>
</tr>
<tr>
<td>2</td>
<td>case 1</td>
<td>0.6</td>
<td>0.5</td>
<td>ymin</td>
</tr>
<tr>
<td>3</td>
<td>case 1</td>
<td>0.6</td>
<td>5</td>
<td>xmin</td>
</tr>
<tr>
<td>4</td>
<td>case 1</td>
<td>0.6</td>
<td>5</td>
<td>ymin</td>
</tr>
<tr>
<td>5</td>
<td>case 1</td>
<td>5</td>
<td>0.5</td>
<td>xmin</td>
</tr>
<tr>
<td>6</td>
<td>case 1</td>
<td>5</td>
<td>0.5</td>
<td>ymin</td>
</tr>
<tr>
<td>7</td>
<td>case 1</td>
<td>5</td>
<td>5</td>
<td>xmin</td>
</tr>
<tr>
<td>8</td>
<td>case 1</td>
<td>5</td>
<td>5</td>
<td>ymin</td>
</tr>
</tbody>
</table>
### Table 2-8: Survey on the ten different scenarios that were defined.

<table>
<thead>
<tr>
<th>Case 1: Wall-mounted (baseline)</th>
<th>Case 6: Roof-mounted pitched roof</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 2: Stand-alone (base)</td>
<td>Case 7: Stand-alone in depression</td>
</tr>
<tr>
<td>Case 3: Wall-mounted in niche</td>
<td>Case 8: Wall-mounted with light-well</td>
</tr>
<tr>
<td>Case 4: Stand-alone with hedge</td>
<td>Case 9: Stand-alone with light-well</td>
</tr>
<tr>
<td>Case 5: Roof-mounted flat roof</td>
<td>Case 10: Wall-mounted second floor</td>
</tr>
</tbody>
</table>

#### 2.2.3.3 Results

As announced the results will be presented as soon as all simulations are completed. But it is exemplarily shown how the cloud building process behaves different depending on the flow resistances that are within the free flow conditions outside the unit. In Figure 2-111 such a comparison of two similar cases is presented. The green isopleth line being projected on the ground is defined as the completely free flow field which is only influenced by wind of the outflow conditions (case 2) and the simulated hedge (case 4).
2.2.3.4 Conclusions and outlook

Simulations were set up to support A3 outdoor monobloc air-to-water heat pump development activities of interested manufacturers. The work is ongoing and will end up with several dozens of 3D-clouds. These data could either be used directly to interpret distribution / outflow effects or to set up experiments. Any gas warning sensor matrix could be arranged much better spatially based on these simulated findings. To solidify the reliability of these results validation work is planned.

2.2.4 References


2.2.5 Availability of components

2.2.5.1 The compressor

F-Gas compressors for heat pumps are already widely available. Shortages in getting approved compressors for specific F-Gases has almost ended. Newest patented blends, especially of the azeotropic R-5XX series, might have such shortages in near future.

However, the choice for relevant compressors is usually done in-between 4-6 manufacturers (in terms of economically meaningful specifications). More and more compressor manufacturers have designed their own R290 models, too, and introduced it to the market, see Figure 2-12 for two typical operation envelopes for a scroll as well as a twin-rotary compressor.

![Figure 2-12: Operation envelope (related to heat pump operation conditions) of R290 for available rotary and scroll compressor.](image)

2.2.5.2 Other components

Not discussed here.

2.2.6 Case Studies

Several R&D stakeholders with any relation in their field of interest in heat pumps were requested to participate with their findings to this task report. We requested and received findings based on new refrigerant-oil investigations for refrigerators and laundry dryers. Please find below a short summary on these activities from Fraunhofer ISE and these third parties.

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1 Meaningful excerpt from the manufacturer's approved operation envelope for scroll and rotary compressor for operating a heat pump.
Annex 54, Heat pump systems with low-GWP refrigerants

2.2.6.1 Fraunhofer ISE: Short review on Experimental Evaluation of a charge reduced Heat Pump Module using 150 g R290

2.2.6.1.1 Introduction

At Fraunhofer ISE between 2018 and 2020 a low refrigerant charge brine-to-water heat pump circuit was evaluated. The aim of the presented work is to create a heat pump using commercially available components only achieving 5 to 10 kW heating capacity with a maximum charge of 150 g propane. The following pages will show first results of a charge-reduced brine-to-water propane heat pump.

Propane is a natural refrigerant, has a low GWP (3) and attractive thermodynamic properties. Due to safety aspects the refrigerant charge was limited to 150 g, resulting in 0.45 kg of CO₂ equivalent.

2.2.6.1.2 Experimental Setup

The reduction of charge was achieved by minimized volumes of the internal components: the condenser and evaporator were chosen with an asymmetric plate profile, the liquid line was designed as short and thin as possible, the filter dryer in the liquid line was shifted to the suction line and the amount of compressor oil was reduced to its minimum. With dimensions of roughly 700x500x200 mm, the refrigeration circuit is small in comparison to other units with the same capacity. The following table gives an overview on the typical characteristics of the main components used in four different modules. The headline indicates the version number and the inner volume of the modules.

<table>
<thead>
<tr>
<th>Version no./ volume</th>
<th>V 1.0 [4.1 l]</th>
<th>V 2.5 [3.6 l]</th>
<th>V 2.6 [3.8 l]</th>
<th>V 3.0 [4.7 l]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Scroll</td>
<td>Rotary v1</td>
<td>Rotary v1</td>
<td>Rotary v1</td>
</tr>
<tr>
<td>Manufacturer</td>
<td>Manufacturer 1</td>
<td>Manufacturer 2</td>
<td>Manufacturer 2</td>
<td>Manufacturer 2</td>
</tr>
<tr>
<td>Condenser</td>
<td>Long</td>
<td>Long</td>
<td>Short</td>
<td>Short</td>
</tr>
<tr>
<td>Asymmetric</td>
<td>Asymmetric</td>
<td>Asymmetric</td>
<td>Asymmetric</td>
<td>Asymmetric</td>
</tr>
<tr>
<td>16 Plates</td>
<td>16 Plates</td>
<td>38 Plates</td>
<td>46 Plates</td>
<td>46 Plates</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Long</td>
<td>Long</td>
<td>Long</td>
<td>Long</td>
</tr>
<tr>
<td>Asymmetric</td>
<td>Asymmetric</td>
<td>Symmetric</td>
<td>Symmetric</td>
<td>Symmetric</td>
</tr>
<tr>
<td>16 Plates</td>
<td>16 Plates</td>
<td>16 Plates</td>
<td>16 Plates</td>
<td>28 Plates</td>
</tr>
<tr>
<td>Piping</td>
<td>Pipes v1</td>
<td>Pipes v1</td>
<td>Pipes v1</td>
<td>Pipes v2</td>
</tr>
</tbody>
</table>

The charge reduced heat pump circuit was tested at various points of operation. Most measurements were taken in a setup in which charge was varied. The temperatures for the secondary circuits were chosen based on typical test parameters in the heat pump sector. For the source temperature -10 °C, -7 °C, 0 °C, 12 °C were evaluated. These values refer to the inlet temperature of the evaporator. For the sink 35 °C, 45 °C, 55 °C and 65 °C were chosen as outlet temperatures of the condenser. For both of the secondary hydraulic circuits, a constant temperature difference between inlet and outlet of 5K and 3K respectively was maintained as specified in EN14511.
Figure 2-113: COP (left) and heating capacity (right) as a function of refrigerant charge for four different versions of heat pump modules. Results are shown for B0/W35 (brine at evaporator with an inlet temperature of 0 °C, water at the condenser with an inlet temperature of 35 °C), a compressor speed of 60 Hz and a superheat of 10 K.

The total range tested of the compressors was from 30 Hz to 120 Hz in 10 Hz increments. The suction superheat (SSH) has been set to 10 K; nevertheless, it was not reached for all operation points due to the limited operational window of the electronic expansion valve (EEV).

Increasing the compressor speed to 120 Hz results into an increased heating capacity of 8.4 kW at 150 g and 10 kW in maximum. The COP reaches 2.9 at 150 g and 3.5 in maximum.

Figures 2-13 and 2-14 show an extraction of the results for B0/W35 (brine at evaporator with an inlet temperature of 0 °C, water at the condenser with an inlet temperature of 35 °C), a compressor speed of 60 and 120 Hz and a superheat of 10 K. The left graph shows
the COP for the different versions, the right the heating capacity, both as a function of the refrigerant charge of the module.

For the mentioned conditions and 150 g of refrigerant charge a COP of 3.4 and a heating capacity of 4 are realized. The maximum of COP and heating capacity is 4.1 (COP) and 5 (heating capacity) at higher amounts of charge.

2.2.6.1.3 Discussion

The results show in general the feasibility of a heating capacity of 8 kW with 150 g of refrigerant charge. Additionally, the sensitivity of the charge reduced system and the strong influence of the components chosen becomes visible. Optical analyses indicate maldistribution of the refrigerant in the heat exchangers, which result

2.2.6.1.4 Outlook

The promising results are the starting point for the project “LC150” (10/2020 − 03/2023), funded by the German Ministry of Economic Affairs and Energy and European Heat Pump manufacturer. The main goal of this project is the development of a charge reduced heat pump module and an adapted operation strategy realizing a COP > 4 with a heating capacity > 8 kW. This will be worked out by a broad experimental campaign supported by simulations.

Component manufacturer supporting the project by suppling heat exchangers, compressors, etc... The heat pump manufacturer and scientific partners as UPV supervising the work.

2.2.6.2 RWTH Aachen: Systematic application of the decision-making process for the fluid selection of natural refrigerants in heat pumps

2.2.6.2.1 Short description

Vering [1] developed a methodology for the selection process of refrigerants. This multi-criteria process allows to include all refrigerants that might fit well to an appliance. Vering defines several scenarios and can use a GUI-based human interface to this tool. Integral decision processes were introduced. Vering concludes in his paper that hydrocarbons will play a greater role in future applications. The decision-making process to arrive at such a judgement is not disclosed in the presentation but might be obviously part of the implemented algorithms. However, a better understanding is possible considering his other work on this [2]. His next steps are to make such a tool more robust for decision-making processes. In a final step Vering wants to conduct this survey with different stakeholders to arrive at more accurate results.
Figure 2-115: Interdependency of stakeholder’s interests and their goals.

2.2.6.2.2 References


2.2.6.3 Technical University Braunschweig: Low-GWP Refrigerants for use in small commercial tumble dryers

2.2.6.3.1 Introduction

In household heat pump tumble dryers, the commonly used refrigerant R134a has a high global warming potential and is nowadays replaced by natural refrigerant R290. Commercial tumble dryers have a demand for short drying time and require high heating power and thus higher loads of refrigerant. Therefore, their environmental impact per dryer is even bigger compared to the household appliance. R290 and R744 are both valid alternatives for R134a in heat pump dryers [1] [2] with advantages like the low GWP but also some disadvantages. R290 has almost similar thermodynamic properties compared to R134a, but it is flammable and heat pump dryers with higher refrigerant loads exceeding the 150g limit require additional safety measures of electronic components and installation conditions [3]. R744 enables high temperatures inside the gas cooler due to the temperature glide and thus a short drying time but due to the different thermodynamic behavior the heat pump components must be completely redesigned [4]. To compare the thermodynamic potential of R290 and R744 with that of R134a, a simulation study has been carried out.
2.2.6.3.2 Setup

Based on a validated heat pump tumble dryer simulation model [5] some modifications have been made. The refrigerant side heat transfer coefficients of the evaporator, gas cooler and cooler were held constant in the simulations for all three refrigerants. The compressor model was exchanged by an efficiency-based model with constant volumetric (0.8) and isentropic efficiency (0.6) to have equal preconditions for every refrigerant. The compressor displacement was set to 6.75 cm² for R290 and R134a and 3.26 cm² for R744. The adjusted simulation model was used in a pareto optimization varying compressor speed, refrigerant mass, and expansion valve area by using the algorithm MOEA/D [6] and a starting population of fifty randomly distributed individuals. The objectives were both low energy consumption and drying time.

![Figure 2-116: Result of pareto optimization with objectives being energy consumption and drying time. Refrigerant R134a is compared with R290 and R744. For R744 to additional system topologies have been investigated: additional internal heat exchanger and combination of ejector, separator, and internal heat exchanger.](image)

As boundary conditions the compressor outlet temperature must not exceed 100°C during the whole drying cycle and a superheat below 1K was avoided all the time. The boundary conditions were implemented as a penalty function inside the simulation environment. The results of the pareto optimization with the three different refrigerants are shown in Figure 2-116. Each optimization case results in a pareto front comprising fifty pareto optimal solutions for the energy consumption and the drying time.

2.2.6.3.3 Discussion

As expected, short drying time goes along with a higher energy consumption and vice versa a lower energy consumption is only possible with slower drying speed. The differences between R134a and R290 at slow compressor speed are rather small. With R134a slightly less energy is consumed, but there is a turning point at higher compressor
speed where faster drying speed is only possible at the cost of a considerably higher energy consumption. With the same compressor displacement, a lower speed is needed to achieve the same drying velocity with R290 in comparison to R134a. That can be explained with the higher volumetric cooling capacity of R290. R744 again has an even higher volumetric cooling capacity and that’s why shorter drying times are more easily accessible with low compressor speed. But in comparison with R290 the energy consumption at equal drying time is bigger. One reason for that are the bigger throttling losses caused by the high-pressure difference. To account for this affect, the topology of the heat pump can be altered. Two different topologies were also optimized and compared to the other pareto fronts in Figure 2-16. With an internal heat exchanger between the evaporator and compressor on the low-pressure side and between the gas cooler and cooler on the high-pressure side the energy consumption can be reduced significantly. The difference between the pareto fronts of R290 and R744 gets smaller. A combination of ejector, separator and internal heat exchanger has additional advantages when the aim is to achieve a small drying time. Also, a lower energy consumption with the same compressor speed is possible.

2.2.6.3.4 Conclusions

In conclusion both refrigerants, R290 and R744, are possible replacements for R134a. R290 is favorable when a small energy consumption is desired, and the refrigerant limit of 150 g can be adhered. R744 has the advantage when there is demand for short drying times. But the heat pump topology must be adjusted to be able to compete with R290 efficiency wise.

2.2.6.3.5 References

2.2.6.4 Technical University Dresden: Work on lubricants for heat pump laundry dryers and refrigerators

2.2.6.4.1 Short introduction

Nosbers et al. [1] and Stöckel et al. [2] investigated a zeotropic mixture out of R-290 and R-E-170 (Dimethylether) with the focus on the determination of the experimental refrigerant-oil mixture behavior of this ternary system. At first the pure refrigerants were tested together with the lubricant. In Figure 2-17 and Figure 2-18 the test apparatus as well as the results of the pure refrigerants with the oil are presented.

![Figure 2-17: Apparatus for the measurement of the phase equilibrium thermodynamics.](image1)

![Figure 2-18: Measured values and values calculated via data fit for refrigerant mass fractions from 0 to 1 kg kg⁻¹ left: for POE + DME, right: for POE + propane](image2)

2.2.6.4.2 Conclusions

The measurement and calculation results indicate that the selected approach is a suitable tool for determining the solubility of individual refrigerants in oil. Here, the average deviation between the measured and theoretically calculated values of the exemplarily selected binary refrigerant-oil mixture with R290 is 5.85 %, of the binary mixture with RE170 4.10 %. A notable drawback of the presented equation is the necessary knowledge of the molar mass of the oil, which may not be known. This is especially true for mineral oil-based lubricants, which usually consist of many different hydrocarbon chains (napthenes). However, a consideration of the results for various assumed molar masses
of the measurements described in this manuscript shows that the influence of the molar mass is sufficiently small.

2.2.6.4.3 References


2.2.7 Conclusions

There are plenty of options on the one side for HFOs and on the other side for A3 refrigerants available. Due to the work in WG12 of the CEN/TC 182 in Europe certain barriers on the normative side are reduced for the usage of A3 refrigerants. Based on this and with a view on the demonstration as well as development activities in Germany (from manufacturers, R&D service providers, Universities) – as partially described above – the introduction of low-GWP and very low GWP refrigerants is on its way. Safety aspects as analyzed deeply for A2L refrigerants from several nations and researchers were intensified and will result into better background information when safety aspects for A3 systems are addressed. It is expected that more types and manufacturers will come up with A3 but also A2L solutions for their new developments.
3 Research and Development Activities on Low-GWP Refrigerants in Italy

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3.1 Summary

This report briefly summarizes the activities carried out during the 2nd year of Annex 54 in Italy. Due to the Covid-19 emergency, the research activities slightly slowed down both at research institutes and companies level. Despite this, advances in the use of low-GWP refrigerants in heat pumps have been made as detailed in the following sections. It should be noted that since the Italian team consists of four different research groups, the document is organised in four sections, one for each research group.

3.2 Activities at Daikin Applied Europe

Daikin is working on low-GWP refrigerants since 2014 introducing R32 (GWP 675) as a substitute of R410A (GWP 2,088) in household equipment. Daikin owned 93 patents covering R32 use in HVAC equipment and in 2015 he offered worldwide free access to all basic patents.

As a consequence of the owned basic knowledge on R32, the same choice has been made for low-end, low-specification, applied products: air-cooled chiller with scroll compressors up to 700 kW cooling capacity.

In 2018 cooling only R32 air-cooled chiller were launched in the market in the capacity range 80 - 700 kW; air-to-water heat pumps development started in 2019 and a relevant development has been performed in 2020.

3.2.1 Research activities on R32 Air to Water heat pumps in 2020

As for many worldwide companies, Daikin research activities in 2020 were strongly affected by Covid-19 pandemic, in particular several schedules were delayed as a consequence of local or national restrictions or lockdowns.

Nevertheless, many research activities were conducted and relevant results were reached. Two main research lines were conducted in 2020:
1. Air-to-refrigerant finned tube heat exchangers characterization.
2. Parallel operation of scroll compressors.

3.2.2 Air-to-refrigerant finned tube heat exchangers characterization

Air-to-refrigerant heat exchanger is one of the most critical component in an air-to-water heat pump; in fact, such heat exchanger has to work, alternatively, as an evaporator or as a condenser. In addition, it has to work in counter flow in one mode and in parallel flow in the other one.

Optimization of such component is a complex job and compromise has to be accepted; the first compromise to be accepted is that advanced technologies today widely adopted in air-to-refrigerant condensers, like mini or micro channel coils, are not yet reliable in the evaporation mode, either on refrigerant-side and air-side heat transfer.

Consequently, copper tubes and aluminum fins coils were investigated either in evaporation and condensation mode, as well as in parallel and counterflow.

Results show better performances of coils when operating with R32 versus R410A in any condition, In particular:
- R32 evaporating coil has 16% higher capacity in counterflow and 12% higher capacity in parallel flow with respect to R410A (Figure 0-1).
- R32 condensing coil has almost same capacity in counterflow while has 5% higher capacity in parallel flow with respect to R410A (Figure 0-2).
- R32 evaporating coil lose 11% capacity from counterflow to parallel flow while condensing coil lose 23% capacity.
A problem that had to be faced in evaporating coils is the distribution of two-phase flow among parallel circuits in the coil; bad distribution occurs when some circuit receives more liquid refrigerant than others, this creates a bad heat exchange since these circuits cannot evaporate all the liquid while other circuits mostly have vapor inside (Figure 0-3).

The solution has to be found looking for the best “circuiting” of coil tubes; this meaning, for a given number of tubes, looking for right number of tubes in series and in parallel. On the other hand, it is clear that number of circuits in the coil is driven by evaporating operation, being the condensing operation quite independent from this parameter.
3.2.3 Parallel operation of scroll compressors

Given the size of scroll compressors, typical operation in heat pumps is with two (tandem) or three (trio) compressors in parallel (Figure 0-4).

When oiled compressors (compressors where oil is injected in the refrigerant flow) like scrolls work in parallel, one relevant problem is to assure that oil leaving any compressor together with the refrigerant flow returns back to the same compressor, otherwise some compressor will suffer for oil lack while others are fully of oil with risk of “solid suction”.

The problem, known as “oil balancing”, was already faced and solved for cooling-only application but heat pump application requires a significant wider compressor envelope and cooling-only solutions are not reliable enough. In particular operation with very high pressure ratio is required when there is a cold environment and hot water has to be produced, also low pressure ratio is necessary in hot climate and medium temperature water. Situation is even more complex at partial load, when some compressor is on and others off; oil trend is to evacuate from stopped compressors to fill in the running ones.
Many test campaigns have been conducted to find the best configuration especially for trio, in particular the following aspects were investigated:

- Compressors size position in uneven combination (position of smallest and largest compressor with respect to other compressors and refrigerant flow direction).
- Piping layout (refrigerant flow direction, straight portions, bends radius, etc.).
- Flow regulation devices (diaphragms) in the piping aimed to create the desiderate pressure distribution among compressors suction.

In addition, activities have been conducted on the compressors sequencing logic to allow a correct operation rotation avoiding a compressor to stay too many time switched off or, for example, running alone.

### 3.3 Activities at National Research Council

The emergency caused by COVID-19 has strongly influenced the activities performed during 2020. In particular, the access to laboratory was very limited: thus, experimental research was almost blocked. The main activities developed at ITC-CNR in the last year can be summarized as follows.

#### 3.3.1 Activities within Go4Civic EU project on geothermal heat pumps

##### 3.3.1.1 Refrigerant selection

The general purpose of the project is to introduce significant technical innovations and solutions for each component necessary to install and exploit geothermal systems for building retrofits: drilling machine and methodology, ground source heat exchanger design and materials, compact and hybrid heat pumps for high and low temperature terminals.

In particular, ITC-CNR is involved in the selection of new low-GWP refrigerants as medium and long term substitutes for the present high GWP refrigerants used in the heat pumps (mainly R410A and R134a).

In the last year, new potential refrigerants have been included in the analysis and some new cycles have been considered. The following fluids have been considered at moment as potential medium term alternatives:

- **Substitutes for R134a (low pressure):** R513A, R1234yf, R1234ze(E), R515A, R515B, R516A, R1224yd(z), R1233zd(E).
- **Substitutes for R410A (high pressure):** R32, R452B, R454B.

Each fluid has been evaluated by simulating its performance in a series of heat pump cycles by means of a specific software developed in Matlab environment, with the support of REFPROP 10.0 database to calculate the thermodynamic properties.

The cycles considered are the following:

- **a)** Basic cycle.
- **b)** Regenerative cycle.
- **c)** Cycle with economizer at the compressor.
- **d)** Cycle with economizer.
- **e)** Cycle with vapor-liquid separator and auxiliary compressor (not for zeotropic mixtures).

Boundary temperatures for the secondary fluids at the geothermal source and the user sink have been assumed corresponding to the typical values for different zones in Europe.

The main results obtained are the following:

- R516A is the most promising substitute for R134a.
- R454B is the most promising substitute for R410A.
- Cycles c) and e) are those giving the best COP.
3.3.1.2 Heat pump monitoring at the CNR demo site

A series of sites have been selected to install demonstrative geothermal heat pumps with the aim to experimentally evaluate the feasibility and effectiveness of the innovative solutions developed within the project. In particular, at the demo site of CNR in Padova, two heat pumps will be installed: one with on/off control, the other with variable control.

Each heat pump and the secondary fluids circuits will be equipped with a monitoring system to evaluate their energetic performance along at least one year of acquisitions. In the last year, the monitoring system has been designed and the necessary components (temperature and pressure sensors, mass and volumetric flowrate measuring devices, acquisition system, software etc.) have been purchased. The installation of sensors and acquisition system is forecast in the beginning of the next year. Immediately after, the heat pumps and the monitoring will be started.

3.3.2 Thermophysical properties of new refrigerants for heat pump applications

Measurements of the saturated pressure and compressed liquid density for the HFO R1224yd(Z) have been performed, mostly before the COVID-19 emergency.

31 saturation pressures have been measured by means of a static vapor–liquid equilibrium (VLE) apparatus in the range of temperatures between 293.15 to 353.15 K.

At the same time, 90 compressed density liquid data have been obtained using a stainless-steel vibrating tube density-meter (Anton Paar DMA 512) in the temperature range between 283 K and 363 K and pressures up to 35 MPa.

3.4 Activities at University of Padua

The research activity performed at the Department of Industrial Engineering (University of Padova) during 2020 has been focused on heat transfer measurements with low-GWP mixtures and on the experimental investigation of a solar assisted heat pump working with CO₂.

3.4.1 Condensation and flow boiling of low-GWP non-azeotrope mixtures

Refrigerants currently employed in heat pumps and in air-conditioning systems (e.g. R410A) cannot be easily substituted with low-GWP pure fluids. Most of the proposed substitutes are binary or ternary mixtures (made of HFCs and HFOs): they usually include R32 and a halogenated olefin like R1234yf or R1234ze(E). In many cases, when the working fluid is going to be replaced, some of the components of the system must be modified and a new design of the heat exchangers must be considered. As a general trend, the diameter of pipes used in heat exchangers is going down. For example, in finned-tubes coils heat exchangers, diameters around 5 mm are often employed. Minichannels heat exchangers (with internal diameter around 1-2 mm) are a common solution for the automotive sector and they are also used in air-cooled chillers. Therefore, it is important to extend available databases encompassing conditions that include HFCs/HFOs mixtures and small diameter channels. These data can then be used for the assessment of heat transfer correlations for heat exchangers design.

At the University of Padova, heat transfer coefficients have been measured during condensation and flow boiling inside a 0.96 mm diameter channel (Figure 0-5). A binary mixture made of R32 and olefin R1234ze(E), 0.748/0.251 by mass composition, has been considered in the study. This blend has been selected because it presents a GWP₁₀₀-years around 500 and it can be considered a possible substitute of R410A.

Condensation heat transfer coefficients have been measured during condensation and flow boiling inside a 0.96 mm diameter channel (Figure 0-5). A binary mixture made of R32 and olefin R1234ze(E), 0.748/0.251 by mass composition, has been considered in the study. This blend has been selected because it presents a GWP₁₀₀-years around 500 and it can be considered a possible substitute of R410A.

Condensation heat transfer coefficients have been measured at saturation pressure of 21.8 bar (corresponding to a dew point temperature of 41.5 °C) with mass flux ranging between 150 kg m⁻² s⁻¹ and 800 kg m⁻² s⁻¹. The condensation heat transfer coefficients for the blend are lower than the ones calculated with a linear interpolation from the values pertaining to the pure fluids (for mass velocity G = 400 kg m⁻² s⁻¹, the penalization is equal to 10.1% at vapor quality equal to 0.6). The deviation from the ideal linear behavior can be explained considering the mass transfer
resistance. A comparison between the condensation performance of the blend and its pure fluids R32 and R1234ze(E) has been done considering exergy losses. In these conditions, the heat transfer coefficient of the 75/25% mixture is on average 32.8% lower than that of R32 and 91.9% higher than that of pure R1234ze(E).

Flow boiling tests have been run at 17 bar (corresponding to a bubble temperature of 28.7 °C), mass velocity between 300 and 600 kg m$^{-2}$ s$^{-1}$, and heat flux between 30 and 245 kW m$^{-2}$. Considering flow boiling tests, the heat transfer coefficient increases with the heat flux and to a less extend with mass velocity. The heat transfer coefficient decreases with the vapor quality for all the values of mass velocity. A degradation of about 30% of the heat transfer coefficient for the blend with respect to a ideal linear behavior from the values pertaining to the pure fluids has been observed. This penalization is due to the mass transfer resistance.

Figure 0-5: Experimental test rig for the measurement of the condensation and vaporization heat transfer coefficient inside minichannels.

Most of the studies available in the literature focus on flow boiling of binary mixtures of HFCs and HFOs while works on multi-component blends are rare. Heat transfer coefficients have been measured during flow boiling of ternary non-azeotropic mixtures inside two horizontal smooth tubes of 8.0 mm and 0.96 mm inner diameter [2, 3]. The experimental tests are performed with mixtures R455A (R32/R1234yf/R744 at 21.5/75.5/3% by mass) and R452B (R32/R1234yf/R125 at 67/26/7% by mass), displaying respectively a temperature glide of about 11 K and 1 K in the present tests. R452B and R455A can be considered substitutes respectively of R410A and R404A. The effects of vapour quality, saturation pressure, heat flux, mass velocity and channel diameter on the heat transfer coefficient of the mixtures have been investigated.

At the same operating conditions, R452B displays higher flow boiling heat transfer coefficients compared to R455A. The reduced heat transfer performance in the case of R455A is due to fluid properties and to the additional mass transfer resistance related to the temperature glide. In the case of the 8.0 mm diameter channel, the heat transfer coefficient of the two blends increases with heat flux, mass velocity and vapour quality. In the 0.96 mm diameter channel, R455A flow boiling heat transfer coefficient increases with the heat flux while it is less sensitive to mass velocity. The heat transfer coefficients in the 0.96 mm channel are higher than those measured in the 8.0 mm diameter channel in the low quality region. Models developed for pure fluids, which generally overestimate the heat transfer performance of the blends and do not account for the penalization due to the mass diffusion effects, must be corrected to take in consideration the additional mass transfer resistance.
3.4.2 Solar assisted heat pump working with CO₂

Solar Assisted Heat Pumps (SAHPs) consist in heat pump systems that work with the solar source as low temperature thermal source. There are two types of SAHPs: indirect solar assisted heat pumps (IDX-SAHPs) where a secondary fluid is heated up in solar collector and then it is sent to the evaporator, and direct solar assisted heat pumps (DX-SAHPs), where the solar collector acts itself as the evaporator. The use of hybrid photovoltaic-thermal (PV-T) solar collectors in SAHPs has the advantage to provide additional electrical power to the system with a higher PV conversion efficiency due to the cooling of the cells. When using SAHPs, the evaporation temperature can be higher than in the case of air-to-refrigerant evaporators and thus the instantaneous COP turns out to be higher. Considering the recent international regulations call for a reduction of the GWP of employed refrigerants, the use of natural fluids such as CO₂ is increasingly growing in heat pump systems. At the Department of Industrial Engineering, a novel innovative direct expansion solar assisted heat pump prototype working with CO₂ as refrigerant has been installed. The study has been realized in the framework of Solair-HP CSEA project. The prototype is a 4 kW heat pump that can produce hot water. The heat pump can work alternatively with a finned coil evaporator or with a PV-T solar evaporator. The solar evaporator is a sheet-and-tube heat exchanger placed in thermal contact with the PV module and it allows to evaporate the CO₂ while cooling the PV cells (Figure 0-6). Therefore, the PV-T collector has a double effect: it allows to evaporate the CO₂ that flows in the tubes and to cool down the photovoltaic cells improving the PV conversion efficiency. The main objective of this system is to increase the seasonal coefficient of performance of the heat pump compared to an air source heat pump and reduce the overall electrical consumption by including the PV modules.

Figure 0-6: Solar assisted heat pump prototype installed at the Solar Energy Conversion Laboratory (Department of Industrial Engineering, University of Padova).

3.5 Activities at Polytechnica of Milan

During 2020, the research activities carried out at Polytechnic of Milan focused on the experimental comparison of the performance of a water-to-water heat pump that uses low-GWP refrigerants alternative to R134a.

An experimental set-up that mimics a water-to-water heat pump is available for studying refrigerant alternatives in drop-in application. The layout of the experimental set-up is shown in Figure 0-7.
The test rig consists of three different loops: the refrigerant loop, the cold water + ethylene glycol loop (evaporator loop) and the hot water loop (condenser loop). The main characteristics of the components in the refrigerant loop are reported in Table 0-1.

Table 0-1: 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>
<tr>
<td></td>
<td>Oil</td>
<td>POE ISO 32</td>
</tr>
<tr>
<td></td>
<td>Oil charge</td>
<td>1.1 dm³</td>
</tr>
<tr>
<td>Condenser</td>
<td>Height x Width x Depth</td>
<td>289 mm x 119 mm x 93.6 mm</td>
</tr>
<tr>
<td></td>
<td>Number of plates</td>
<td>40</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Height x Width x Depth</td>
<td>376 mm x 119 mm x 71.2 mm</td>
</tr>
<tr>
<td></td>
<td>Number of plates</td>
<td>30</td>
</tr>
<tr>
<td>Expansion valve</td>
<td>Capacity range</td>
<td>1,200 W – 12,000 W</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1,690 W – 16,900 W</td>
</tr>
<tr>
<td>Liquid receiver</td>
<td>Volume</td>
<td>2.8 dm³</td>
</tr>
<tr>
<td>Suction accumulator</td>
<td>Volume</td>
<td>2.33 dm³</td>
</tr>
<tr>
<td>Oil separator</td>
<td>Type</td>
<td>Coalescence</td>
</tr>
<tr>
<td></td>
<td>Volume</td>
<td>2.8 dm³</td>
</tr>
<tr>
<td>Pumps</td>
<td>Nominal flow rate</td>
<td>28.7 m³/h</td>
</tr>
<tr>
<td></td>
<td>Nominal head</td>
<td>160 kPa</td>
</tr>
<tr>
<td></td>
<td>Shaft rotational frequency</td>
<td>16 Hz - 58 Hz</td>
</tr>
<tr>
<td>Recuperator</td>
<td>Height x Width x Depth</td>
<td>193 mm x 76 mm x 71.2 mm</td>
</tr>
<tr>
<td></td>
<td>Number of plates</td>
<td>30</td>
</tr>
</tbody>
</table>

The experimental set-up is equipped with instrumentations allowing for the measurement, acquisition and storing of the main parameters such as pressures, temperatures, flow rates and power. Their main feature are reported in...
Annex 54, Heat pump systems with low-GWP refrigerants

Table 0-2.
Table 0-2: 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 - 700 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 - 4,000 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 test rig is used to compare the drop-in performance of the use of R134a, R1234yf, R1234ze(E), R450A and R513A under the testing conditions reported in Table 0-3. Tests 1-5 are carried out in pure drop-in conditions, whereas tests 6-10 are carried out with the aim of identifying the rotational frequency of the compressor shaft that leads to the same heating capacity measured with R134a.

Table 0-3: Test Conditions

<table>
<thead>
<tr>
<th>Run</th>
<th>Frequency</th>
<th>Superheating</th>
<th>Evaporator</th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_OUT</td>
<td>ΔT</td>
</tr>
<tr>
<td>1</td>
<td>50</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>2</td>
<td>50</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>3</td>
<td>50</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>4</td>
<td>50</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>5</td>
<td>50</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>6</td>
<td>Identified</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>7</td>
<td>Identified</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>8</td>
<td>Identified</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>9</td>
<td>Identified</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
<tr>
<td>10</td>
<td>Identified</td>
<td>5 K</td>
<td>5 ºC</td>
<td>5 K</td>
</tr>
</tbody>
</table>

The heating capacity and the COP of the heat pump measured in drop-in conditions, i.e. under runs 1-5 conditions, are shown in Figure 0-8 and Figure 0-9 respectively. Starting from the heating capacity, it is possible to state that the use of any low-GWP alternative to R134a leads to a reduction of the heat pump heating capacity. The capacity reduction with R1234yf, R450A or R513A is quite little since it lies in the range 95%-99% whereas a larger capacity reduction is achieved with R1234ze(E) since it is in the range 77%-79%. A similar trend is found with the COP since the use of any alternative refrigerants lead to a COP reduction. The COP lies in the range 93%-99% when R1234yf or R513A are considered while it is closer to R134a, in the range 98%-100%, with R1234ze(E) or R450A. Overall, the following trend is found: the closer to R134a is the heating capacity and the lower is the COP and vice-versa.
Annex 54, Heat pump systems with low-GWP refrigerants

Figure 0-8: Heat pump heating capacity as a function of condenser outlet temperature for the five refrigerants considered under run 1-5 conditions.

Figure 0-9: Heat pump COP as a function of condenser outlet temperature for the five refrigerants considered under run 1-5 conditions.

The rotational frequency of the compressor shaft and the COP of the heat pump measured under constant heating capacity conditions, i.e. under runs 6-10 conditions, are shown in Figure 0-10 and Figure 0-11, respectively. Discussing the data about the rotational frequency of the compressor shaft first, in order to let the heat pump to supply the same R134a heating capacity when a low-GWP alternative is used, an increase in the rotational frequency of the compressor shaft is needed. The refrigerant that exhibits the lowest heating capacity at 50 Hz, i.e. R1234ze(E), requires the largest frequency increase, in the range 10%-50%. Conversely, the frequency increase is in the range 2%-17% with all the other refrigerants since their heating capacity is more...
similar to that of R134a. The increase in shaft rotational frequency leads also to a reduction on the heat pump COP. The COP is within 93%-98% when R1234yf, R450A or R513A are considered whereas it lies in the range 82%-86% with the refrigerant R1234ze(E).

Overall, the following trend is found: the higher is the rotational frequency of the compressor shaft, the lower is the heat pump COP and vice-versa.

Figure 0-10: Rotational frequency of the compressor shaft as a function of condenser outlet temperature for the five refrigerants considered under run 6-10 conditions.

Figure 0-11: Heat pump COP as a function of condenser outlet temperature for the five refrigerants considered under run 6-10 conditions

3.6 Acknowledgment

The support of Italian Ministry for Education and Research (MIUR) through the project PRIN 2015 (Grant Number 2015M8S2PA) is acknowledged.
3.7 References


Experimental Evaluation of R410A, R407C and R134a Alternative Refrigerants in Residential Heat Pumps

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4.1 Summary

Alternative refrigerants with low-GWP are under investigation for residential heat pumps, air-conditioners and heat pump water heaters, since R410A, R407C and R134a have GWP of 2088, 1650 and 1430, respectively. In this study, five alternative refrigerants: R459A, R454B, R447A, HPR2A and R32 were investigated for the replacement of R410A in a 10 kW Air-to-Water (A/W) reversible heat pump. Three alternative refrigerants for R134a: R1234yf, R513A and R450A were tested in a split Heat Pump Water Heater (HPWH) having a water tank of 200 liters. R454C and R455A were evaluated as a possible alternative to R407C in a 3 kW Water-to-Air (W/A) reversible heat pump. A total of 10 alternative refrigerants with low-GWP were evaluated with not less than 130 performance tests. These experimental results will be useful for the HVAC community for facilitating the selection of the most promising candidates for replacement of R410A, R134a and R407C in residential heat pumps.

4.2 Introduction

Protocols and regulations such as the Montreal Protocol (1987), the Kyoto Protocol (1997), the European F-gas regulation (2006 revised 2014) cause a shift toward refrigerants with both zero Ozone Depletion Potential (ODP) and low Global Warming Potential (GWP) [01]. These new limitations lead to the progressive phase-out of HFC and to their replacement by the 4th generation of refrigerants based on HFO mixtures.

Alternative refrigerants with low-GWP are under investigation for residential heat pumps, air-conditioners and heat pump water heaters, since R410A, R407C and R134a have GWP of 2088, 1650 and 1430, respectively. These investigations are numerical ([1], [2]) or experimental ([3], [4], [5], [6]).

The objective of this work is to assess and to compare the heat pump performance when drop-in tests are carried with:

- 5 alternative refrigerants to R410A in a 10 kW air-to-water reversible heat pump: R459A, R454B, R447A, HPR2A and R32, with GWP of 460, 466, 583, 600 and 675, respectively;
- 2 alternative refrigerants to R407C in a 3 kW water-to-air reversible heat pump: R454C and R455A with a GWP of 148 and 146;
- 3 alternative refrigerants to R134a in a split Heat Pump Water Heater (HPWH) having a water tank of 200 liters: R1234yf, R513A, R450A with GWP of 4, 631 and 604, respectively.

The choice of the alternative refrigerants is based on the result analysis of the AHRI Low-GWP AREP Program ([3], [4], [7]).

This work is divided in three parts:

- experimental evaluation of R410A alternative refrigerants in an air-to-water reversible heat pump;
- experimental evaluation of R407C alternative refrigerant in a water-to-air reversible heat pump;
- experimental evaluation of R134a alternative refrigerants in a split water heater heat pump.

For each part, the refrigerant properties are presented, then the experimental procedure is described, and finally the experimental results are reported and analyzed.

In this study, only performance tests were carried out, the use of these alternative refrigerants will require a complete study of the risks, sizing and compatibility.
4.3 Experimental evaluation of R410A alternative refrigerants in an Air-to-water reversible heat pump

4.3.1 Properties of alternative refrigerants to R410A

Table 4-1 presents the main properties of the refrigerants studied to replace R410A. The data source is the software NIST REFPROP Version 10 [8]. Figure 4-1 shows the refrigerant's GWP and safety class.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Composition</th>
<th>GWP_{100}</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 wt.%)</td>
<td>2088</td>
<td>70.2</td>
<td>-51.6</td>
<td>0.1</td>
<td>A1</td>
</tr>
<tr>
<td>R32</td>
<td>R32 (100 wt.%)</td>
<td>675</td>
<td>78.0</td>
<td>-52.0</td>
<td>0</td>
<td>A2L</td>
</tr>
<tr>
<td>HPR2A</td>
<td>R32/R134a/R1234ze (76/6/18 wt.%)</td>
<td>600</td>
<td>82.0</td>
<td>-50.7</td>
<td>4.1</td>
<td>A2L</td>
</tr>
<tr>
<td>R447A</td>
<td>R32/R1234ze(E)/R125 (68/28.5/3.5 wt.%)</td>
<td>583</td>
<td>80.2</td>
<td>-47.6</td>
<td>5.1</td>
<td>A2L</td>
</tr>
<tr>
<td>R454B</td>
<td>R32/R1234yf (68.9/31.1 wt.%)</td>
<td>466</td>
<td>78.1</td>
<td>-50.4</td>
<td>1.3</td>
<td>A2L</td>
</tr>
<tr>
<td>R459A</td>
<td>R32/R1234yf/R1234ze (68/26/6 wt.%)</td>
<td>460</td>
<td>76.5</td>
<td>-49.5</td>
<td>1.9</td>
<td>A2L</td>
</tr>
</tbody>
</table>

Figure 4-1: Refrigerant's GWP and safety class

Alternative refrigerants have a lower GWP than R410A, between -67 % and -78 %. With the exception of R32, which is a pure refrigerant, the other alternatives are mixtures and mainly composed of R32 (~70 wt.%) and of a HFO (~30 wt.%), R1234ze(E) or R1234yf. The safety class of alternative refrigerants is A2L, which means they have a low flammability and are non-toxic. All alternative mixtures have a glide. The saturation properties (pressure-temperature) are shown in Figure 4-2. The five refrigerants have equivalent saturation properties. R447A, R454B, R459A and HPR2A remain slightly less volatile than R410A and R32.
4.3.2 Experimental investigation

Drop-in tests were carried out to assess the heat pump performance. The heating capacity of the tested air-to-water heat pump is close to 10 kW at H1 rating condition. It is a reversible, packaged and non-ducted appliance. The heat pump is equipped with a fixed capacity scroll compressor and a calibrated orifice as expansion device. The initial charge of R410A is 2.35 kg.

The test conditions in cooling mode and in heating mode are described in Table 4-2 and Table 4-3, respectively. For each refrigerant, a charge optimization was done at the C1 rating condition, then the rating and operating limit condition tests were performed, and finally, performance verification with R410A was carried out on C1 and H1 rating conditions to detect any anomaly after the use of the alternatives. The tests were carried out in one of CETIAT climatic rooms according to EN 14511 standard [11]. During tests, measurements allowed the determination of thermal capacities, electric energy consumptions, efficiencies (EER or COP), as well as pressures and temperatures on the refrigerant circuit.

According to the uncertainty of measurement on the laboratory’s instrumentation, capacities were determined with a maximal uncertainty of 5 % and electric energy consumptions with a maximal uncertainty of 1 %.

Table 4-2: Rating (C) and operating limit conditions (CL) in cooling mode

<table>
<thead>
<tr>
<th></th>
<th>Air temperature (°C)</th>
<th>Inlet water temperature (°C)</th>
<th>Outlet water temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>35</td>
<td>12</td>
<td>7</td>
</tr>
<tr>
<td>C2</td>
<td>35</td>
<td>23</td>
<td>18</td>
</tr>
<tr>
<td>CL1</td>
<td>18</td>
<td>*</td>
<td>5</td>
</tr>
<tr>
<td>CL2</td>
<td>42</td>
<td>*</td>
<td>25</td>
</tr>
</tbody>
</table>

* Inlet water temperature obtained with the C1 water flow rate.
Table 4-3: Rating (H) and operating limit conditions (HL) in heating mode

<table>
<thead>
<tr>
<th></th>
<th>Dry air temperature (wet bulb) (°C)</th>
<th>Inlet water temperature (°C)</th>
<th>Outlet water temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>7(6)</td>
<td>30</td>
<td>35</td>
</tr>
<tr>
<td>H2</td>
<td>7(6)</td>
<td>40</td>
<td>45</td>
</tr>
<tr>
<td>H3</td>
<td>7(6)</td>
<td>47</td>
<td>55</td>
</tr>
<tr>
<td>H4</td>
<td>-7(8)</td>
<td>*</td>
<td>35</td>
</tr>
<tr>
<td>H5</td>
<td>2(1)</td>
<td>*</td>
<td>35</td>
</tr>
<tr>
<td>H6</td>
<td>12(11)</td>
<td>*</td>
<td>35</td>
</tr>
<tr>
<td>HL1</td>
<td>-15</td>
<td>*</td>
<td>22</td>
</tr>
<tr>
<td>HL2</td>
<td>-10</td>
<td>*</td>
<td>42.5</td>
</tr>
<tr>
<td>HL3</td>
<td>24 (20)</td>
<td>*</td>
<td>54.8</td>
</tr>
</tbody>
</table>

* Inlet water temperature obtained with the H1 water flow rate.

The operating limit conditions were fixed by the heat pump manufacturer: they correspond to the boundary conditions of operation of the heat pump with R410A. During the test, the discharge temperature was limited to 115 °C to avoid any damage to the compressor.

4.3.3 Results of the experimental evaluation of R410A alternative refrigerants

4.3.3.1 Charge optimization

To perform the charge optimization, the initial alternative refrigerant charge was about 1.65 kg (corresponding to 70% of the initial R410A charge). At C1 rating condition (see Table 4-2), refrigerant was added (+50 g every 30 minutes) while four parameters were monitored: EER, cooling capacity, superheating and subcooling. The objective was to identify the performance curve inflexion point to determine the optimal charge. Particular attention was paid to the fact that superheating and subcooling have to be comprised between 4 and 7 K. The optimal charges obtained are reported in Table 4-4.

Table 4-4: Charge optimization results (at C1 rating condition)

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R410A (baseline)</th>
<th>R32</th>
<th>R454B</th>
<th>R459A</th>
<th>HPR2A</th>
<th>R447A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charge (kg)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R410A</td>
<td>2.35</td>
<td>1.52</td>
<td>2.00</td>
<td>1.96</td>
<td>1.80</td>
<td>1.86</td>
</tr>
<tr>
<td></td>
<td>(-35.1 %)</td>
<td>(-14.9 %)</td>
<td>(-16.5 %)</td>
<td>(-23.4 %)</td>
<td>(-20.8 %)</td>
<td></td>
</tr>
<tr>
<td>Cooling capacity (kW)</td>
<td>8.01</td>
<td>8.63</td>
<td>8.25</td>
<td>7.93</td>
<td>7.75</td>
<td>7.44</td>
</tr>
<tr>
<td></td>
<td>(+7.7 %)</td>
<td>(+3.0 %)</td>
<td>(+1.0 %)</td>
<td>(+3.2 %)</td>
<td>(+7.1 %)</td>
<td></td>
</tr>
<tr>
<td>EER (-)</td>
<td>2.71</td>
<td>2.83</td>
<td>2.97</td>
<td>2.90</td>
<td>2.90</td>
<td>2.83</td>
</tr>
<tr>
<td></td>
<td>(+4.4 %)</td>
<td>(+9.6 %)</td>
<td>(+7.1 %)</td>
<td>(+7.0 %)</td>
<td>(+4.4 %)</td>
<td></td>
</tr>
<tr>
<td>Superheating (K)</td>
<td>9.40</td>
<td>8.10</td>
<td>4.10</td>
<td>4.30</td>
<td>4.30</td>
<td>4.10</td>
</tr>
<tr>
<td></td>
<td>(-1.3 K)</td>
<td>(-5.3 K)</td>
<td>(-5.1 K)</td>
<td>(-5.1 K)</td>
<td>(-5.3 K)</td>
<td></td>
</tr>
<tr>
<td>Subcooling (K)</td>
<td>6.10</td>
<td>1.20</td>
<td>4.50</td>
<td>5.60</td>
<td>2.00</td>
<td>3.70</td>
</tr>
<tr>
<td></td>
<td>(-4.9 K)</td>
<td>(-1.5 K)</td>
<td>(-0.5 K)</td>
<td>(-4.1 K)</td>
<td>(-2.4 K)</td>
<td></td>
</tr>
</tbody>
</table>

Alternative refrigerant charges are lower (-35 % to -15 %) than with R410A. These results are consistent with the literature ([3], [5], [9], [10]).

4.3.3.2 Cooling mode

Figure 4-3 presents the results obtained in cooling mode: ratios of performance (alternative/R410A) and discharge temperature. Table 4-5 and Table 4-6 provide values for the heat pump cooling capacity and EER, respectively.
Table 4-5: Cooling capacity (green color highlights best performance)

<table>
<thead>
<tr>
<th>Cooling capacity (kW)</th>
<th>R410A (base)</th>
<th>R32</th>
<th>R454B</th>
<th>R459A</th>
<th>HPR2A</th>
<th>R447A</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1 (A35/W12-7)</td>
<td>8.01</td>
<td>8.83 (107.7 %)</td>
<td>8.25 (103.0 %)</td>
<td>7.93 (99.0 %)</td>
<td>7.75 (96.7 %)</td>
<td>7.44 (92.9 %)</td>
</tr>
<tr>
<td>C2 (A35/W23-18)</td>
<td>8.80</td>
<td>9.88 (112.3 %)</td>
<td>9.29 (105.5 %)</td>
<td>9.00 (102.3 %)</td>
<td>8.90 (101.1 %)</td>
<td>8.54 (97.0 %)</td>
</tr>
<tr>
<td>CL1 (A18/W*-5)</td>
<td>9.30</td>
<td>9.09 (97.8 %)</td>
<td>8.22 (88.4 %)</td>
<td>8.44 (90.7 %)</td>
<td>7.97 (85.7 %)</td>
<td>7.65 (82.3 %)</td>
</tr>
<tr>
<td>CL2 (A42/W*-25)</td>
<td>8.87</td>
<td>8.32 (105.1 %)</td>
<td>9.11 (102.7 %)</td>
<td>9.06 (102.1 %)</td>
<td>8.78 (99.0 %)</td>
<td></td>
</tr>
</tbody>
</table>

Table 4-6: EER (green color highlights best performance)

<table>
<thead>
<tr>
<th>EER (-) Ratio (alternative/base)</th>
<th>R410A (base)</th>
<th>R32</th>
<th>R454B</th>
<th>R459A</th>
<th>HPR2A</th>
<th>R447A</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1 (A35/W12-7)</td>
<td>2.71</td>
<td>2.83 (104.5 %)</td>
<td>2.97 (109.6 %)</td>
<td>2.90 (107.1 %)</td>
<td>2.90 (107.2 %)</td>
<td>2.83 (104.4 %)</td>
</tr>
<tr>
<td>C2 (A35/W23-18)</td>
<td>2.93</td>
<td>3.16 (107.8 %)</td>
<td>3.29 (112.1 %)</td>
<td>3.24 (110.3 %)</td>
<td>3.27 (111.5 %)</td>
<td>3.20 (109.1 %)</td>
</tr>
<tr>
<td>CL1 (A18/W*-5)</td>
<td>3.78</td>
<td>3.54 (93.6 %)</td>
<td>3.62 (95.9 %)</td>
<td>3.51 (93.0 %)</td>
<td>3.35 (88.7 %)</td>
<td>3.32 (87.8 %)</td>
</tr>
<tr>
<td>CL2 (A42/W*-25)</td>
<td>2.53</td>
<td>2.83 (111.9 %)</td>
<td>2.79 (110.1 %)</td>
<td>2.85 (112.7 %)</td>
<td>2.89 (114.1 %)</td>
<td></td>
</tr>
</tbody>
</table>
With the exception of the CL1 limit condition, the alternative refrigerants show higher performance than R410A. The cooling capacities at C1, C2 and CL2 conditions are increased with R454B (+3 % to +5.5 %), equivalent or even higher with R459A (-1 % to +2.7 %), equivalent and lower with HPR2A (-3.3 % to -2.1 %) and lower with R447A (-7.1 % to -1 %). R32 leads to higher capacities (+7.7 % to +12.3 %) than R410A at C1 and C2 conditions.

All refrigerants show cooling capacities lower than those with R410A at CL1 limit condition, from -17.7 % with R447A to -2.2 % with R32. EER are better with alternative refrigerants at conditions C1, C2 and CL2 (+4.4 % to + 14.1 %). For CL1 limit condition, all refrigerants give lower EER (-12.2 % to -4.1 %) than R410A. With the exception of CL1, alternative refrigerants achieve equivalent or even better performance than R410A.

The discharge temperatures observed for alternative refrigerants (except R32) in cooling mode are close to those with R410A. R32 did not allow performing CL2 limit condition test because the discharge temperature was higher than 115 °C. To reach a temperature below 115 °C, the outlet water temperature was set to 14 °C.

### 4.3.3.3 Heating mode

Figure 4-4 presents the results obtained in heating mode: ratios of performance (alternative/R410A) and discharge temperature. Table 4-7 and Table 4-8 give values of the heat pump heating capacity and COP, respectively.

![Figure 4-4: Heat pump performance in heating mode: a) Capacity ratio; b) COP ratio; c) Discharge temperature](image)
Table 4-7: Heating capacity (green color highlights best performance)

<table>
<thead>
<tr>
<th>Heating capacity (KW) (ratio alternative/base)</th>
<th>R410A (base)</th>
<th>R32</th>
<th>R454B</th>
<th>R459A</th>
<th>HPR2A</th>
<th>R447A</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1 (A7(6)/W30-35)</td>
<td>10.10</td>
<td>7.06 (69.9 %)</td>
<td>9.60 (95.0 %)</td>
<td>9.55 (94.6 %)</td>
<td>6.73 (66.6 %)</td>
<td>6.99 (69.2 %)</td>
</tr>
<tr>
<td>H2 (A7(6)/W40-45)</td>
<td>10.44</td>
<td>9.81 (94.0 %)</td>
<td>9.63 (92.3 %)</td>
<td>9.51 (91.1 %)</td>
<td>8.94 (85.7 %)</td>
<td>9.03 (86.5 %)</td>
</tr>
<tr>
<td>H3 (A7(6)/W47-55)</td>
<td>9.95</td>
<td><strong>Discharge T &gt; 115 °C</strong></td>
<td>9.38 (94.3 %)</td>
<td>9.26 (93.1 %)</td>
<td>8.98 (90.2 %)</td>
<td>8.71 (87.6 %)</td>
</tr>
<tr>
<td>H4 (A-7(-8)/W*-35)</td>
<td>4.33</td>
<td>4.95 (114.4 %)</td>
<td>4.40 (101.7 %)</td>
<td>4.20 (97.0 %)</td>
<td>4.37 (101.1 %)</td>
<td>4.12 (95.3 %)</td>
</tr>
<tr>
<td>H5 (A2(1)/W*-35)</td>
<td>6.21</td>
<td>6.23 (100.4 %)</td>
<td><strong>6.63 (106.8 %)</strong></td>
<td>5.68 (91.6 %)</td>
<td>5.54 (89.3 %)</td>
<td>5.43 (87.5 %)</td>
</tr>
<tr>
<td>H6 (A12(11)/W*-35)</td>
<td>10.86</td>
<td>7.36 (67.7 %)</td>
<td>10.43 (96.0 %)</td>
<td>10.34 (95.2 %)</td>
<td>9.57 (86.3 %)</td>
<td>9.64 (88.7 %)</td>
</tr>
<tr>
<td>HL1 (A15/W*-22)</td>
<td>3.50</td>
<td>3.95 (113.0 %)</td>
<td>3.23 (92.5 %)</td>
<td>3.35 (95.9 %)</td>
<td>3.52 (100.6 %)</td>
<td>3.34 (95.6 %)</td>
</tr>
<tr>
<td>HL2 (A10/W*-42.5)</td>
<td>3.71</td>
<td>4.21 (113.6 %)</td>
<td>3.91 (105.6 %)</td>
<td>3.67 (99.1 %)</td>
<td>3.76 (101.5 %)</td>
<td>3.62 (97.6 %)</td>
</tr>
<tr>
<td>HL3 (A24(20)/W*-54.8)</td>
<td>12.33</td>
<td><strong>Discharge T &gt; 115 °C</strong></td>
<td>11.75 (95.3 %)</td>
<td>11.71 (95.0 %)</td>
<td>11.06 (89.7 %)</td>
<td>11.18 (90.7 %)</td>
</tr>
</tbody>
</table>

Table 4-8: COP (green color highlights best performance)

<table>
<thead>
<tr>
<th>COP (-) (ratio alternative/base)</th>
<th>R410A (base)</th>
<th>R32</th>
<th>R454B</th>
<th>R459A</th>
<th>HPR2A</th>
<th>R447A</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1 (A7(6)/W30-35)</td>
<td>3.88</td>
<td>3.47 (89.6 %)</td>
<td>4.07 (104.0 %)</td>
<td><strong>4.07 (105.1 %)</strong></td>
<td>3.65 (94.2 %)</td>
<td>3.73 (96.2 %)</td>
</tr>
<tr>
<td>H2 (A7(6)/W40-45)</td>
<td>3.30</td>
<td>3.11 (94.3 %)</td>
<td><strong>3.51 (100.5 %)</strong></td>
<td>3.30 (100.2 %)</td>
<td>3.19 (96.8 %)</td>
<td>3.28 (99.7 %)</td>
</tr>
<tr>
<td>H3 (A7(6)/W47-55)</td>
<td>2.67</td>
<td><strong>Discharge T &gt; 115 °C</strong></td>
<td>2.70 (101.3 %)</td>
<td>2.70 (101.2 %)</td>
<td>2.65 (99.3 %)</td>
<td>2.61 (97.9 %)</td>
</tr>
<tr>
<td>H4 (A-7(-8)/W*-35)</td>
<td>2.11</td>
<td>2.40 (113.9 %)</td>
<td>2.33 (110.5 %)</td>
<td>2.16 (102.5 %)</td>
<td>2.23 (105.8 %)</td>
<td>2.16 (102.7 %)</td>
</tr>
<tr>
<td>H5 (A2(1)/W*-35)</td>
<td>3.03</td>
<td>3.06 (101.1 %)</td>
<td><strong>3.32 (108.8 %)</strong></td>
<td>3.03 (100.2 %)</td>
<td>3.04 (100.5 %)</td>
<td>3.01 (99.5 %)</td>
</tr>
<tr>
<td>H6 (A12(11)/W*-35)</td>
<td>4.10</td>
<td>3.58 (87.3 %)</td>
<td><strong>4.38 (106.8 %)</strong></td>
<td>4.34 (105.9 %)</td>
<td>4.16 (101.4 %)</td>
<td>4.36 (106.3 %)</td>
</tr>
<tr>
<td>HL1 (A15/W*-22)</td>
<td>2.16</td>
<td>2.45 (113.5 %)</td>
<td>2.09 (96.5 %)</td>
<td>2.21 (102.1 %)</td>
<td>2.32 (107.6 %)</td>
<td>2.24 (103.5 %)</td>
</tr>
<tr>
<td>HL2 (A10/W*-42.5)</td>
<td>1.58</td>
<td>1.77 (112.2 %)</td>
<td><strong>1.78 (112.7 %)</strong></td>
<td>1.63 (103.5 %)</td>
<td>1.68 (106.7 %)</td>
<td>1.63 (103.1 %)</td>
</tr>
<tr>
<td>HL3 (A24(20)/W*-54.8)</td>
<td>3.05</td>
<td><strong>Discharge T &gt; 115 °C</strong></td>
<td>3.33 (109.4 %)</td>
<td>3.29 (108.0 %)</td>
<td>3.30 (108.1 %)</td>
<td><strong>3.39 (111.2 %)</strong></td>
</tr>
</tbody>
</table>

With the exception of H4, H5, HL1 and HL2 conditions, the alternative refrigerants lead to lower heating capacities than with R410A. COPs are equivalent or greater for all the conditions. The heating capacity at H1 rating condition (Figure 4-4 a)) is significantly reduced with R447A, HPR2A and R32 because the heat pump has carried out defrosting cycles that did not occur during the tests with R454B, R459A and R410A.

R454B and R459A lead to heating capacities lower or equivalent than R410A, from -7.5 % to +6.8 % and from -8.9 % to -1.1 % respectively. Heating capacities with HPR2A are lower or equivalent to those with R410A (-33.4 % to +1.3 %). R447A shows lower heating capacities than R410A (-30.8 % to -2.4 %). R459A achieves equal or greater COPs than R410A (+0.2 % to +8 %).

All COPs with R454B are equivalent or greater than with R410A (+0.5 % to +12.7 %), with the exception of the HL1 limit condition where it is lower (-3.5 %), HPR2A obtains COPs equal or greater than R410A (-0.7 % to +8.9 %) for H3 to HL2 conditions and lower for H1 and H2 conditions (-5.8 % and -3.2 %). R447A shows lower or equivalent COPs than R410A for the conditions between H1 and H3 (-3.8 % to -0.3 %) and higher or equivalent for the conditions H4 to HL2 (-0.5 % to +11.2 %). Capacities and COPs obtained with R32 for negative air temperatures are significantly greater than those with R410A, between +0.4 % to +14.4 % and +1.1 % to +13.9 %, respectively. For these conditions (H4, HL1, HL2), R32 show the best performances.

There is an important dispersal of the discharge temperatures, but with the exception of R32, the four alternatives get discharge temperatures close to those of R410A. R32 did not allow
performing H3 rating condition and HL3 limit conditions, because the discharge temperature was higher than 115 °C. To reach a discharge temperature below 115 °C, the outlet water temperatures were set to 48 °C for H3 and 43 °C for HL3.

4.3.3.4 Performance verification

To make sure that the use of the alternative refrigerants did not damage the heat pump, tests with the initial R410A charge (2.35 kg) were performed after each series of tests with the alternative refrigerants. This verification allowed determining the heat pump performance deviation but it does not give any answer concerning the long term use of the alternative refrigerants. The performance gaps obtained are quite small (from -1 % to +5 %) and within the uncertainty of measurement. According to the results, we can conclude that there was no notable damage of the heat pump after the use of the refrigerant alternatives.

With the exception of R32, the alternative refrigerants might be considered as alternatives to R410A for both modes and all the conditions tested in this study. R454B and R459A showed the best performances. R32 could be used, but the heat pump operating map should be decreased because of high discharge temperatures, especially when condensation occurs at high temperatures.

4.4 Experimental evaluation of R407C alternative refrigerant in a Water-to-air reversible Heat pump

4.4.1 Properties of alternative refrigerant to R407C

Table 4-9 presents the main properties of the refrigerant studied to replace R407C. The data source is the software NIST REFPROP Version 10 [8]. Figure 4-6 shows the refrigerant's GWP and safety class.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Composition</th>
<th>GWP 100</th>
<th>Critical temperature (°C)</th>
<th>Normal boiling point (°C)</th>
<th>Glide (K, at 40°C)</th>
<th>Safety class</th>
</tr>
</thead>
<tbody>
<tr>
<td>R407C</td>
<td>R32/R125/R134a (23/25/52 wt.%)</td>
<td>1,650</td>
<td>86.1</td>
<td>-40.1</td>
<td>7.0</td>
<td>A1</td>
</tr>
<tr>
<td>R454C</td>
<td>R1234yf/R32 (78.5/21.5 wt.%)</td>
<td>148</td>
<td>85.7</td>
<td>-42.4</td>
<td>8.5</td>
<td>A2L</td>
</tr>
<tr>
<td>R455A</td>
<td>R1234yf/R32/R744 (75.5/21.5/3 wt.%)</td>
<td>146</td>
<td>85.6</td>
<td>-45.6</td>
<td>9.7</td>
<td>A2L</td>
</tr>
</tbody>
</table>

Figure 4-5: Refrigerant's GWP and safety class
The GWP of R454C and R455A are significantly lower than that of R407C (-91 %) and they are below the most compelling GWP limit (150) of the European F-Gas regulation. R454C and R455A have an A2L safety class, which means they have a low flammability and are non-toxic. R454C and R455A have a glide slightly higher than R407C.

The saturation properties (pressure-temperature) are shown in Figure 4-6. The refrigerants have equivalent saturation properties.

![Figure 4-6: Saturation properties: Pressure-Temperature](image)

4.4.2 Experimental investigation

Drop-in tests were carried out to assess the heat pump performance. The heating capacity of the tested water-to-air heat pump is close to 2.9 kW at H1 rating condition (see Table 4-11). It is a reversible, packaged and ducted appliance. The heat pump is equipped with a fixed capacity hermetic rotary compressor and a capillary tube as expansion device. The initial charge of R407C is 0.64 kg.

The test conditions in cooling mode and in heating mode are given in
Table 4-10 and Table 4-11, respectively. For each refrigerant, a charge optimization was done at the CL2 limit condition, then the rating and operating limit condition tests were performed, and finally performance verification with R407C was carried out on the C1 rating condition to detect any anomaly. The tests were carried out in one of CETIAT climatic rooms, according to EN 14511 standard [11]. During tests, measurements allowed the determination of thermal capacities, electric energy consumptions, efficiencies (EER or COP), as well as pressures and temperatures on the refrigerant circuit.

According to the uncertainty of measurement on the laboratory’s instrumentation, capacities were determined with a maximal uncertainty of 5 % and electric energy consumptions with a maximal uncertainty of 1 %.
Table 4-10: Rating (C) and operating limit conditions (CL) in cooling mode

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Inlet water temperature (°C)</th>
<th>Outlet water temperature (°C)</th>
<th>Water flow rate (l/h)</th>
<th>Air temperature (°C)</th>
<th>Air flow rate (m³/h) (at 1013 mbar and 20°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>30</td>
<td>35</td>
<td>-</td>
<td>27(19)</td>
<td>475</td>
</tr>
<tr>
<td>C2</td>
<td>22</td>
<td>*</td>
<td>485</td>
<td>22(15)</td>
<td>450</td>
</tr>
<tr>
<td>CL1</td>
<td>41</td>
<td>*</td>
<td>250</td>
<td>37(27.7)</td>
<td>500</td>
</tr>
<tr>
<td>CL2</td>
<td>42</td>
<td>*</td>
<td>250</td>
<td>22(15)</td>
<td>500</td>
</tr>
</tbody>
</table>

Table 4-11: Rating (H) and operating limit conditions (HL) in heating mode

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Inlet water temperature (°C)</th>
<th>Outlet water temperature (°C)</th>
<th>Water flow rate (l/h)</th>
<th>Air temperature (°C)</th>
<th>Air flow rate (m³/h) (at 1013 mbar and 20°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>20</td>
<td>17</td>
<td>-</td>
<td>20(15)</td>
<td>475</td>
</tr>
<tr>
<td>H2</td>
<td>12</td>
<td>*</td>
<td>250</td>
<td>12(7.2)</td>
<td>500</td>
</tr>
<tr>
<td>HL1</td>
<td>36</td>
<td>*</td>
<td>250</td>
<td>27(19.5)</td>
<td>450</td>
</tr>
<tr>
<td>HL2</td>
<td>36</td>
<td>*</td>
<td>485</td>
<td>27(19.5)</td>
<td>450</td>
</tr>
</tbody>
</table>

The operating limit conditions were fixed by the heat pump manufacturer: they correspond to the boundary conditions of operation of the heat pump with R407C.

4.4.3 Results of the experimental evaluation of R407C alternative refrigerant

4.4.3.1 Charge optimization

To perform the charge optimization, the initial alternative refrigerant charge was about 0.416 kg (corresponding to 65 % of the initial R407C charge). At CL2 limit condition, refrigerant was added (+25 g in every 30 minutes) while four parameters were monitored: EER, cooling capacity, superheating and subcooling. The objective was to determine the optimal charge for a superheating close to 2 K. The optimal charges obtained for both refrigerants are reported in Table 4-12.

Table 4-12: Charge optimization results (at CL2 limit condition)

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R407C (baseline)</th>
<th>R454C</th>
<th>R455A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charge (kg)</td>
<td>0.64</td>
<td>0.64 (0 %)</td>
<td>0.71 (+11.0 %)</td>
</tr>
<tr>
<td>Cooling capacity (kW)</td>
<td>1.75</td>
<td>1.89 (+7.7 %)</td>
<td>1.68 (-4.2 %)</td>
</tr>
<tr>
<td>EER (-)</td>
<td>2.46</td>
<td>2.44 (-0.6 %)</td>
<td>2.06 (-15.5 %)</td>
</tr>
<tr>
<td>Superheating (K)</td>
<td>3.20</td>
<td>4.80 (+1.6 K)</td>
<td>0.7 (-2.5 K)</td>
</tr>
<tr>
<td>Subcooling (K)</td>
<td>6.80</td>
<td>13.90 (+7.1 K)</td>
<td>12.0 (+5.2 K)</td>
</tr>
</tbody>
</table>

R454C and R407C lead to the same optimum load of 640 g. The charge of R455A is higher, 710 g (+11 %). With the R454C the cooling capacity is higher than that of the R407C, on the other hand the EER is lower. The R455A has lower performance.

4.4.3.2 Cooling mode

Figure 4-7 presents the results obtained in cooling mode: ratios of performance (alternative/R407C) and the discharge temperature. Table 4-13 provides values for the heat pump cooling capacity and EER.
Figure 4-7: Heat pump performance in cooling mode: (a) Capacity ratio, (b) EER ratio; (c) Discharge temperature

Table 4-13: Cooling capacity (green color highlights best performance)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Refrigerant</th>
<th>Cooling capacity (kW) (ratio alternative/base)</th>
<th>EER (-) (ratio alternative/base)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>R407C (baseline)</td>
<td>R454C</td>
<td>R455A</td>
</tr>
<tr>
<td>C1 (W30-35/A27(19))</td>
<td>2.16</td>
<td>2.24</td>
<td>2.24</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(103.7 %)</td>
<td>(104.1 %)</td>
</tr>
<tr>
<td>C2 (W22-*/A22(15))</td>
<td>1.97</td>
<td>2.08</td>
<td>2.19</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(105.6 %)</td>
<td>(111.2 %)</td>
</tr>
<tr>
<td>CL1 (W41-*/A37(27.7))</td>
<td>2.73</td>
<td>2.72</td>
<td>2.58</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(99.5 %)</td>
<td>(94.6 %)</td>
</tr>
<tr>
<td>CL2 (W42-*/A22(15))</td>
<td>1.75</td>
<td>1.89</td>
<td>1.68</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(107.7 %)</td>
<td>(95.8 %)</td>
</tr>
</tbody>
</table>

The R455A achieves cooling capacities equivalent to or greater than those of R407C (from -5.4 % to +11.2 %) but significantly lower EER than those of R407C (from -15.5 % to -8.4 %). The R454C achieves higher cooling capacities or equivalent to those of R407C (from -0.5 % to +7.7 %)
Annex 54, Heat pump systems with low-GWP refrigerants

and lower or equivalent EER (from -10.7 % to 0 %). Under the conditions tested in cooling mode, the R454C shows better performance than the R455A. The discharge temperatures of the three fluids are equivalent.

For R455A, CL1 limit condition was carried out with a water inlet temperature of 36 °C, against 41 °C for the other two, in order to limit the condensing pressure to 30 bar (HP pressure switch limit).

4.4.3.3 Heating mode

Figure 4-8 presents the results obtained in heating mode: ratios of performance (alternative/R407C) and discharge temperature. Table 4-14 provides values for heat pump heating capacity and COP.

Figure 4-8: Heat pump performance in heating mode: (a) Capacity ratio, (b) EER ratio; (c) Discharge temperature
Table 4-14: Heating capacity (green color highlights best performance)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heating capacity (kW) (ratio alternative/base)</th>
<th>COP (-) (ratio alternative/base)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>R407C (baseline)</td>
<td>R454C</td>
</tr>
<tr>
<td></td>
<td>R455A</td>
<td>R407C (baseline)</td>
</tr>
<tr>
<td>H1 (W20-17/A20(15))</td>
<td>2.91 (101.4 %)</td>
<td>3.15 (108.5 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.29 (98.2 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.33 (100.8 %)</td>
</tr>
<tr>
<td>H2 (W12-*A12(7.2))</td>
<td>2.32 (103.6 %)</td>
<td>2.51 (108.0 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.37 (100.0 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.30 (98.2 %)</td>
</tr>
<tr>
<td>HL1 (W36-*A27(19.5))</td>
<td>3.23 (101.2 %)</td>
<td>3.51 (108.6 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.93 (97.9 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.98 (101.3 %)</td>
</tr>
<tr>
<td>HL2 (W36-*A27(19.5))</td>
<td>3.46 (102.6 %)</td>
<td>3.67 (106.1 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.00 (97.3 %)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4.03 (100.9 %)</td>
</tr>
</tbody>
</table>

R454C obtains higher heating capacities than those of R407C (from +1 % to +4 %) and COP equivalent to those of R407C (from -3 % to 0 %). R455A achieves higher heating capacities than those of R407C (from +6 % to +8 %) and COP close to those of R407C (from −2 % to +1 %). Under the conditions tested in heating mode, R455A shows better performance than R454C and R407C. The discharge temperatures are equivalent for the three fluids.

4.4.3.4 Performance verification

To make sure that the use of R454C and R455A did not damage the heat pump, tests with the initial R407C charge (0.64 kg) were performed. The performance was checked at C1 rating condition. The performance gaps obtained for the cooling capacity and the EER are quite small, +2.5 % and +2.7 %, respectively, and within the uncertainty of measurement. According to the results, we can conclude that there was no notable damage of the heat pump after the use of R454C and R455A.

R454C can be considered as a replacement fluid for R407C without a significant reduction of performances of the heat pump in its operating range. R455A obtains very good performances in heating mode while in cooling mode, its performance is degraded. In cooling mode, the fluid flow in the finned coil and in the plate heat exchanger is co-current. This type of circulation results in a lowering of the evaporating temperature and an increase in the condensing temperature, to the detriment of the EER. With a fluid having a high glide, the performance losses are accentuated.

R454C and R455A have shown encouraging results for the replacement of R407C. The R454C appears to be suitable for replacing the R407C for a reversible machine, since the performance is almost equivalent to that of the R407C in both modes of operation. In heating only mode, with counter-current exchangers, R455A will be the most suitable.

4.5 Experimental evaluation of R134a alternative refrigerant in a split Heat Pump Water Heater

4.5.1 Properties of alternative refrigerants to R134a

Table 4-15 presents the main properties of the refrigerants studied to replace R134a. The data source is the software NIST REFPROP Version 10 [8]. Figure 4-9 shows the refrigerant's GWP and safety class.
Table 4-15: Refrigerant properties

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Composition</th>
<th>GWP&lt;sub&gt;100&lt;/sub&gt;</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>R134a</td>
<td>R134a (100 wt.%)</td>
<td>1,430</td>
<td>101.1</td>
<td>-26.1</td>
<td>0</td>
<td>A1</td>
</tr>
<tr>
<td>R513A</td>
<td>R1234yf/R134a (56/44 wt.%)</td>
<td>631</td>
<td>96.5</td>
<td>-29.2</td>
<td>0</td>
<td>A1</td>
</tr>
<tr>
<td>R450A</td>
<td>R1234ze(E)/R134a (58/42 wt.%)</td>
<td>604</td>
<td>104.5</td>
<td>-22.6</td>
<td>0.5</td>
<td>A1</td>
</tr>
<tr>
<td>R1234yf</td>
<td>R1234yf (100 wt.%)</td>
<td>4</td>
<td>94.7</td>
<td>-29.4</td>
<td>0</td>
<td>A2L</td>
</tr>
</tbody>
</table>

Figure 4-9: Refrigerant's GWP and safety class

The GWP of R513A and R450A are lower than that with R134a. R513A and R450A have an A1 safety class, which means they are non-flammable and non-toxic. R513A have no glide. R450A has a glide of 0.5 K. The GWP of R1234yf is significantly lower than that with R134a and it is below the most compelling GWP limit (150) of the European F-Gas regulation. R1234yf has an A2L safety class, which means it has a low flammability and is non-toxic. The saturation properties (pressure-temperature) are shown in Figure 4-10. The three refrigerants have equivalent saturation properties. R450A remains slightly less volatile than R134a.

Figure 4-10: Saturation properties: Pressure-Temperature
4.5.2 Experimental investigation

Drop-in tests were carried out to assess the heat pump water heater (HPWH) performance. The HPWH in test is a split system having a water tank of 200 l. It is equipped with a fixed capacity hermetic rotary compressor and an electronic expansion device. The initial charge of R134a is 1.6 kg.

The tests consisted in a heating up of the water in the tank. When the desired temperature was reached (measured at the top of the tank by a Pt100 sensor), a hot water tapping of 10 liters/min was performed to determine the energy content until the tapped water reached the initial water temperature. During all tests, measurements allowed the determination of electric power inputs, refrigerant pressures and temperatures, water tank temperature and energy of the hot water tapping. For each refrigerant, a charge optimization was done, and then heating up of the tank was performed for three outdoor air temperatures. Finally, performance verification with R134a was carried out to detect any anomaly. Refrigerants are compared based on the heating up time, the COP (= water energy content/ electric energy consumption) and the maximal discharge temperature. The tests were carried out in one of CETIAT climatic rooms. According to the uncertainty of measurement on the laboratory’s instrumentation, capacities were determined with a maximal uncertainty of 5 % and electric energy consumptions with a maximal uncertainty of 1 %. The test conditions in charge optimization and in performance evaluation are described in Table 4-16 and Table 4-17, respectively.

### Table 4-16: Test conditions for charge optimization

<table>
<thead>
<tr>
<th>Outdoor air dry bulb (wet bulb) temperatures (°C)</th>
<th>Ambient air dry bulb temperature around the tank (°C)</th>
<th>Initial water tank temperature (°C)</th>
<th>Heating up of the tank</th>
<th>Water tapping flow rate (l/min)</th>
<th>Inlet water temperature (°C)</th>
<th>Stopping temperature of water tapping (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7 (6)</td>
<td>20</td>
<td>15</td>
<td>From 15 °C to 45 °C</td>
<td>10</td>
<td>14</td>
<td>15</td>
</tr>
</tbody>
</table>

### Table 4-17: Test conditions for performance evaluation

<table>
<thead>
<tr>
<th>Outdoor air dry bulb (wet bulb) temperatures (°C)</th>
<th>Ambient air dry bulb temperature around the tank (°C)</th>
<th>Initial water tank temperature (°C)</th>
<th>Heating up of the tank</th>
<th>Water tapping flow rate (l/min)</th>
<th>Inlet water temperature (°C)</th>
<th>Stopping temperature of water tapping (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 (1) 7 (6) 35</td>
<td>20</td>
<td>10</td>
<td>From 10 °C to 60 °C</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
</tbody>
</table>

In addition, for each refrigerant EN16147:2011 [12] tests were performed. The tests consist of the following four principal stages:

A: Heating up period;
B: determination of standby power input;
C: determination of the energy consumption and the coefficient of performance for heating domestic water by using the reference tapping cycles (L);
D: determination of a reference hot water temperature and the maximum quantity of usable hot water in a single tapping.
4.5.3 Results of the experimental evaluation of R134a alternative refrigerants

4.5.3.1 Charge optimization

To perform the charge optimization, the initial alternative refrigerant charge was about 1.12 kg (corresponding to 70% of the initial R134a charge). Charge optimization was carried out at the conditions. When refrigerant charge was added (+80 g), a new heating up of the tank was done to determine the electricity consumption, the energy content in the water tank and pressures and temperatures of the refrigerant circuit. The objective was to identify the performance curve inflexion point to determine the optimal charge. The optimal charges obtained are reported in Table 4-18.

Table 4-18: Charge optimization results (Heating up from 15 °C to 45 °C)

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R134a (baseline)</th>
<th>R513A</th>
<th>R1234yf</th>
<th>R450A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charge (kg)</td>
<td>1.60</td>
<td>1.60 (0 %)</td>
<td>1.68 (+5 %)</td>
<td>1.68 (+5 %)</td>
</tr>
<tr>
<td>Heating up time (hh:mm:ss)</td>
<td>03:35:31</td>
<td>03:29:13 (- 6 min 13 s)</td>
<td>03:36:19 (+ 48 s)</td>
<td>03:59:40 (+24 min 09 s)</td>
</tr>
<tr>
<td>COP (-)</td>
<td>3.79</td>
<td>3.79 (0 %)</td>
<td>3.79 (0 %)</td>
<td>3.64 (- 4.9 %)</td>
</tr>
</tbody>
</table>

R513A and R1234yf show optimal charge close to R134a and equivalent performances to R134a. R450A shows optimal charge close to R134a and slightly lower performance than R134a.

4.5.3.2 Heating up performance evaluation

Figure 4-11 presents the results obtained during the heating up of the water tank for three outdoor air temperatures.
provides the values for various parameters. R513A, R1234yf and R450A show equivalent performances (heating up time, COP, electric energy consumption) to those of R134a. The discharge temperatures reached with alternatives are lower than with R134a (-14 K for 450A, -10 K for R1234yf and -5 K for R513A). At 2(1) °C with R1234yf and R513A, the heating up times are lower than with R134a, but with R450A the heating up times is higher than with R134a.

Table 4-19: Heating up performance evaluation (green color highlights best performance)

<table>
<thead>
<tr>
<th>Dry air temperature (wet bulb) (°C)</th>
<th>2(1)</th>
<th>7(6)</th>
<th>35</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Refrigerant</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R134a (base)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R513A</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1234yf</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R450A</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Heating up time (hh:mm:ss)</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R134a (base)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R513A</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1234yf</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R450A</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Electric energy consumption (Wh)</strong></td>
<td>5.76</td>
<td>5.568</td>
<td>5.446</td>
</tr>
<tr>
<td><strong>COP (-)</strong></td>
<td>2.0</td>
<td>2.1 (+3.3 %)</td>
<td>2.2 (+5.9 %)</td>
</tr>
<tr>
<td><strong>Maximal discharge temperature (°C)</strong></td>
<td>83.9</td>
<td>78.9 (-5.0 K)</td>
<td>73.9 (-10.0 K)</td>
</tr>
</tbody>
</table>
Figure 4-11: Heating up performance evaluation: (a) Heating up time; (b) Electric energy consumption; (c) Water energy content; (d) COP; (e) Maximal discharge temperature.
4.5.3.3 EN 16147:2011 tests

Table 4-20 provides the main results of the EN 16147:2011 tests.

<table>
<thead>
<tr>
<th>Test conditions</th>
<th>R134a</th>
<th>R513A</th>
<th>R1234yf</th>
<th>R450A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant charge kg</td>
<td>1.6</td>
<td>1.6</td>
<td>1.68</td>
<td>1.68</td>
</tr>
<tr>
<td>Outdoor air temperature (DB / WB) °C / °C</td>
<td>7(6)</td>
<td>7(6)</td>
<td>7(6)</td>
<td>7(6)</td>
</tr>
<tr>
<td>Size of tapping cycle °C</td>
<td>L</td>
<td>L</td>
<td>L</td>
<td>L</td>
</tr>
<tr>
<td>DHW set point (fixed on the product) °C</td>
<td>55</td>
<td>55</td>
<td>55</td>
<td>55</td>
</tr>
</tbody>
</table>

Performances in accordance with standard EN16147:2011

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating up energy input Wh</td>
<td></td>
<td>3214</td>
<td>3179</td>
<td>3222</td>
<td>3183</td>
</tr>
</tbody>
</table>

Determination of Standby power

| Duration of the last on-off cycle to determine the standby power input t<sub>es</sub> | -     | 27:01:18 | 26:54:36 | 27:12:54 | 24:59:03 |
| Energy input during last on-off cycle Wh W<sub>es</sub> |       | 961      | 926      | 949      | 918      |
| Standby power input W P<sub>es</sub> |       | 36       | 34       | 35       | 37       |

Determination of COP

<table>
<thead>
<tr>
<th>Total useful heat energy during the whole tapping cycle Wh Q&lt;sub&gt;TC&lt;/sub&gt;</th>
<th>11655</th>
<th>11655</th>
<th>11655</th>
<th>11655</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time period of test cycle s T&lt;sub&gt;TTC&lt;/sub&gt;</td>
<td>43h 52m 3s</td>
<td>43h 36m 17s</td>
<td>44h 9m 15s</td>
<td>42h 10m 7 s</td>
</tr>
<tr>
<td>Measured electrical energy consumption during the whole tapping cycle Wh W&lt;sub&gt;EL-M-TC&lt;/sub&gt;</td>
<td>4940</td>
<td>4774</td>
<td>4852</td>
<td>4991</td>
</tr>
<tr>
<td>Electrical energy consumption of fans or liquid pumps Wh W&lt;sub&gt;EL-CORR&lt;/sub&gt;</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>Calculated heat energy produced by electricity during the whole tapping cycle Wh Q&lt;sub&gt;EL-TC&lt;/sub&gt;</td>
<td>56</td>
<td>60</td>
<td>58</td>
<td>53</td>
</tr>
<tr>
<td>Total electrical energy consumption during the whole tapping cycle Wh W&lt;sub&gt;EL-TC&lt;/sub&gt;</td>
<td>4289</td>
<td>4159</td>
<td>4208</td>
<td>4376</td>
</tr>
<tr>
<td>Coefficient of performance COP&lt;sub&gt;DHW&lt;/sub&gt;</td>
<td>2.72</td>
<td>2.80</td>
<td>2.77</td>
<td>2.66</td>
</tr>
<tr>
<td>Reference hot water temperature °C θ&lt;sub&gt;WH&lt;/sub&gt;</td>
<td>53.1</td>
<td>53.1</td>
<td>53.1</td>
<td>53.1</td>
</tr>
<tr>
<td>Maximum quantity of usable hot water liters V&lt;sub&gt;max&lt;/sub&gt;</td>
<td>282.1</td>
<td>292.6</td>
<td>292.2</td>
<td>290.7</td>
</tr>
</tbody>
</table>

The normative tests provide the same conclusions as the heating up performance tests, i.e., R513A, R1234yf and R450A show equivalent performances to those of R134a.

4.5.3.4 Performance verification

To make sure that the use of alternative refrigerants did not damage the HPWH, a heating up with the initial R134a charge (1.6 kg) was performed. The performance gaps obtained for heating up time and electric energy consumption are quite small, +7 min 21 s and +1.5 %, respectively, and within the uncertainty of measurement. We can conclude that there was no notable damage of the heat pump after the use both alternative refrigerants.

According to these results, R513A, and R1234yf might be considered as alternatives to R134a without performance impact, and R450A might be considered as alternatives to R134a, but with a decrease of the thermal capacity of the system (Heating up time longer).

4.6 Conclusions

A total of 10 low-GWP alternative refrigerants were evaluated with not less than 130 performance tests. The principal results of the study are summarized below.

R459A, R454B, R447A, HPR2A and R32 were investigated for the drop-in replacement of R410A in a 10 kW air-to-water reversible heat pump. R410A replacement by HFC/HFO mixtures showed no particular problem and the performance obtained is, aside from some very few exceptions, almost equivalent (+/- 10 %) to that with R410A. Furthermore, in operating limit conditions, the heat pump worked normally with alternative refrigerants HFC/HFO mixtures. With
R32, the operating map of the heat pump would be decreased because of the high discharge temperatures reached. R454B and R459A showed the best performances.

R454C and R455A were evaluated as a possible alternative to R407C in a 3 kW water-to-air reversible heat pump. The R454C seems to be more suited to replacing the R407C for a reversible machine, since the performance is almost equivalent to that of the R407C in both operating modes. For heating only mode, R455A achieves the best results (+ 6.1 % to + 8.6 % heating capacities and equivalent COP). R454C and R455A can therefore be considered as alternative for R407C.

R1234yf, R513A and R450A were tested for the replacement of R134a in a split heat pump water heater having a water tank of 200 liters. They showed equivalent performances to R134a. The discharge temperatures reached with alternatives are lower than those with R134a, of -14K for R450A, -10 K for R1234yf and -5 K for R513A. R513A and R1234yf might be considered as alternatives to R134a without significant performance impact and R450A might be considered as alternatives to R134a, but with a decrease of the thermal capacity of the system (Heating up time longer).

With the exception of R450A and R513A, which have a A1 safety class, all the others alternative, R1234yf, R459A, R454B, R447A, HPR2A, R32, R454C and R455A, have a A2L safety class. It means that a new risk must be handled: the flammability. The use of these alternative refrigerants will require a complete study of the risks, sizing and compatibility.

Finally, what are the most promising Low-GWP refrigerants to replace the HFC commonly used in heat pumps? Figure 4-12 tries to answer to this question.

![Image of a chart showing GWP values and performance rankings for different refrigerants.]

**Figure 4-12: Most promising Low-GWP refrigerants**

Long-term alternatives (GWP<150) exist for R407C and R134a, but for R410A only transition alternatives are available (150<GWP<750). Other solutions need to be studied, especially natural fluids.

These experimental results will be useful to the HVAC community for selecting the most promising refrigerant candidates for replacement of R410A, R134a and R407C. Beyond drop-in, improving the thermal performances of the heat pumps would require component optimization. For example, it would be necessary to re-size and to replace the expansion valve, especially when a calibrated orifice or a capillary tube is used, or to optimize the design of the heat exchanger(s).

### 4.7 References


5 Country Report, Sweden

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Stockholm, Sweden
5.1 Introduction

Sweden was late in joining Annex 54 and did not deliver a country report for 2019. The present report therefore combines the topics foreseen in the reports for 2019, Review of the state of the art technologies and 2020, Case studies and design guidelines for optimization of components and systems.

The report starts with a historic review of research on heat pumps with low-GWP refrigerants in Sweden. As will be shown, this research goes back to the 1990’s and is mainly focused on natural refrigerants as alternatives to synthetic fluids. Only during the last ten years, there has been some activities on synthetic low-GWP refrigerants.

Sweden is a small country in terms of population with about 10 million inhabitants, much less than the mega-cities in other parts of the world. As a result, the total number of universities is limited. Most of the university research related to heat pump technology has been performed at KTH Royal Institute of Technology. Another active institution is RI.SE (Research Institutes of Sweden) and their group in Borås. Representatives for both these organizations are participating in the work for IEA Annex 54.

5.2 Early History

5.2.1 Introduction to heat pump research in Sweden

Sweden has very good conditions for using heat pumps for heating: First, the climate is cold with a heating season lasting from Oct 1 to April 30. The potential energy savings by using heat pumps are therefore high. Second, the ratio of the cost of electricity to the cost of fuel is lower than in most countries. There are several reasons for this: Sweden has no fossil fuel resources, but several rivers suitable for hydropower. Also, Sweden is one of the countries in the world with the highest number of nuclear power plants per capita. There has thus never been a shortage of electricity, and the generation has never been based on fossil fuels to a large extent.

During the first and second oil crisis in the 1970’s many of the nuclear power plants were set in operation, so while oil prices were going up, the price of electricity was going down. As most single family houses at the time were heated by oil furnaces, there was a large interest for switching to electric heating. At the time, heat pumps were unknown to the general public, and direct or indirect electric heating was installed in many houses. From the government there was interest to ensure that the electricity was utilized in the best possible way, so research funding was made available already in the first half of the 1980’s for developing the heat pump technology, together with other technologies to decrease the energy use in general. Subsidies were also introduced to support house owners in installing heat pumps, and this also had a positive effect on industry, with several companies starting production of heat pumps for domestic use. Heat pumps were also introduced in the district heating systems in all major cities where a suitable heat source could be found, typically lake- or sea water.

During the first years of the 1980’s, there was little discussion about the refrigerants used. The effect on the ozone layer by CFCs had been proposed by Rowland and Molina already 1974, and resulted in restrictions on the use of CFC as propellants, but had little effect on the refrigeration industry. R12 and R22 were the refrigerants of choice for heat pumps. However, with the Montreal protocol in 1987 there was a strong focus on new refrigerants with low ODP. At the time, there was no vivid discussion on the contribution of the refrigerants to global warming, but there was, amongst researchers and NGOs, a general concern that new synthetic fluids may have unknown environmental effects. For this reason, there was a push for investigating the use of natural refrigerants. Professor Gustav Lorentzen in Trondheim, Norway, was among the first to argue for the use of natural fluids, in particular proposing the use of carbon dioxide as refrigerant.
In a few years, the industry had to switch to low ODP-refrigerants and several research projects were financed by the Swedish Energy Agency related to the new refrigerants, in particular the use of R407C. However, there were also projects related to natural, low-GWP refrigerants.

In 1995, the government agency NUTEK opened up a research program called Alternativa Köldmedier (Alternative Refrigerants), financing research on the use of alternatives to the refrigerants used up to then. This was the first in a series of research programs, later financed by the Swedish Energy Agency, each typically lasting three years, with different names but similar focus: Enhancing the performance of heat pumping equipment, while at the same time using refrigerants with low environmental impact. These programs were: Klimat21, Effsys, Effsys2, Effsys+, Effsys Expand and Termo. The last of these is still running since a couple of years and is the main national funding source for research in the area.

There has also been international funding available for research on alternative refrigerants, in particular through the EU research programs. Several such programs with Swedish participation will be mentioned in the following.

5.2.2 Previous national projects on low-GWP refrigerants

As indicated above, there have been a range of projects related to low-GWP refrigerants within national research programs financed by the government through NUTEK and the Swedish Energy Agency. A selection of these projects are mentioned below.

5.2.2.1 Projects within the research program Alternativa Köldmedier (Alternative refrigerants) 1995 - 1997

- **Consequences of changes in composition of mixtures:** The aim of this project was to investigate how changes in composition of non-azeotropic mixtures, due to accumulation of the working fluid in different parts of the system influence the COP and capacity in different applications. It was carried out with an extensive simulation program for vapor compression cycles. It was concluded that the composition of some mixtures which are normally non-flammable, may become flammable due to differences in the solubility of the components in the compressor oil. A result which may still be of interest in a time where new mixtures are designed with the purpose of suggesting fluids which have low-GWP but are non-flammable.

- **Evaluation of cyclopropane as working agent in small refrigeration system:** The stability and health effects of cyclopropane was investigated and the performance in a household refrigerator was tested. No negative effects were found, and the refrigerator is still in operation after 25 years in the coffee-room at KTH.
• **Retrofit of refrigeration systems**: In this project, retrofit of non-(H)CFC refrigerants with fluids with less environmental impact was investigated. Experiences from a large number of systems were collected and evaluated regarding performance, chemical stability and life span.

• **Propane in small heat pumps**: The performance of a small heat pump was compared when operated with R22 and with propane. As expected, propane gave lower capacity due to difference in volumetric refrigerating effect. Heat transfer in evaporation and condensation in plate heat exchangers were similar, but pressure drop lower with propane.

• **Propane for heat pump applications using brazed plate heat exchangers**: Heat transfer and pressure drop were measured in plate heat exchangers comparing R22 and propane. The results showed 40-50% lower pressure drop and slightly lower heat transfer. It was concluded that if the heat exchangers were designed for higher pressure drop using propane, the heat transfer performance would be similar. It was also found that theoretical performance was well in line with laboratory tests.

• **Safety with flammable refrigerants**: Risks with HC-refrigerants were discussed. It was concluded that highly flammable refrigerants should not be used for retrofit, but that new systems with charges up to 5 kg (depending on installation) probably could be designed to be considered safe. Mixtures of fluids with low flammability with non-flammable (quenching) fluids was suggested.

5.2.2.2 Projects within the research program Klimat21 1998 – 1999

• **Alternatives to R22 in new and existing plants**: At the time, R22 was the most used refrigerant. It was shown that the existing alternatives led to both lower capacity and lower efficiency. A retrofit has to be accompanied with a general consideration of the necessary requirements for the plant.

• **Refrigerant mixtures in systems**: The performance of shell-and-tube heat exchangers as condensers was compared for R22 and R407C through numerical simulation. It was found that a decrease in performance can be expected for R407C due to selective condensation of the gases in the mixture, and resulting concentration variations. Design alternatives of the heat exchanger in order to get even flow and avoid stagnant zones was suggested.

• **New refrigerant blends to replace R22**: In Sweden there were at the time several large (>25MW) heat pumps in the district heating systems operating with R22. As this
Annex 54, Heat pump systems with low-GWP refrigerants

refrigerant was being phased out, an alternative was sought. The existing pure HFC-alternative, R134a, would result in a large decrease in capacity (up to 35%). Based on simulations of several possible blends, some alternatives were suggested and evaluated. It is interesting to note that one of the blends suggested was a mix of R32 and R134a, still an interesting couple for new mixtures. The glide of some of the mixtures was considered a possible problem. The final recommendation was to accept lower capacity and use R134a, but with a suggestion to consider adding R32 during winter when the higher capacities is required.

5.2.2.3 Projects within the research program Eff-Sys 2001 – 2004

- Technical possibilities and potential for high temperature refrigerants in public and industrial heat pumps: The project investigated the refrigerant options for heat pumps with condensing temperatures above 80°C. It was concluded that isobutane is a good refrigerant for this application, having good transport properties, low saturation pressure and giving high COP, but with the only drawback that it is highly flammable. Among non-flammable fluids HFC236fa is mentioned, giving lower COP, lower volumetric capacity and having high GWP. Ammonia, CO2 and water are also mentioned, with the comment that ammonia and water due to low molecular weight give low pressure rise per stage in turbo-compressors.

- CO2-AC and Bottoming cycle reducing fuel consumption: This project investigated the possibility of using the waste heat from combustion engines as a heat source for bottoming cycles with the intention to decrease energy use of the combustion engine. CO2 was assumed as the working fluid for this cycle.

5.2.2.4 Projects within the research program Effsys2 2006 – 2010

- CO2 heat pumps for the Swedish market, Test and analysis of the SANYO ECO-CUTE heat pump modified for Swedish conditions: CO2 heat pumps for domestic use have been common in Japan since long. However, they have not been designed for the Swedish market. In this project, a modified heat pump from Sanyo was tested in the lab. It was concluded that the performance was as expected from the manufacturer’s data. It was also concluded that the European test standard EN 14511-1, is not suitable for testing CO2 heat pumps as the return water temperature to the heat pump is high according to the standard, which is unfavorable for CO2 heat pumps. Also lab tests revealed low COP compared to conventional heat pumps for the same reason: The water temperature in the tank acting as the cold sink for the heat pump was relatively high, and stratification of the water in the tank was not achieved. Suggestions for improving the design of the heat pump were given.

- Properties of new low-GWP refrigerants: Thermo-physical properties of low-GWP refrigerants were obtained and analyzed to compare the performance in comparison to R22 and R134a. The low-GWP refrigerants were propane and HFO1234yf. Figures of Merit (FoM) regarding heat transfer for single phase flow, condensation and evaporation as well as for pressure drop were calculated and tabulated. The results based on fluid properties were compared to experimental results from tests in a small test rig with plate heat exchangers as evaporator and condenser. In the tests, HFO1234yf did not perform as well as expected.
5.2.2.5 Projects within the research program Effsys+ 2010 – 2014

- **Refrigerants with low-GWP, cost and energy efficiency optimization of vapor compression systems:** As the title tells, this project had two parts of which the first was related to low-GWP refrigerants. This part of the project aimed to provide data, support and prerequisite information of alternative refrigerants with low-GWP at the phasing out of HFC refrigerants for existing and new heating / cooling systems. Within this part, life cycle climate performance was used to compare alternative low-GWP refrigerants for heat pumps. The conclusion was that propane was the best option for this application.

- **Ammonia as refrigerant in small refrigeration and heat pump systems:** It is well known in the industry that ammonia is an excellent refrigerant for many applications. It is frequently used in large industrial systems such as breweries and large cold stores. However, it is not used for small systems such as domestic heat pumps. In this project a prototype heat pump was designed, built and tested, designed for a heating capacity of 8 kW, with a dedicated desuperheater for domestic hot water production at 60°C. It was found to be difficult to find suitable components for the system as copper cannot be used. Hermetic compressors typically have copper windings and plate heat exchangers are brazed with either copper or nickel. The system was built around an open compressor using fully welded plate heat exchangers as desuperheater and condenser. The evaporator was a prototype aluminum heat exchanger with flat multichannel tubes. The system could be run with good performance with as little as 120g ammonia.

- **CO2 as refrigerant for cooling of milk:** Two 5 m³ milk tanks, normally using R134a for cooling, were redesigned to use CO2 as refrigerant. Two different solutions were evaluated, one direct and one indirect (using a brine for cooling the tanks). Unfortunately, it was found that the energy use for cooling of the milk was almost twice as high with the CO2 system as with the original R134a system. This was partly because the CO2 system was not optimized for the current application. A TEWI analysis still indicated that the equivalent CO2 emissions could be reduced by 35% using a CO2 system under Swedish conditions.

5.2.2.6 Projects within the research program Effsys Expand 2014 – 2018

- **Refrigerants with low global warming potential:** The purpose of this project was to provide data, information and support for the utilization of alternative refrigerants with low-GWP for existing and new heating/cooling systems at the phasing out of HFCs. The focus was on thermal properties, requirements for the safety of components and energy efficiency. An important aspect of the project was to supply industry with information about new developments. To this end, articles were presented under a standing heading to the technical journal Kyla and Värme (Cooling and Heating) about 7 times per year.

- **Technology for an environmentally friendly ground source heat pump:** Within this project a domestic geothermal heat pump with low environmental impact was developed and tested. The heat pump is designed to use propane as
refrigerant and pure water as the secondary fluid in the boreholes. To minimize the charge of propane, a number of special solutions were used: Special plate heat exchangers with low pressing depth were designed and manufactured and the compressor was a DC scroll compressor normally used for cooling and heating in an electric vehicle. This resulted in an extremely compact system with a charge of about 100 g of propane for a capacity of up to 10 kW.

The reports from the projects mentioned above can be found through the following web site: 
https://varmtochkallt.se/ Results from many of the projects were also presented at scientific conferences and/or in scientific journal publications. The full references are found in the reports.

5.2.3 Previous international / EU-projects

- **SHERHPA, Sustainable Heat and Energy Research for Heat Pump Applications:** 3M€ project running from 2004 to 2007. The Swedish contribution to the project was the design and testing of an ammonia heat pump for domestic applications. This was the initiation of the project mentioned above financed through Effsys+. More information can be found following these links: https://cordis.europa.eu/project/id/500229
  https://iifiir.org/en/fridoc/23855

- **GreenHP, Next generation heat pump for retrofitting buildings:** 5 M€ project running from 2012 to 2016. A propane heat pump for multifamily buildings was designed and tested. All components of the heat pump were optimized and re-designed to reach good performance and low refrigerant charge. The Swedish contribution was calculation of environmental impact with different refrigerants before arriving at the final choice, design and testing of condenser, calculation of expected charge of the system. More information is found through the following links: https://cordis.europa.eu/project/id/308816
  https://cordis.europa.eu/project/id/308816/reporting

- **NXTHPG, Next Generation of Heat Pumps working with Natural fluids:** 3,8 M€ project running from 2012 to 2016. The Swedish contribution to the project was testing of two (20 – 40 kW) heat pumps using propane as refrigerant, one air to water and one water to water. The heat pumps were designed for low charge of refrigerant. Both heat pumps are still in operation at KTH. Other groups worked on hydrocarbon heat pumps and CO2 heat pumps. More information is found here: https://cordis.europa.eu/project/id/307169

- **NARECO2, Natural Refrigerant CO2:** The Swedish contribution to this EU-funded project was to support with our experiences from a national perspective. At the time, CO2 technology for supermarket refrigeration was just introduced and the Nordic countries were first in adapting this technology. Initially the purpose of the project was to compile educational material for students and technicians concerning CO2 technology. It has however grown to be a 500 page document about CO2 technology. More information is found in the following links: https://iifiir.org/en/fridoc/3910,
5.3 Ongoing activities in Sweden

5.3.1 Overview of ongoing projects

As described above, the Swedish government has supported research related to heat pumps and heat pumping technologies through the Swedish Energy Agency since about 30 years. Since 2018, a new research program is in operation called Termo. The program supports not only heat pump research but also other technologies related to heating and cooling of buildings, such as district heating and thermal energy storage of different types. Heat pump research is also supported by other agencies, but to our best knowledge all research projects related to low-GWP refrigerants are funded through the Termo-program.

All the major heat pump manufacturers in Sweden (Thermia, IVT/Bosch, NIBE, CTC) are probably also working on new products with low-GWP refrigerants but details of this work are not public.

Below follows a description of projects within the research program Termo, related to low-GWP refrigerants.

5.3.2 ECO-Pack, Economizer heat pump with isobutane:

The intention with this project is to investigate the possibility of increasing the energy efficiency and capacity of a multi-apartment building heat pump by adding a second smaller heat pump, using the subcooling of the refrigerant in the larger heat pump as a heat source. As the evaporation temperature of this smaller heat pump will be high, a low pressure refrigerant needs to be used. The selected refrigerant for the smaller heat pump is isobutane, the same fluid which is already used in almost all refrigerators sold in Europe. To decrease the risks of using a highly flammable refrigerant, measures have been taken to keep the charge of isobutane as low as possible. Two versions of the economizer heat pump have been constructed. The first tested unit has a scroll compressor originally used in the HVAC system of an electric vehicle, allowing a very large capacity range through speed control. It also has specially designed plate heat exchangers with low channel height as evaporator and condenser. The unit has so far been tested as a standalone unit with good results. In the next stage it will be connected to a water to water heat pump with propane as refrigerant (see NXTHPG project described above). The second heat pump unit will be a more traditional design with less focus on charge reduction.

The type of high temperature heat pump developed within the project can easily deliver heating at the temperature levels required for domestic hot water. A second application for this type of high temperature heat pumps could be in combination with 4th generation district heating with temperature levels well below 60°C.

Within the project a large effort has been put into developing low charge heat exchangers. This work will continue in a new project specifically focused on heat transfer during evaporation and condensation in flat channels.

Figure 5-4
5.3.3 PROPACK, Air to air propane split system with 150g of propane:

In previous projects it has been demonstrated that water to water heat pumps for domestic use (up to 10 kW) can be designed to operate with less than 150g of propane. In this project the intention is to demonstrate that a split AC/HP air/air system can also be designed to function with the same refrigerant charge. The project is led by Klas Andersson Engineering, a small consultancy firm previously working with the demonstration of water to water systems and other participants are Electrolux (systems manufacturer), SANHUA (heat exchangers, control), GRÅNGES (aluminum channels), SANDEN (compressors), ebmpapst (fans) AGA/Linde (refrigerants), Lundahl (consulting), Granryd Consulting and KTH.

5.3.4 Low-GWP refrigerants for high temperature heat pumps

High temperature heat pumps is gaining more and more interest due to the necessity to phase out fossil fuels for heating, and the corresponding increase in primary energy supply in the form of electricity from wind and solar. High temperature heat pumps can be defined as any heat pump delivering heat at or above 70°C. There is no upper boundary for the definition, and heat pumps are sometimes mentioned as a means of converting power to heat in Carnot batteries, where the storage temperature could be several hundred degrees C. The main applications however can be expected to be within the temperature range 70 - 150°C. Applications could be for process heat in the industry, for boosting the temperature from low (or normal) temperature district heating systems or for thermal energy storage. In the present project, the goal is to demonstrate heat pumps reaching at least 120°C. A test rig is currently under construction at KTH where different refrigerants can be tested. Both natural fluids and synthetics will be investigated. The system will be single stage, so the heat source is assumed to be at an elevated level. The experimental work will be done to assess the results from theoretical analyses based on fluid properties.

5.3.5 New refrigerants for environmentally friendly heat pump systems:

This project is a continuation of previous projects on low-GWP refrigerants. Presently an evaluation is ongoing concerning the comparison between R449A and R404A using artificial neural networks. The intention in this study is to investigate the capability of R449A to be used as an alternative to R404A and article will be published soon.

Another part of the project investigates hydrocarbons as low-GWP alternatives to HFCs. A simulation study is done in which different configurations for the refrigeration cycle, including a single stage cycle, a cycle with internal heat exchanger and a cycle with economizer when operating with R290, R1270 and R600a were investigated. An article is prepared and will be submitted within January.

As a part of this project, a literature review was done concerning the formation of tri-fluoro acetic acid (TFA) and other decomposition products from synthetic refrigerants. Most sources indicate that the concentrations of TFA will not reach limits which can be expected to cause environmental impact within the next few decades. However, it is pointed out that TFA is extremely stable and
could remain in the environment for several hundred years. TFA is formed during the decomposition of certain HFOs but also from some HFCs. As HFOs have short atmospheric life time, the local concentrations close to population centers may be much higher than those in more remote areas.

Recent measurements of halogenated organic acids in the ice cap of Greenland have indicated that the concentration of decomposition products (not only TFA) have increased during the last 15 years. Of course, there is no guarantee that unexpected environmental effects from decomposition products will not appear, as with the effect of the CFCs on the ozone layer.

5.3.6 Heat exchangers with low charge:

This project is only indirectly related to low-GWP refrigerants. As noted in the literature, with the exception of CO2, there are no low-GWP refrigerants (GWP less than about 600) for normal temperature ranges which are non-flammable. As the flammability always constitutes a risk, it is desirable to decrease the charge of flammable refrigerant. Reduction of charge can also be motivated by known or unknown environmental risks related to the release of synthetic refrigerants, e.g. formation of TFA and other decomposition products. Another reason for decreasing the charge is that new refrigerants are complex to manufacture and therefore costly to purchase. There are thus several reasons for trying to decrease the charge of refrigerant in heat pump systems. Normally, the largest share of the charge is in the heat exchangers, closely followed by the compressor.

This project is focusing on the fundamental problem of determining heat transfer coefficients in boiling and condensation in flat channels. It is hypothesized that in evaporators, most evaporation takes place in a thin film in between the gas bubbles and the heated wall. In flat channels the areas of these films may be proportionally larger than in standard size channels and the heat transfer may therefore be influenced. At higher heat flux or higher vapor fraction, the bubble size may be large enough for the thin film to dry out at the center of the bubble. These effects are expected to be investigated experimentally using high speed IR thermography and other methods.

5.3.7 Refrigerants with low-GWP – participation in Annex 54

This project is financing the work within IEA Annex 54, i.e. collection of information for State of the Art reports/National reports, and market surveys to estimate the market for low-GWP-systems in Sweden. As part of the project LCCP calculations will be done for some different designs and refrigerants. Finally, as a special task not yet initiated, refrigerant will be sampled from commercial systems with refrigerant blends and analyzed with gas chromatograph to determine if the circulating refrigerant has the composition of the original charge. Reasons for differences could be: 1) that the original concentrations of the components were not according to specifications, 2) that leakage has resulted in a shift in composition, 3) that fluid stored in tanks in the system does not have the same composition as the circulating fluid and 4) that the components have different affinity to the compressor oil. A clear understanding of these phenomena is necessary to correctly evaluate the performance of the system and to make sure that the system is operating safely. A shift in composition may for example give the wrong signal concerning the superheat at the outlet of the evaporator.

The project will cover both natural and synthetic refrigerants.
5.3.8 Trans-critical CO2 systems for supermarkets

Trans-critical CO2 systems are becoming the cooling systems of choice in supermarkets in Scandinavia. As the refrigeration systems require a large share of the supermarkets’ total energy consumption, it is important to design the systems to be as efficient as possible, and to utilize the heat released on the hot side of the cycle for useful purposes. This project aims to compare different solutions for designing the refrigeration systems of supermarkets, with the purpose of using the hot side of the cycle for different heating purposes, e.g. for space heating, sanitary hot water production and even selling excess heat to the district heating system. The analysis also includes the possibility of thermal energy storage in boreholes. State-of-the-art solutions using liquid and vapor ejectors, liquid overfeed evaporators and liquid fraction sensors are also included in the analysis. The project is run in close cooperation with some of the most prominent companies delivering systems to the market.

During the spring of 2021, a new CO2-refrigeration system will be installed for servicing the climatic chambers of the Department of Energy Technology at KTH. This new system will allow experimenting with different options and different running conditions in a way which is not possible in field tests in actual supermarkets.

5.4 Research needs

It is quite clear that we will have to accept using flammable refrigerants in the future, if we want to get away from high GWP fluids. It has been shown in several scientific publications that there is a clear connection between increasing GWP and decreasing flammability. The simple explanation to this relationship is that fluids which are flammable are reactive and thereby has a short atmospheric lifetime, which in turn means that they will have a short time to contribute to global warming, which is typically measured in a 100-year perspective.

A research need for the future is thus how to mitigate the risks associated with flammable refrigerants. One way, already mentioned several times in this report, is to decrease the charge of refrigerant. However, other methods related to how we design the systems and the safety measures around the systems must also be developed. This could be related to identifying leaks at an early stage and disconnecting any electric devices which could be potential ignition sources, high power ventilation of the location of the heat pump, or using inert gas for the volume around the system. Such research is necessary as a basis for new norms and standards concerning flammable refrigerants. Without clear regulations for the design, flammable refrigerants will have a difficulty of penetrating the market.

Another research area which needs more attention is the effect on system performance of using refrigerant blends. Several such blends are being suggested by the chemical industry. Many of the blends are non-azeotropic and thus have large glides. The systems have to be designed to avoid the disadvantages of the glides, and, if possible, instead benefit from the glides. Using non-azeotropic blends also requires good control of the composition of the refrigerant, both at the first charging and during service or top-up. If the concentrations of the different constituents are different than expected, this may lead to malfunctioning of the system.

Finally, the effect of release of synthetic low-GWP refrigerants requires constant monitoring and research as the decomposition products may have unexpected negative effect on the environment.