HPT Annex 54

Heat Pump Systems with Low GWP Refrigerants

Task 2: Design guidelines for optimization of components and systems of Low GWP heat pumps

Country Report

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Executive Summary

National and international regulations aim to reduce the greenhouse gas emissions caused by the release of refrigerants to the atmosphere by defining restrictions for refrigerants with high GWP-values in certain applications and by limiting the total CO_2-equivalent of refrigerants on the market through a phase-down concept. However, currently most used refrigerants have GWPs in the magnitude of 1000 or even more. While in other sectors like refrigerators or freezers, natural refrigerants as e.g. R290 and R600a can easily be used, for heat pumps the situation is different, also due to higher capacities and higher refrigerant charges.

Starting with the finding from Task 1 that – for now – R290 is the only “real” Low GWP refrigerant (GWP < 150) for common domestic heat pump applications, the differences in physical and chemical behavior of R290 compared to R410A are investigated. Based on the findings, the implications of changing the refrigerant from R410A to R290 and the necessary cycle modifications are investigated using the Modelica-based TIL Suite package in Dymola. Furthermore, based on legislation and standards, this research is completed by an analysis of necessary precautions, especially when using flammable refrigerants.

The investigations show that larger modifications have to be taken into consideration, e.g. R290 requires larger piping diameters due to the lower volumetric refrigeration capacity and the sizing of the expansion device needs to be verified. Performance calculation results show that in a wide range of temperature lifts, R290 has a higher COP than R410A, especially for higher temperature differences, making it especially an energy efficient alternative for unrefurbished or partly refurbished buildings as well as for domestic hot water heat pumps or various non-household applications. Moreover, the behavior of the system changes when using R290 instead of R410A, e.g. the compressor’s volumetric flow rate needs to be adjusted and the refrigerant mass in the evaporator and condenser is much lower.

R290 seems therefore to be a promising alternative not only for systems at lower temperatures as refrigerators and air conditioners, but also for a wide range of applications for heat pumps. As for the European Union, Regulation 517/2014 requires a significant drop in the total GWP of all refrigerants brought on the market, the replacement of R410A with a GWP of 2088 by R290 with a GWP of 3 can be the decisive step towards fulfilment of these requirements. Nevertheless, questions regarding high temperature systems and systems with high refrigerant loads in sensible areas still have to be investigated. In this respect, techniques which reduce the refrigerant load for given use cases will gain importance.

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1. Introduction

In the Austrian Task 1 Report it was found that refrigerants with a GWP of 2000 or even more still have a huge market share in the field of residential heat pumps. The only refrigerant with a GWP below 150 that is currently implemented in heat pumps for residential purposes is R290 – propane, which is flammable.. For flammable refrigerants, special restrictions and systemic requirements apply which are not feasible in every case.

Therefore, for Task 2 it is necessary to define exactly in which cases an elevated flammability is problematic and when this can be tolerated. In cases where flammability class 2L, 2 or 3 is not tolerable, a higher GWP threshold has to be considered.

The aims of this report are accordingly:

- Definition of safety-relevant and design requirements for heat pumps using "low-GWP" refrigerants
- Development of design guidelines based on the requirements
- Investigation of the necessary modifications when changing the refrigerant from R410A to R290

"Low-GWP" refrigerants differ from currently used refrigerants in their thermodynamic properties. Special properties regarding heat transfer and pressure drop have to be taken into account in the design of heat exchangers, and different behaviour during compression in the compressor and expansion in the throttle element has to be considered.

Based on the thermodynamic properties and necessary safety requirements of the identified "low-GWP" refrigerants investigated in Task 1, design guidelines will be developed with regard to the changed requirements for the components of heat pumps.

Based on these requirements, system configurations to be defined will be simulated and optimised with regard to the expected refrigerant charge and the design guidelines will be evaluated on the basis of the simulation results.

2. Design and safety requirements when utilizing R290

This chapter aims to form the basis for the design guidelines by investigating the design and safety requirements of the refrigerant R290. Section 2.1 compares the thermodynamic properties of R290 with the commonly used refrigerant R410A and covers the aspects flammability as well as oil and material compatibility. To conclude this chapter, relevant safety standards are listed in section 2.2.

2.1 Comparison of Refrigerants: R410A vs. R290

R410A is commonly used in residential heat pumps and air conditioning systems. It is a near-azeotropic mixture of 50% R32 and 50% R125 with a temperature glide of 0.1 K at 1 bar. Categorization in safety class A1 (according to ISO 817 [1]) enables the utilization without further safety measures. Due to the high global warming potential (GWP) of 2088, R410A is affected by the phase out covered by the Kigali Amendment to the Montreal protocol [2].

R290 is a natural refrigerant with a negligible GWP of 3. Despite its promising thermodynamic properties, safety measures due to the high flammability (safety class A3) need to be considered. According to Bitzer [3], refrigeration systems with R290 have been in operation worldwide for many years, primarily in the industrial sector. Meanwhile, R290 is also used in smaller compact systems (air conditioners, heat pumps) with lower refrigerant charge. There is also a growing trend towards its use in commercial refrigeration systems and chillers.

2.1.1 Thermodynamic properties

Figure 2.1 (left) depicts the saturation pressure of R290 and R410 as a function of the dew point temperature. It can be seen that the pressure level of R290 is significantly lower than for R410A. This is also reflected in the different critical values: p_{crit} = 42.5 bar, t_{crit} = 96.7 °C for R290 and p_{crit} = 49 bar, t_{crit} = 71.3 °C for R410A. The comparison of the t/h-diagrams of both refrigerants in Figure 2.1 (right) shows the higher enthalpy of vaporization of R290 and the already mentioned difference in critical temperatures.

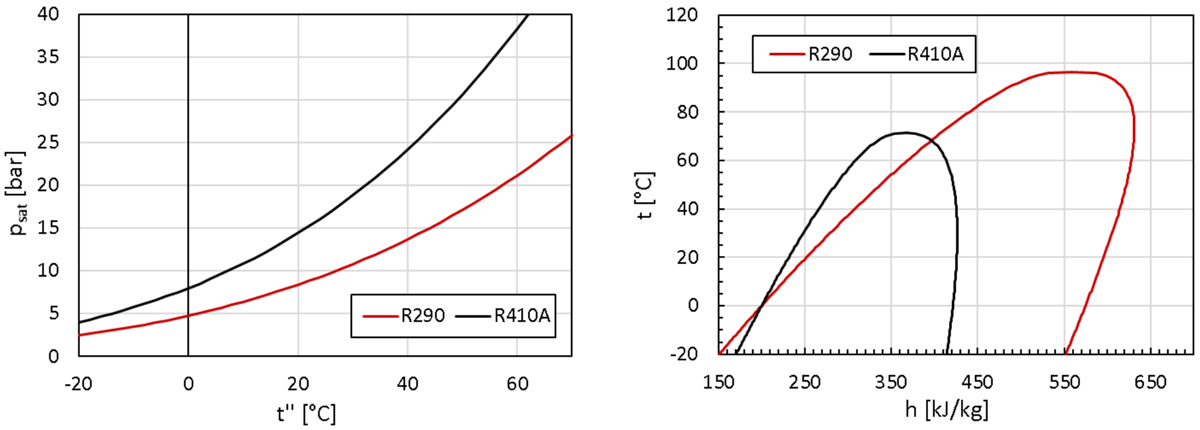


Figure 2.1: Saturation pressure as a function of the dew point temperature (left) and t/h-diagrams (right) of the refrigerants R290 and R410A. Property data extracted from [4]

Taking the refrigerant density at suction state into consideration, the volumetric refrigeration capacity can be evaluated according to Eq. 1 with the vapor density (ρ'') and the enthalpy of vaporization ($h'' - h'$).

$$q_0 = \rho'' \cdot (h'' - h')|_{p=const.} \qquad \text{Eq. 1}$$

Although the enthalpy of vaporization of R290 is significantly higher, the lower density leads to a lower volumetric refrigeration capacity as can be seen from Figure 2.2.

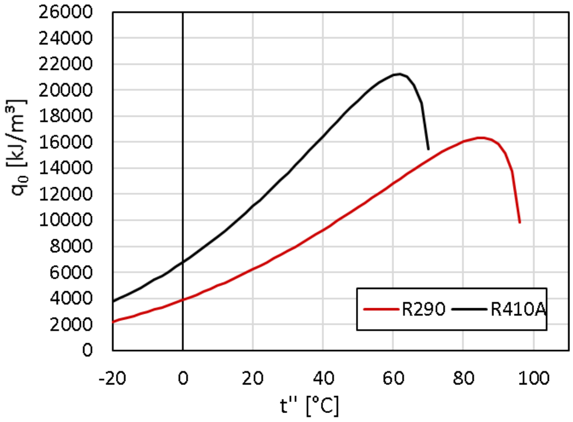


Figure 2.2: Volumetric refrigeration capacities of R290 and R410A as functions of the dew point temperature, property data extracted from [4]

In order to compare the characteristics regarding heat transfer and pressure drop, figures of merit can be derived from known correlations. The Dittus-Boelter correlation for single-phase heat transfer (α_{SP}) according to Eq. 2 can be transposed (Eq. 3) that the figure of merit for single-phase heat transfer (F_α) in Eq. 4 is only depending on the refrigerant properties Prandtl number (Pr), thermal conductivity (k) and dynamic viscosity (μ). Re denotes the Reynolds number, d the diameter of the considered tube and G the refrigerant mass flux.

$$\alpha_{SP} = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot k \cdot d^{-1} \qquad \text{Eq. 2}$$

$$\alpha_{SP} = F_\alpha \cdot G^{0.8} \cdot d^{0.2} \qquad \text{Eq. 3}$$

$$F_\alpha = 0.023 \cdot Pr^{0.4} \cdot k \cdot \mu^{-0.8} \qquad \text{Eq. 4}$$

A similar approach can be followed, starting from the pressure drop (Δp_{SP}) in turbulent single-phase flow according to Blasius (Eq. 5), where L denotes the length of the considered tube. Transposing (Eq. 6) yields the figure of merit for single-phase pressure drop ($F_{\Delta p}$) in Eq. 7 which only depends on the refrigerant properties specific volume (v) and dynamic viscosity (μ).

$$\Delta p_{SP} = \frac{0.3164}{Re^{0.25}} \cdot \frac{L}{d} \cdot \frac{G^2 \cdot v}{2} \qquad \text{Eq. 5}$$

$$\Delta p_{SP} = F_{\Delta\mathrm{p}} \cdot 10^{-6} \cdot \frac{G^{1.75} \cdot L}{d^{1.25}} \qquad \text{Eq. 6}$$

$$F_{\Delta\mathrm{p}} = 0.1582 \cdot 10^6 \cdot v \cdot \mu^{0.25} \qquad \text{Eq. 7}$$

These figures of merit are calculated for the liquid and the vapor phase separately. Heat transfer is investigated in Figure 2.3 (left) and pressure drop in Figure 2.3 (right). Regarding heat transfer, the higher values for R290 indicate better heat transfer characteristics as R410A. This can be fostered through the decrease in heat transfer area needed for a certain load. Considering pressure drop, the higher values for R290 indicate a higher pressure drop which makes a larger diameter for the refrigerant piping necessary.

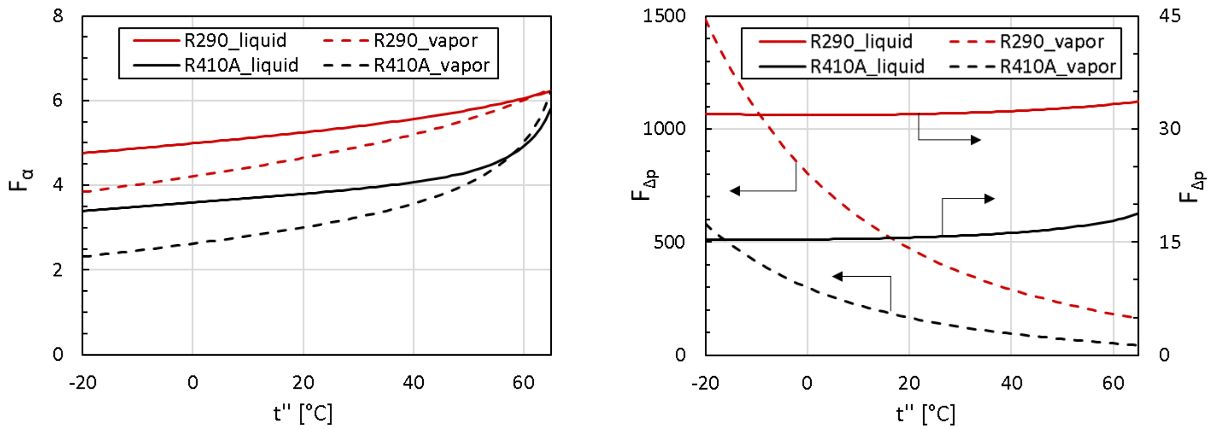


Figure 2.3: Comparison of heat transfer characteristics (left) and pressure drop characteristics (right) as functions of the dew point temperature of R290 and R410A. Property data extracted from [4]

2.1.2 Flammability

The biggest difference between R290 and R410A lies in the flammability of R290, leading to an ASHRAE safety classification of A3 compared to A1 for R410A. Further properties regarding flammability can be found in Table 2-1.

Table 2-1: Properties of R290 concerning flammability. Data extracted from [5], [6], [7]

Lower Flammability Level (LFL)	2.1 $\%_{Vol}$ [6] 0.038 kg/m³ [5,6]	Ignition Temperature	470 °C [5]
Upper Flammability Level (UFL)	9.5 $\%_{Vol}$ [6] 0.171 kg/m³ [6]	Temperature class	T1 [7]
Density @ 1 bar, 15 °C	1.874 kg/m³ [7]	Ignition protection class	IIA [7]

2.1.3 Oil and material compatibility

The compressor manufacturer Bitzer provides technical information for the use of R290 in their semi-hermetic compressors [8], giving the following information: R290 is uncritical in combination with common metals and elastomers. Zinc and alloys with magnesium contents exceeding 2% should be avoided. The extraordinarily high solubility in lubricants reduces the oil viscosity in the compressor. This may lead to a strong outgassing effect in the oil sump and insufficient lubrication upon reduction in pressure, foaming and reduced performance and strong wear. Because of that, oil with sufficiently high viscosity should be used, the refrigerant charge should be as low as possible and the installation of an oil sump heater is recommended. Sufficiently high suction gas superheat needs to be ensured, the compressor discharge temperature must be at least 20 K higher than the condensation temperature for reciprocating compressors and 30 K for compact screw compressors.

2.2 Relevant safety standards and regulations

The following sections summarize the most relevant regulations and standards, sectioned into European legislation and agreements (2.2.1), international standards (2.2.2) and national standards (2.2.3).

2.2.1 European legislation and agreements

The European Union counts with three important regulations in context with flammable refrigerants:

F-Gases Regulation

Regulation (EU) No 517/2014 of the European Parliament and of the Council of 16 April 2014 [9] essentially describes the gradual ban on placing equipment and HFC refrigerants on the market depending on the refrigerant's GWP. Furthermore, the regulation initiated the phase down of HFC refrigerants and defines the requirements for labelling and periodic testing of refrigeration equipment with HFC refrigerants. This regulation does neither cover natural refrigerants nor HFOs but covers a variety of synthetic A2L refrigerants such as R32.

CLP Regulation

Regulation (EC) No 1272/2008 of the European Parliament and of the Council of 16 December 2008 (CLP Regulation for short) [10] defines the classification, labelling and packaging of substances and mixtures and is based on the Globally Harmonised System of Classification and Labelling of Chemicals (GHS). This ensures that the chemicals' hazards are labelled in a globally harmonized way.

REACH Regulation

Regulation (EC) No 1907/2006 of the European Parliament and of the Council of 18 December 2006 [11] regulates the registration, evaluation, authorisation and restriction of chemicals (REACH). It established an European Chemicals Agency and aims at protecting human health and the environment. To this end, a central register has been set up with data on chemical substances marketed in Europe. The core element of this is the safety data sheet (MSDS).

ADR

The "European Agreement concerning the International Carriage of Dangerous Goods by Road" of the United Nations Economic Commission for Europe (UNECE) [12] describes specifications such as markings, packaging, prohibitions on mixed loading and separation as well as loading and handling regulations, load securing and tunnel restrictions depending on the hazardous substance.

The United Nations Economic Commission for Europe (UNECE, set up in 1947) is one of five regional commissions of the United Nations and aims at promoting pan-European economic integration.

2.2.2 International standards

Standards set benchmarks for best practice (the "state of the art"), compliance with which is highly advisable. There are different types of standards, which are hierarchically ordered.

This list of types of standards includes some exemplary standards related to (flammable) refrigerants:

- harmonized, mandated standards, e.g.
    - EN 378, Part 2 [13]
    - EN 14276 Parts 1 [14] and 2 [15]
- product standards, e.g.
    - IEC 60335-2-40 [16]
- process standards, e.g.
    - EN 378, Parts 1 [5], 3 [17] and 4 [18]

Standards can be elevated to the status of law by the legislator ("harmonized standards") and in this case must be complied with. Like laws and regulations, such harmonized, mandated standards have been prepared on behalf of the European Commission and are publicly available. Harmonised standards support fundamental requirements of EU directives and thus contribute to the presumption of conformity. This means that if such standards are complied with, there is a presumption that the legal requirements are met.

The most commonly known standards dealing with flammable refrigerants are listed here:

- IEC 60335-2-11 “Tumble dryers” [19]
- IEC 60335-2-24 “Domestic refrigerators” [20]
- IEC 60335-2-40 “Household heat pumps” [16]
- IEC 60335-2-89 “Commercial refrigeration” [21]
- ISO 13043 “Automotive air conditioning“ [22]
- ISO 20854 “Refrigerated containers“ [23]
- EN 378, Parts 1-4 “Refrigerating systems and heat pumps – safety and environmental requirements” [5,13,17,18]

2.2.3 National legislation

In Austria, numerous laws and regulations deal with safety aspects regulating equipment with (flammable) refrigerants.

Refrigeration Plant Regulation (KAV) [24]

This regulation applies to local facilities where labour is regularly carried out (minimum refrigerant charge of 1.5 kg). The regulation includes requirements for the installation and operation of refrigerant plants and prescribes periodic inspections.

Dual Pressure Equipment Regulation (DDGV) [25]

This regulation is the implementation of Austria’s Pressure Equipment Act [26] and describes the requirements for the manufacturing of pressure equipment. The manufacturing module and category can be determined via a “pressure-volume product” or a “nominal diameter-pressure product”. This is necessary for the CE marking and the declaration of conformity, which are obtained by means of a conformity assessment and a risk assessment.

Pressure Equipment Monitoring Regulation (DGÜW-V) [27]

This regulation is also an implementation of the Pressure Equipment Act [26]. It describes the requirements for an initial operational test as well as recurring tests depending on the “pressure-volume product” or a “nominal diameter-pressure product” and the fluid, i.e. the refrigerant.

Industrial Gas Regulation (HFC-FKW-SF6-V) [28]

This regulation regulates the placing on the market and the use of stationary equipment and installations by specifying charge limits.

Workplace Regulation – ASTV [29]

This regulation defines general provisions for workplaces, requirements for work spaces, sanitary precautions and social services, first aid, fire protection etc.

Limit Values Regulation 2021 (GKV 2021) [30]

This regulation applies to workplaces, construction sites and external workplaces as defined by Austria’s Employee Protection Act [31] and specifies the maximum workplace concentration of substances.

Explosion Protection Regulation 2015 (ExSV 2015) [32]

This regulation applies to

- equipment intended for its use in potentially explosive atmospheres
- protective systems intended for its use in potentially explosive atmospheres
- components of equipment and protective systems intended for its use in potentially explosive atmospheres
- safety equipment, monitoring and control equipment for the use outside potentially explosive atmospheres, but which contribute to the safe operation of equipment and protective systems with regard to explosion risks

Regulation on Explosive Atmospheres (VEXAT) [31]

This regulation describes protective measures for workplaces, construction sites and external workplaces with reference to Austria's Employee Protection Act.

3. Design Guidelines

Starting from the thermodynamic and safety specific requirements when utilizing R290, this chapter summarizes design guidelines including possible refrigerant cycle adaptions and strategies for refrigerant charge reduction.

3.1 Design criteria arising from thermodynamic and safety specific requirements

The most significant differences between R290 and R410A are the high flammability of R290 (safety class A3) compared to the non-flammable R410A (A1) and the significantly lower volumetric refrigeration capacity of R290 (see Figure 2.2). In order to serve the same heating capacity, a heat pump utilizing R290 requires a higher volumetric flow of refrigerant, thus a compressor with a larger displacement is needed. Additionally, the diameters of the refrigerant piping need to be larger if similar flow velocities are targeted.

Due to the flammability of R290, refrigerant charge limits are in place for certain applications as listed in section 3.1.1. Measures due to flammability are summarized in section 3.1.2. Section 3.2 compares different cycle designs for R290 regarding the possible reduction of compressor displacement and Seasonal Coefficient of Performance (SCOP) according to EN 14825 [33] for new and unrefurbished multi-family-houses.

3.1.1 Charge limits according to international regulations

IEC 60335-2-40 Household and similar electrical appliances – Safety; Part 2-40: Particular requirements for electrical heat pumps, air-conditioners and dehumidifiers

This standard covers factory-made refrigerators such as air-to-air heat pumps and air conditioners, air-to-water heat pumps and chillers. The upper limit for the refrigerant charge is 5 kg for R290 (A3) and 80 kg for R32 (A2L). The actual refrigerant charge limits according to IEC 60335-2-40 [16] depending on whether the location of the heat pump or refrigeration unit is outdoors or indoors. Outdoors, there are no special technical requirements regarding the refrigerant charge. If the heat pump or refrigeration unit is located indoors, there are certain requirements for the floor area, the room volume and the ventilation of the room or the device itself. For A2L refrigerants, the refrigerant charge limit can be increased by applying additional safety measures [34].

EN 378 Refrigerating systems and heat pumps — Safety and environmental requirements

The calculation of the refrigerant charge according to EN 378 only applies to refrigerating systems and refrigerators that are not covered by existing harmonized product standards. This means that when refrigerating systems and refrigerators, declared according to harmonized standards IEC 60335-2-40 [16] or IEC 60335-2-89 [21], are placed on the market, the charge limits are preferentially calculated according to IEC 60335-2-40 [16] or IEC 60335-2-89 [21] instead of calculating them according to EN 378 [34].

The charge limits in EN 378, part 1 [13], are determined by a refrigerant safety assessment defined in ISO 817 [1], the access category, the system classification (both defined in EN 378 [5]) and the installation location and refers to the largest refrigerant charge per refrigeration circuit and not for the total installed quantity in all refrigeration circuits [34].

The charge limit determination according to EN 378 is equivalent to a risk assessment. In EN 378, Part 1 [5] Annex C, tables are shown that can be used to identify which solutions offer the highest allowable charge volume with the lowest possible risk [34].

Further standards defining charge limits for other refrigeration applications:

- IEC 60335-2-11 (tumble dryers) [19]
- IEC 60335-2-24 (refrigerating appliances, ice-cream appliances and ice makers) [20]
- IEC 60335-2-89 (commercial refrigerating appliances) [21], defines upper limits of refrigerant charge, e.g. 500 g for R290 or 1200 g for R32

3.1.2 Measures due to flammability

Safety classification

If a chemical product is flammable or has other hazards (e.g. toxicity), it is a legal requirement to provide information about this hazard in a safety data sheet (MSDS). The REACH regulation (Registration, Evaluation, Authorization and Restriction of Chemicals) [11] contains rules according to which the supplier of a chemical product must include a safety data sheet (MSDS). Here, the user can see which measures are required to limit the risk. The purpose of the regulation is to ensure that all chemical substances and mixtures are carefully examined and used safely. According to the CLP Regulation [10] (see section 2.2.1), the classification, labelling and packaging of chemical products is performed in accordance with the above-mentioned GHS (Globally Harmonized System of Classification and Labelling of Chemicals). In the CLP Regulation [10], flammable refrigerants belong to flammable gases of category 1A, 1B or 2.

The refrigeration industry classifies flammable refrigerants according to ISO 817 [1] which is the basis for the risk assessment of refrigerant charge in refrigerating systems. In both the general process standard EN 378 and the product standards IEC 60335-2-24, -40 and -89 [20,21,35], the classification according to ISO 817 and the maximum permissible charge of flammable refrigerants are specified.

Refrigerants are classified according to ISO 817 [1] into safety classes. Non-toxic flammable refrigerants are differentiated into three classes A2L, A2 and A3 according to the following criteria:

- Class 3 means flammable refrigerants with a combustion enthalpy > 19 MJ/kg or a lower explosion limit of < 100 g/m³.
- In class 2 there are refrigerants with low flammability, with a combustion enthalpy < 19 MJ/kg and the lower explosion limit > 100 g/m³.
- In class 2L the max. flame spread is < 10 cm/s.
- Non-flammable refrigerants are in class 1.

Health hazards and respective precaution measures

Risk assessment to identify and evaluate hazards and take appropriate action to avoid them is crucial when working with (flammable) refrigerants. The most important source of information for all hazards is the material safety data sheet (MSDS) for the refrigerant and its decomposition products [34].

<u>Refrigerant leaks</u>

It is important to equip the refrigeration technician with a personal gas detector. Flammable refrigerants in safety classes A2L, A2 and A3 are odourless, colourless and tasteless (except for R1270). A large leak in a poorly ventilated room can displace air without being detected by the human senses. Regardless of the refrigerant, inhalation of refrigerants should always be avoided as it can be harmful to health. Special attention must be paid to synthetic HFO, as released HFO decomposes within one to two weeks into toxic chemicals. Leaking gas that is flammable, but does not burn at the leakage point immediately can be more problematic than burning gas because it cannot be easily detected and therefore can ignite at any moment after having been distributed in a wider area. The immediate action is to eliminate all potential ignition sources. Water should never be used to cool tanks containing flammable refrigerants. As flammable refrigerants have low boiling points, adding water increases evaporation and emissions. A leak of liquid refrigerant is more problematic than of a gaseous refrigerant, because with liquid refrigerant, a larger quantity escapes more quickly [34].

Fire

If leaked gas caught fire, it is advisable not to extinguish the fire but to let it burn out so that the flame can be controlled. When the fire is extinguished, there is a risk that the odourless and colourless gas will sink to the floor and spread outward, so that it cannot be easily detected. The flame should be extinguished only if this can be done by closing a valve. This requires proper fire safety training and practice, as well as knowledge of the system. A gas cylinder in upright position does not present a problem, because the pressure is automatically relieved via its safety valve. When a refrigerant cylinder is stored horizontally, the safety valve may be blocked by liquid refrigerant, which poses a risk of bursting. It may be necessary to relocate the gas cylinder or evacuate the area. A respirator with fresh air supply should be used [34].

Frostbite

Leaks of refrigerants in both, gas phase and liquid phase can cause very low temperatures. It is therefore important to wear clothing that provides good protection against cold. Direct splashes of refrigerant, liquid or gas vapor can cause frostbite. Eyes can be permanently damaged. Therefore, eyes should be immediately flushed with eyewash or clear water for at least 10 minutes, holding the eyelids apart.

Injured persons should be moved to a warm room, clothing and shoes that prevent blood circulation should be removed. Frozen skin must be constantly rinsed with lukewarm water and a physician should be notified immediately [34].

Further health hazards

EN 378, Part 2, Annex D lists hazards and hazardous situations that are significant in connection with work on refrigerating systems and require risk reduction measures. The most common injuries when working on refrigerating systems are caused by falling accidents, followed by cutting and crushing injuries and splashes of refrigerant on the skin or in the eyes [34].

Personal protective equipment and proper tools

The CLP Regulation [10] classifies most refrigerants that belong to the safety classes A2L, A2 and A3 as extremely flammable gases. If there is a risk of creating an explosive atmosphere, all equipment and tools used for the work must be approved for the explosion hazard concerned and bear the CE mark. All equipment and tools used by refrigeration engineers should be able to be used inside and outside explosion hazard zones [34].

Working with A2L refrigerants

Low-energy sparks (e.g. due to electrostatic switches) are unlikely to ignite A2L refrigerants. Even if they did, the refrigerant will burn with little or no flame spread. Protective equipment and tools that cannot produce a strong spark (short circuit) can therefore be used when working on refrigerating systems with A2L refrigerants. Hence, gloves, safety shoes, work clothing and protective eyewear does not have special requirements [34].

Working with A2 and A3 refrigerants

Gloves, safety shoes and work clothing must be electrostatically dissipative while protective eyewear does not require special requirements. Requirements for tools can be found in Table 3-1 [34].

Table 3-1 shows the recommended tools for working at refrigerating systems with refrigerants of the safety classes A2L, A2 and A3.

Table 3-1: Recommended tools for safety classes A2L, A2 and A3 [34]

Tool	A2L	A2 and A3
Personal gas detector	approved for the refrigerant	
Manometer bridge with hoses	approved for the refrigerant	
Vacuum pump	explosion proof, non-sparking switch and motor, 5m cable	
Fan	explosion proof, plastic fan blades, explosion proof motor and switch, 5m cable	
Refrigerant scale for systems under 150 g refrigerant charge	scale with +/- 1 g accuracy	
Refrigerant scale for systems over 150 g refrigerant charge	scale with +/- 10 g accuracy	
Refrigerant recovery unit	explosion proof, non-sparking switch and motor, 5m cable	
Leakage detector	approved for the refrigerant	
Electrical measuring instruments	conventional	explosion proof
Manual hand tools	conventional	non-sparking
Electric hand tools	explosion proof	
Refrigerant cylinders	special threads	
Large refrigerant cylinders	red bottle neck left hand thread	
Refrigerant cylinders for disposal	red bottle neck labelled with danger sign, field for declaration number	

3.2 Cycle adaptions

Refrigeration circuit design has an impact on the heat pump's efficiency and further crucial aspects. In this section, results from comparing the following three R290 refrigeration circuits with thermodynamic simulations, performed within IEA HPT Annex 50, are presented, see Figure 3.4 [36].

Single stage

The single stage refrigeration cycle represents the simplest refrigeration circuit design and control (see Figure 3.1).

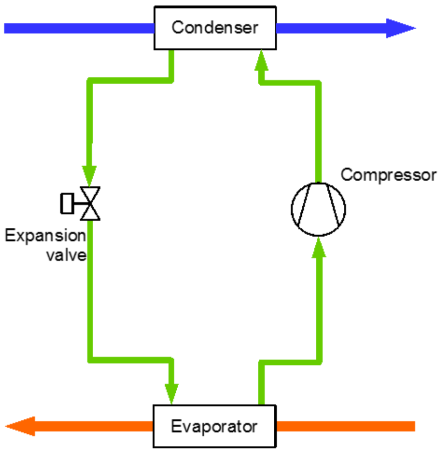


Figure 3.1: Single stage refrigeration circuit [36]

iHX (internal heat exchanger)

The suction gas heat exchanger superheats the suction gas while subcooling the liquid refrigerant from the condenser, before the suction gas enters the compressor (see Figure 3.2). As R290 requires significant superheating for a safe compressor operation, the use of iHX is a good option.

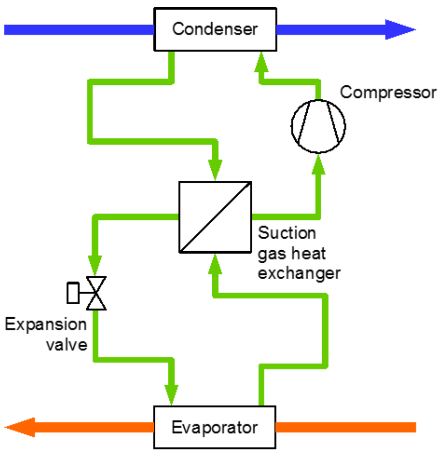


Figure 3.2: Refrigeration circuit with iHX [36]

EVI (enhanced vapour injection)

Refrigeration circuits with EVI (see Figure 3.3) are equipped with a two-stage compressor with an injection port, a second expansion valve and an economizer. One part of the refrigerant is expanded in the first expansion valve to evaporating pressure and another part is expanded in the second expansion valve to intermediate pressure level. The refrigerant fraction at intermediate pressure level is evaporated in the economizer and enters the compressor via the injection port.

Depending on the present pressure ratio, the specific compressor work is reduced. The injection into the compressor on intermediate pressure level leads to a decrease of the compressor's discharge temperature, which is beneficiary to the lubricant's lifespan [36].

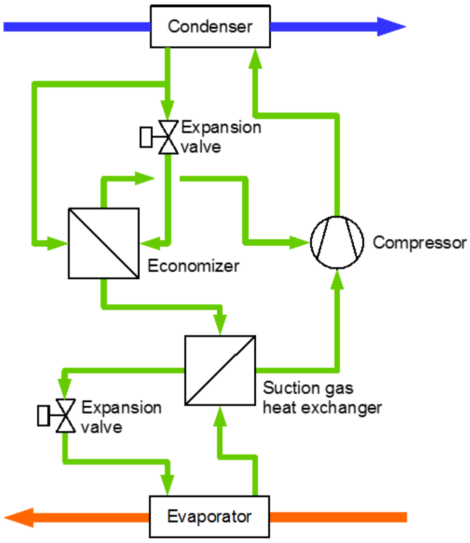


Figure 3.3: Refrigeration circuit with EVI [36]

Modelica models

Within the course of IEA HPT Annex 50, the above-mentioned refrigeration circuits were compared based on results from numerical simulations, the simulation models were created within the Dymola/Modelica[1] environment. The parameterisation and validation of the models were done by using results from experimental investigations performed within the European research project GreenHP [37]. The main model (see Figure 3.4) can be easily adapted to simulate all three refrigeration circuits described above [36].

[1] See https://www.3ds.com/de/produkte-und-services/catia/produkte/dymola/

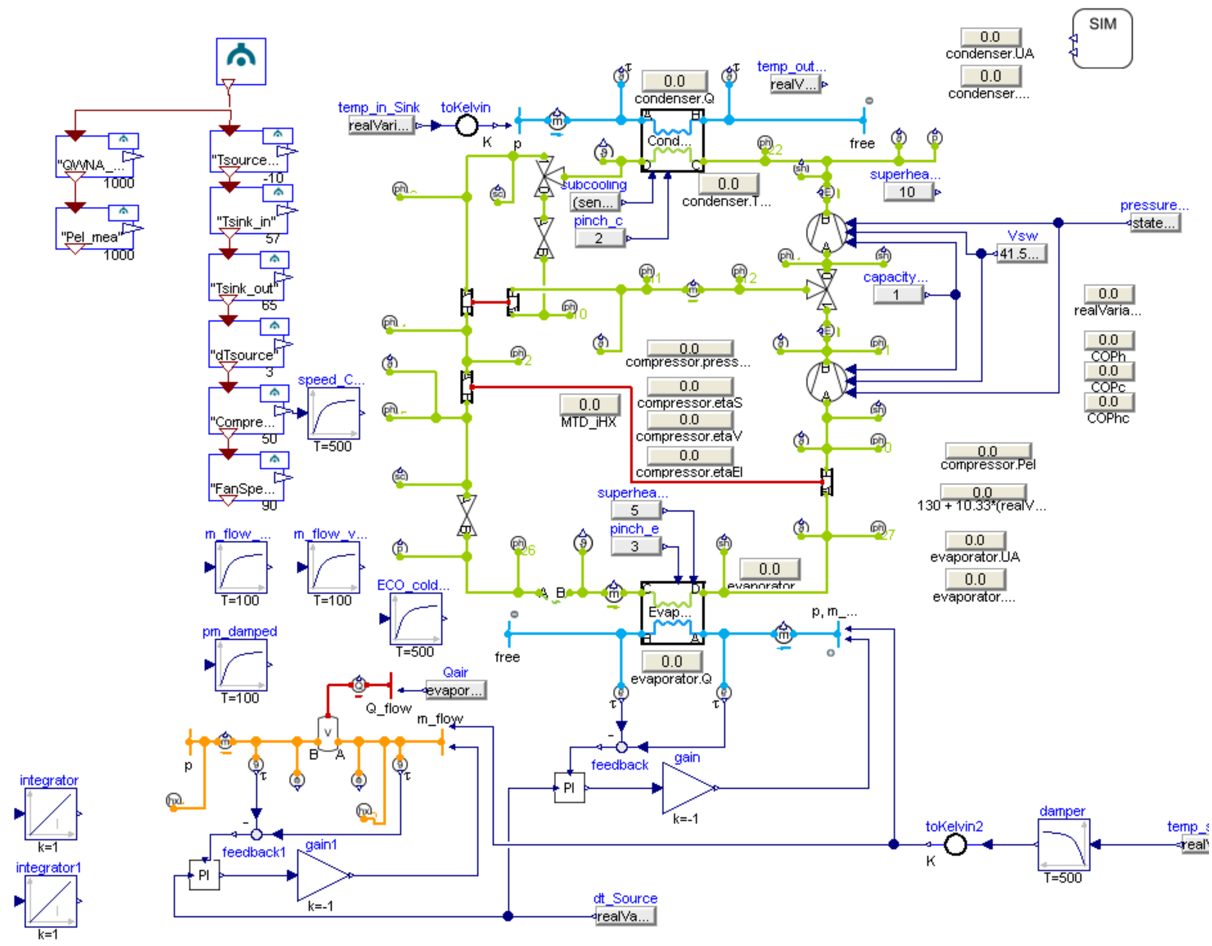


Figure 3.4: Numerical model in Dymola/Modelica [36]

The three refrigeration circuit variants and sub-variants regarding different heating capacities/temperatures (due to different building standards) and further refrigerant types were simulated in different operating points in accordance with EN 14825 [33].

Beside other findings, the results from these investigations indicate that the application of R290 achieves higher COPs than refrigeration circuits with the other investigated refrigerants only in buildings with higher heating supply temperatures (e.g. unrefurbished buildings). In buildings with lower heating supply temperatures (e.g. refurbished buildings), the other investigated refrigerants show a better performance than R290. Another finding is that the application of enhanced vapor injection (EVI) is beneficial in all investigated cases.

The following comparisons in Figure 3.5 show the reduction of the compressor displacement for different refrigerants (left for newly constructed MFH[2] and right for unrefurbished MFH). The basis for comparison is always the worst candidate in terms of compressor displacement, i.e. in the new MFH this is the refrigerant R290 (both with iHX and as a single stage) and in the unrefurbished MFH it is R1234zee. These two configurations require the largest compressor displacement. All other values refer to these values. In the new building, R410A requires 50% less compressor displacement than R290. In contrast, R290 is the best candidate for old buildings, with a reduction in compressor

[2] The investigated buildings were multi-family-houses (MFH)

displacement of over 50% compared to R1234zee. Potential saving in refrigerant charge correlates with the reduction in compressor displacement.

For unrefurbished buildings with higher heating supply temperatures (see Figure 3.5, right), R290 provides the highest compressor displacement reduction values with best results for the refrigeration circuit with EVI (above 50 %) [36].

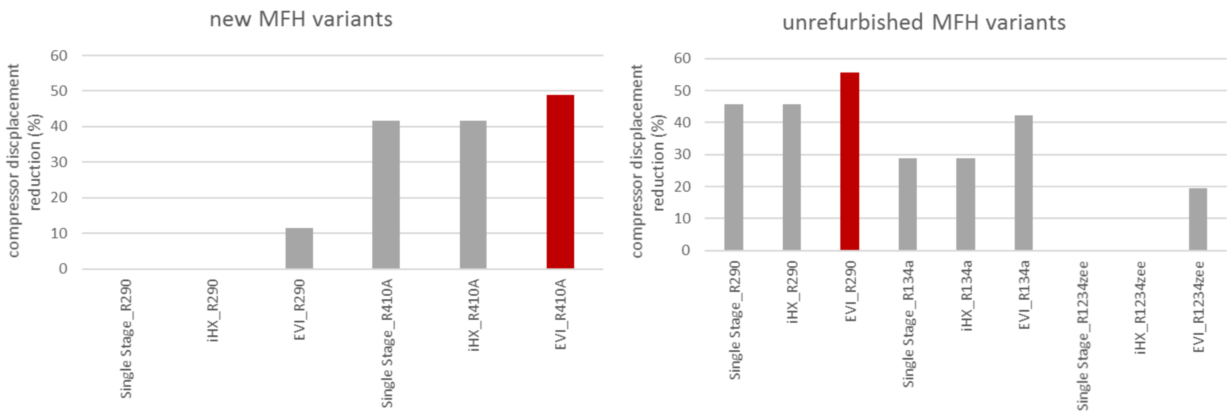


Figure 3.5: Compressor displacement reduction for new buildings (MFH) (left) and for unrefurbished buildings (MFH) (right). The basis is the candidate which achieves the least compressor displacement reduction [36].

Regarding the overall efficiency, the investigations showed for both, new buildings (see Figure 3.6, left) and unrefurbished buildings (see Figure 3.6, right) that R290 provides the highest SCOP values with best results for the refrigeration circuit with EVI (4.35 for new MFH and 3.0 for unrefurbished MFH). The SCOP represents the seasonal efficiency according to EN 14825 and is defined as calculated seasonal performance factor. In contrast to the COP, not only the efficiency of a single operating point is considered, but the entire operating behaviour of a season. [36]

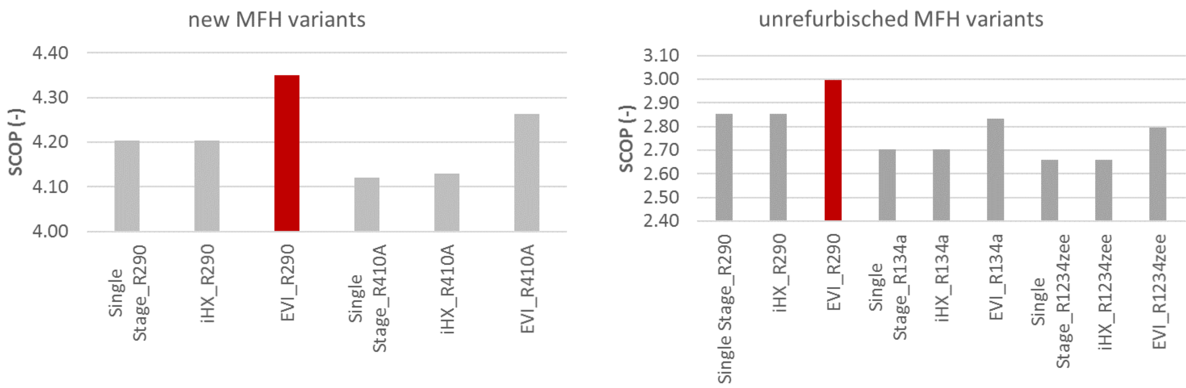


Figure 3.6: SCOP results for new buildings (MFH) (left) and for unrefurbished buildings (MFH) (right) [36]

3.3 Refrigerant Charge Reduction

The application of charge reduction measures helps to reach charge thresholds defined in the regulations listed in section 3.1.1.

3.3.1 General considerations for the main components

Refrigerant charge is reduced by reducing the volume of the refrigerating system, while the parts which contain liquid refrigerants are crucial due to the high density of liquids. Within the GreenHP project, theoretical investigations showed that the lowest theoretically possible refrigerant charge for an air/water heat pump with the refrigerant R290 could be approx. 17 g/kW thermal capacity. With the heat pump prototype, examined in the GreenHP project, a refrigerant charge reduction to approx. 65 g per kW thermal capacity, which is still a very good value, was achieved. The deviation from the theoretical values are caused by several circumstances, e.g. [37]:

- Higher required refrigerant charge during defrost than during normal operation (1900 g instead of 1600 g)
- More refrigerant trapped in the evaporator's headers than expected as it is not perfectly drained (manufacturing issues)
- More refrigerant absorbed in the oil than according to the calculations

The following section presents general considerations regarding refrigerant charge reduction for each main component in refrigeration circuits and experiences gained within the GreenHP project.

Compressor

Conventional compressors in stationary refrigerating systems are usually not optimised regarding refrigerant charge. For safety reasons, compressors usually contain more oil than technically necessary. As refrigerants (particularly R290) dissolve in oil very well, refrigerant charge reduction can be achieved by reducing the amount of compressor oil. In R290 refrigeration circuits, a typical share of refrigerant located in the compressor is approximately a third of the total refrigerant charge. This value can be reduced by utilizing a synthetic PAG-oil with lower solubility. In order to ensure enough oil charge in all operating conditions, the amount of oil in compressors is higher than technically necessary ("safety margin"). Oil circulates through the whole refrigeration circuit. Hence, the oil level in the compressor decreases after commissioning. Depending on the design of the components and piping, oil is transported from the compressor into the refrigeration circuit in larger or minor amounts [38]. The oil charge of the GreenHP prototype's compressor was reduced from 2.5 litres (conventional design) to 1.5 litres. According to simulations, the GreenHP compressor's refrigerant charge might be reduced to approx. 100 g, the actual charge is higher [37].

Condenser

Measures to reduce refrigerant charge in the condenser are avoiding subcooling inside the condenser and reducing the condenser's volume by innovative geometries, e.g. a flat multichannel condenser with aluminium multi-port extrusion tubes (MPE), as used in the GreenHP project. The prototype's condenser consists of such 175 parallel, vertical tubes with narrow openings in between for the water. According to simulations, the GreenHP condenser's total volume is approx. 2 liters which should achieve a refrigerant charge close to only 100 g [38], the prototype's actual charge is higher [37].

Receiver

As a refrigerant charge optimized condenser may not provide any subcooling and as the refrigerant charge in the other components varies with the changing operating conditions, a receiver is mandatory. According to the simulations performed in GreenHP, the difference between the simulated maximum and the simulated minimum charge necessary for other components in the systems was about 250 g. The corresponding design lead to a volume of 0.6 litres [38].

Expansion valve

As the expansion valve has a very small internal volume, it is not subject to refrigerant charge optimisation [38].

Evaporator

For evaporators, similar considerations apply as to condensers. Both are heat exchangers where the main heat transfer is achieved by phase change between liquid and vapor respectively vice versa. The evaporator's volume can also be reduced by using a system with a compressor that tolerates wet vapor suction, as the heat transfer area usually required for super heating, can be omitted. The Green HP prototype's air-charged evaporator is made of aluminium multi-port extrusion tubes (MPE) with optimized headers. The evaporator reaches a large heat transfer area by utilizing waved fins and a high heat transfer value. The evaporator's volume is approx. 10 litres resulting in a theoretically possible refrigerant charge of around 100 g [38]. The actual value is higher [37].

Connecting tubing

The refrigerant charge in the connecting tubing depends on the components' location and distances to each other. Shorter pipes allow the reduction of their diameter while still providing acceptable pressure losses. The length of the liquid line is the most crucial, the length of the suction line is the least crucial. The pipe diameter should be just large enough to avoid significant drop in saturation temperature [38].

3.3.2 Falling Film Evaporator

Falling film evaporation is a technology that has been used for a long time successfully in other technology fields like desalination processes or chemical industry [39]. However, for evaporation of refrigerants in heat pumps it is rather a new and rarely used approach with less experience.

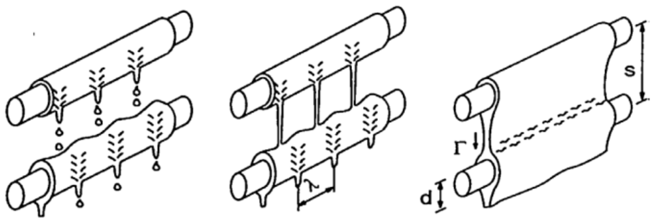
Compared with pool boiling or flooded evaporation, falling film evaporation allows operation with lower temperature differences and less refrigerant [40].

The principle of horizontal-tube falling film evaporators, which is the most common typology, is as follows:

The heat exchangers for falling film evaporators consist of a shell and tube structure [39]. A spraying system distributes the refrigerant from the top over the tube structure. Parts of the refrigerant already evaporate at the top, the rest flows down, parts evaporate there, the rest flows further down, etc. Refrigerant that comes to the lower end is collected and sprayed over the structure again. The tubes are filled with a fluid, which is cooled down via the evaporation of the refrigerant. The heat transfer works via conduction and convection.

By this process a falling film is built. It is important that this film is maintained, which is secured by the correct combination of refrigerant flow and temperature regime [39,40]. If the film breaks down, the system performance is impeded severely [39–41]. This breakdown must therefore be avoided. The refrigerant flow (due to gravitation) can lead to this breakdown as well as the evaporation or even boiling. Also the temperature-dependent surface tension can cause dry patches.

The flow pattern of the refrigerant between the tubes is crucial as it affects the heat transfer coefficient heavily. Different modes were observed and defined. Mainly three modes as depicted in Figure 3.7 exist, ordered by increasing Reynolds number: droplet mode, jet or column mode and sheet mode (least risk of film breakdown and therefore recommended). The film flow rate and the properties of the refrigerant, mainly the Reynolds number of the refrigerant film together with the tube structure, influence the flow pattern. Details are explained e.g. in [39].



a) Droplet b) Column c) Sheet

Figure 3.7: Illustration of the different flow modes [41]

For R134a it was observed that the effect of the tube diameter on the heat transfer coefficient is sometimes positive and in other cases negative, depending on heat flux and film flow rate. For low heat flux, heat transfer rises with saturation temperature, while decreasing at higher heat flux [41].

Dryout, especially in the lowest tube rows, is a severe problem for the performance of the system. This can be avoided by increasing the refrigerant flow or by including a spraying system in a lower part of the tube structure as well [39].

Normally the heat transfer coefficient increases with the refrigerant flow rate. Only if boiling occurs, there is no connection between these two parameters. In addition, the refrigerant itself plays a role. In most cases, the heat transfer coefficient increases with the heat flux as well.

Many different distribution methods for the refrigerant have been analysed. An overview is given in [39], but the advantages and disadvantages for different refrigerants, flows, geometries, etc. have not been investigated in depth for most of them until now.

The main advantages compared to pool boiling evaporation are [39–41]:

- higher heat transfer coefficients
- increase of boiling temperature due to hydrostatic head in pool poiling evaporators is avoided
- lower refrigerant charge
- lubricant can be removed easier
- higher thermodynamic efficiency by lower temperature difference between refrigerant and fluid
- smaller evaporators can be used
- lower thermal inertia → faster response to transient operation

The main disadvantage, beside limited experience, is:

- when the refrigerant film breaks down, heat transfer is significantly reduced
- the system performance depends on many parameters and the system behaviour is – at least with current knowledge and techniques – difficult to control

[40] points out that the differences in results of experimental data, e.g. regarding heat transfer coefficients is still high and therefore a high need of further experiments and measurements can be stated.

Falling film evaporation was studied for a number of refrigerants like R134a (where most literature exists [41]), R22 and R123. However, e.g. for R290 there are very limited research results so far. In [41,42] one of the first investigations of R290 was performed; it was shown for R290 that the film flow rate and the heat flux heavily influence the heat transfer coefficient, in line with results from other experiments with other refrigerants. With rising Reynolds number, the heat transfer coefficient increases. In [41] the effect of optimized tubes (i.e. copper tubes with a special surface structure and optimized diameter and geometry) on the system performance was analysed. It was observed that compared with unoptimized (i.e. mainly smooth) tubes, the heat transfer coefficient could be enhanced by a factor 2.5 to 4.5, depending on the size of the array (larger factor for smaller structures). The heat transfer coefficient as a function of the film flow rate shows a plateau stage for larger film flow rates and a sharp drop stage for smaller film flow rates. For the plateau stage the heat transfer coefficient for R290 is higher than for R134a. Contrary, the performance of R600a is worse than of R134a [42]. Lower tubes are always at higher risk of drying out. Below a certain Reynolds number, this risk increases. The heat flux influences this effect as well: with rising heat flux, this critical Reynolds number increases as well. A rising heat flux is beneficial for the heat transfer in both full-wetting and partial dry-out regimes [42].

Conclusion

The advantages of falling film evaporation compared to state-of-the-art pool boiling are numerous. However, the experience is limited and research results still leave many open questions. The experimental results for R290 are even more limited, but promising. Altogether, research focused more on cooling appliances (refrigeration, air conditioners) than on heat pumps. Considering the need for heat pump systems with reduced overall environmental impact, this technology should be investigated in more detail in future as it allows systems with less refrigerant and higher system performance.

4. Theoretical study of retrofitting R410A systems to R290

This chapter investigates the measures to adapt an R410A air-to-water heat pump for the utilization of R290 by means of simulation. The Modelica-based TIL Suite software package [43] is used to model the heat pump in the Dymola environment.

As a first step, the R410A-baseline system is investigated. In order to show the necessary adaptions due to the different thermodynamic properties, the system is analyzed with only changing the refrigerant. In the final step, the components are optimized for R290 and to reduce the refrigerant charge in order to show the potential of previously discussed measures. Specifications of the system as well as the improved evaporator geometry are based on the results of the European research project GreenHP [37].

4.1 Set-up of the simulation model

The following sections describe the investigated cycle configuration, the used models and assumptions.

4.1.1 Refrigerant circuit

Figure 4.1 depicts the refrigerant circuit of the heat pump. The single-stage circuit is equipped with a high-pressure receiver, the refrigerant superheat of 5 K at the evaporator outlet is controlled by the expansion valve. Figure 4.1 (left) depicts the refrigerant circuit in heating mode. In order to investigate the defrosting of the evaporator, the circuit is reversed as depicted in Figure 4.1 (right).

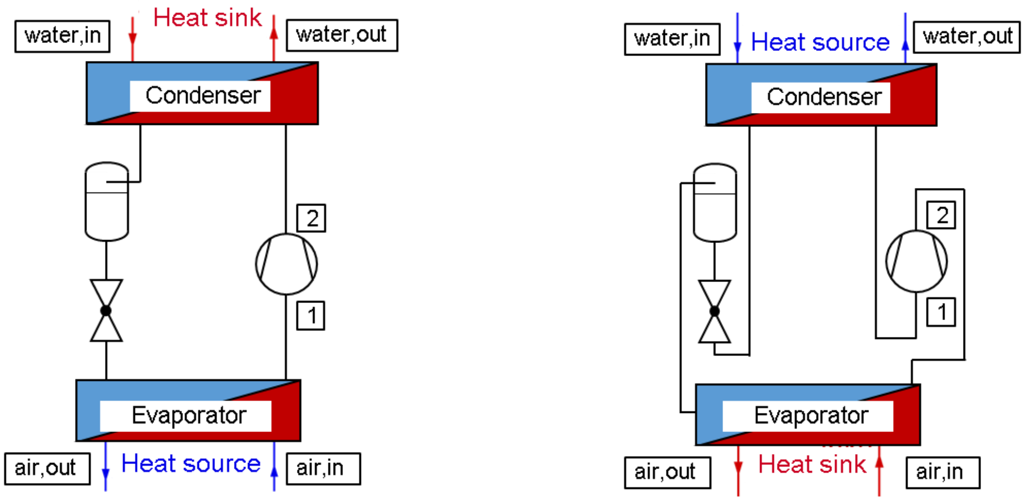


Figure 4.1: Refrigerant circuit of the heat pump in heating mode (left) and defrost configuration (right)

4.1.2 Compressor

The compressor is modelled according to the efficiency approach using the volumetric efficiency (λ_{vol}) as in Eq. 8, the overall isentropic efficiency ($\eta_{is,ov}$) as in Eq. 9 and the inner isentropic efficiency ($\eta_{is,i}$) as in Eq. 10. State 1 refers to the compressor suction port, state 2 to the discharge port. The volumetric efficiency relates the actual refrigerant volume flow, which is calculated via the refrigerant mass flow ($\dot{m}_{ref}$) and the density at suction state ($\rho_{ref,1}$), to the swept volume flow of the compressor ($\dot{V}_{swept}$)

$$\lambda_{vol} = \frac{\dot{m}_{ref}}{\rho_{ref,1} \cdot \dot{V}_{swept}} \qquad \text{Eq. 8}$$

The overall isentropic efficiency relates the power required for an isentropic compression of the refrigerant, which is calculated using the enthalpy values at suction state ($h_{ref,1}$) and after isentropic compression to the high-side pressure level ($h_{ref,2s}$), to the actual electric power consumption of the compressor ($P_{el,comp}$).

$$\eta_{is,ov} = \frac{\dot{m}_{ref} \cdot (h_{ref,2s} - h_{ref,1})}{P_{el,comp}} \tag{Eq. 9}$$

The inner isentropic efficiency relates the enthalpy difference of an isentropic compression to the enthalpy difference of the actual compression.

$$\eta_{is,i} = \frac{h_{ref,2s} - h_{ref,1}}{h_{ref,2} - h_{ref,1}} \tag{Eq. 10}$$

4.1.3 Heat exchangers

The heat exchangers are modeled using the finite-element method. The plate-type heat exchanger with geometries depicted in Figure 4.2 is used as condenser in all configurations. For the plate-type condenser, headers are not considered when calculating the refrigerant mass.

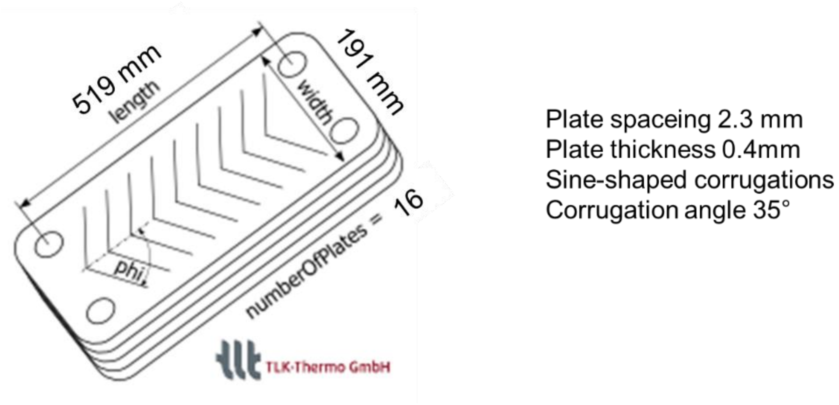


Figure 4.2: Geometry of the used condenser, screen capture from the TIL Suite [43]

Heat transfer is calculated using correlations supplied by the TIL library (based on [44] and [45] for the refrigerant side and [46] for the water side), pressure drop in the condenser is neglected. Geometry and used correlations for the evaporator are described at the sections 4.2.1 for the R410A case and 4.2.4 for the improved R290 case.

4.1.4 Design point and investigated operating conditions

The heat pump systems are investigated at four operating points in heating mode. In order to investigate the defrost operation, the cycle is reversed according to Figure 4.1 (right). Table 4-1 lists the heat source and heat sink temperatures of the investigated operating points, the heating capacity of 30 kW is controlled with the compressor speed within the allowable limits. The sizing of the evaporator enables an evaporating temperature of -17 °C at the design point. Dry air is used as medium for ambient air.

Table 4-1: Investigated operating points

Operating point	Heat source (air) inlet/outlet	Heat sink (water) inlet/outlet
H1 (design point)	-10/-15°C	47/55 °C
H2	-8.3/-9.4 °C	30/35 °C
H3	1.7/0.6 °C	30/35 °C
H4	8.3/6.1 °C	30/35 °C
DF (defrost)	35/30 °C (heating circuit)	-5/40 °C (ambient air)

4.2 Investigation of simulated systems

As a first step, the baseline R410A system is investigated at the defined operating points. In order to demonstrate the necessary changes to the refrigerant cycle for the use of R290, the consequences of changing the refrigerant without adaptions to the refrigerant cycle are presented. As a last step, the improved R290 system is analyzed and the charge reduction measures are evaluated.

4.2.1 Baseline R410A system

The baseline system uses a scroll compressor with a displacement of 24.8 m³/h at 2900 rpm. The used efficiency curves are fitted to values calculated from manufacturer's data [47] according to Eq. 8 to Eq. 10 as depicted in Figure 4.3. The condenser as described in Figure 4.2 is used.

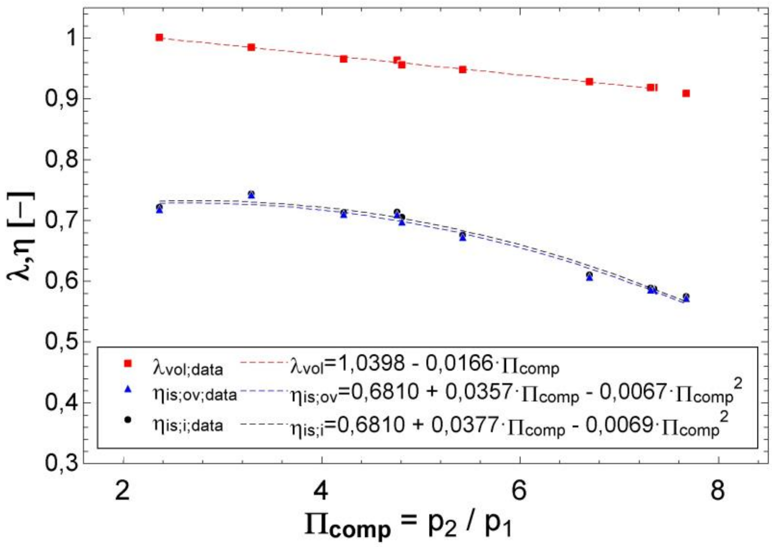


Figure 4.3: Compressor efficiencies calculated from manufacturer's data [47] and used interpolation curves

The evaporator used in this configuration is a plate-and-fin heat exchanger with geometries according to Figure 4.4. To account for the refrigerant mass in distributor and headers, a certain volume filled with liquid refrigerant at low-pressure level is considered. Assuming that the ports of the evaporator are located at the midpoints of the sides, which are 1.5m long, the longest distance a distributor needs to bridge is half the side length. To assure equal pressure drop, all distributor lines are assumed to have the same length. The sum of the cross section areas of all distributor lines is assumed to be equivalent to the suction line cross section area to assure a reasonable refrigerant flow velocity. These assumptions lead to the volume equivalent to 0.75 m of the suction line for the distributors. To account for the headers as well, in total, the volume of 1m of the suction line, filled with liquid refrigerant, is assumed.

A suction line inner diameter of 25 mm leads to flow velocities between 10 and 16 m/s, a liquid line inner diameter of 16 mm leads to velocities between 0.7 and 0.8 m/s.

The following correlations for heat transfer and pressure drop supplied by the TIL suite are used: Refrigerant heat transfer based on [48], [49] and [50], refrigerant pressure drop based on [51], air-side heat transfer based on [52] and fin efficiency based on [53]. Air-side pressure drop is neglected.

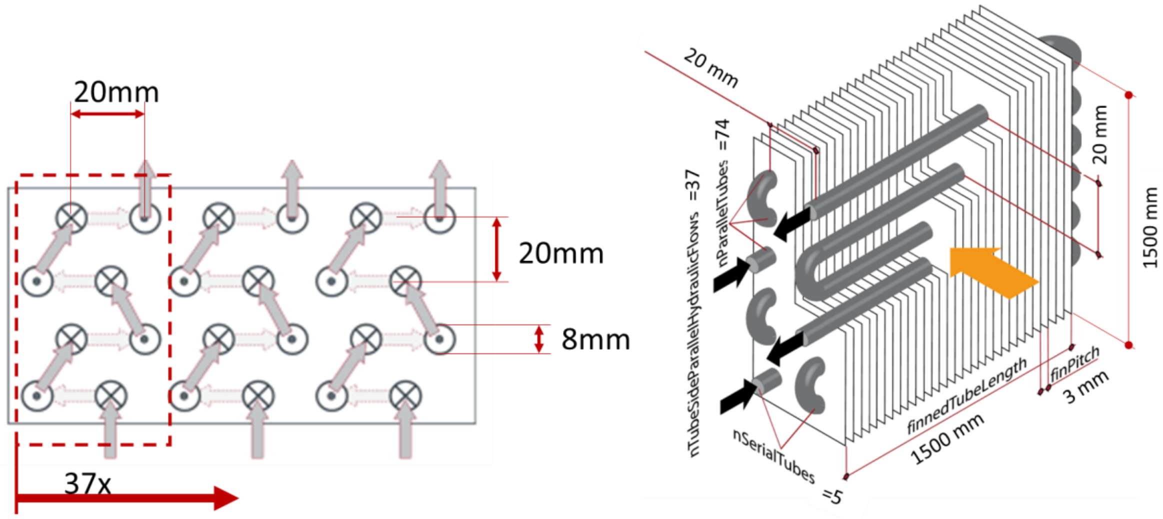


Figure 4.4: Geometry of the used evaporator, modified screen capture from the TIL Suite [43]

4.2.2 Results of the baseline system

The refrigerant mass in the evaporator, the condenser and one meter of the liquid line as well as the compressor speed, the heating capacity and the reached COP of the R410A system are listed in Table 4-2. For the refrigerant mass in the evaporator, the assumed mass of refrigerant in the distributor and headers is listed in brackets. The baseline R410A-system reaches COPs between 2.10 and 5.14. Regarding the refrigerant mass it can be noted that the large internal volume of the evaporator leads to the high refrigerant mass of over 5 kg in this component during defrost operation. Since the receiver installed in the refrigerant cycle (see Figure 4.1) acts as buffer to provide enough refrigerant for all operating conditions, the receiver needs to accommodate this additional refrigerant charge. Furthermore, it needs to be noted that the present study only investigates the refrigerant charge in the evaporator and condenser. To consider the refrigerant charge in the liquid line, the mass of refrigerant per meter liquid line is given. The refrigerant in the compressor's internal volume and refrigerant dissolved in the compressor oil, which would further increase the refrigerant charge of the system, is not considered.

Table 4-2: Results of the baseline R410A system (* heating capacity >30 kW)

Operating point	Refrigerant (R410A) mass in			Compressor speed [rpm]	Heating capacity [kW]	COP [-]
	Evaporator (out of which distributor and headers) [kg]	Condenser [kg]	1m liquid line (d_i=16 mm) [kg]			
H1	1.26 (0.61)	0.55	0.17	3610	30.0	2.10
H2	1.30 (0.60)	0.47	0.20	3150	30.0	3.45
H3	1.59 (0.58)	0.47	0.20	2335	30.0	4.38
H4*	2.00 (0.57)	0.48	0.20	2030 (min)	31.2	5.14
DF (defrost)	5.42 (0.48)	0.16	0.22	2900	not applicable	not applicable

4.2.3 Necessary changes for the use of R290 compared to R410A

In order to demonstrate necessary changes to adapt the refrigerant cycle to R290, the consequences of only changing the refrigerant to R290 without any modifications to the R410A system are presented. Table 4-3 lists the refrigerant mass in the investigated components, the refrigerant flow velocity in the suction line, the compressor speed and the achieved heating capacity. Evaluation of the COP was dropped for this case since the compressor used is not available for R290. Since the refrigerant charge is decisive for the necessary safety measures, and the refrigerant mass in the evaporator during defrost operation is evaluated to almost 2 kg, charge reduction measures targeting this component will be investigated.

Table 4-3: Results of the non-adapted R410A system using R290 (*heating capacity < 30 kW)

Operating point	Refrigerant (R290) mass in				Compressor speed [rpm]	Heating capacity [kW]
	Evaporator (out of which distributor and headers) [kg]	Condenser [kg]	1m liquid line (d_i=16 mm) [kg]	Suction line flow velocity [m/s]		
H1*	0.50 (0.27)	0.23	0.09	19	4350 (max)	20.6
H2*	0.55 (0.27)	0.20	0.09	20	4350	23.5
H3	0.62 (0.26)	0.20	0.09	19	4020	30.0
H4	0.75 (0.25)	0.20	0.09	16	3370	30.0
DF (defrost)	1.94 (0.23)	0.05	0.10	14	2900	not applicable

Besides the different safety requirements, the different thermodynamic properties have the following consequences:

- The utilization of R290 leads to a significantly lower refrigerant mass in the investigated components compared to R410A.

- The heating capacity is significantly lower, even at the maximum compressor speed of 4350 rpm, only 21 kW can be reached at operating point H1. This is caused by the lower volumetric refrigeration (heating) capacity of R290 compared to R410A as can be seen in Figure 2.2.
- The higher volumetric flow of R290 due to the lower volumetric refrigeration capacity requires larger piping diameters compared to R410A if similar refrigerant flow velocities are targeted.
- Due to the different properties of the refrigerants, the sizing of the expansion device needs to be verified thoroughly.

4.2.4 Charge optimized R290 system

In order to reach the nominal heating capacity, a compressor with larger displacement is used. Manufacturer's data [47] utilizing R134a is used to evaluate the efficiencies according to Eq. 8 – Eq. 10, since no data was available fot R290. Curves fitted to these values as depicted in Figure 4.5 are used in the simulation model.

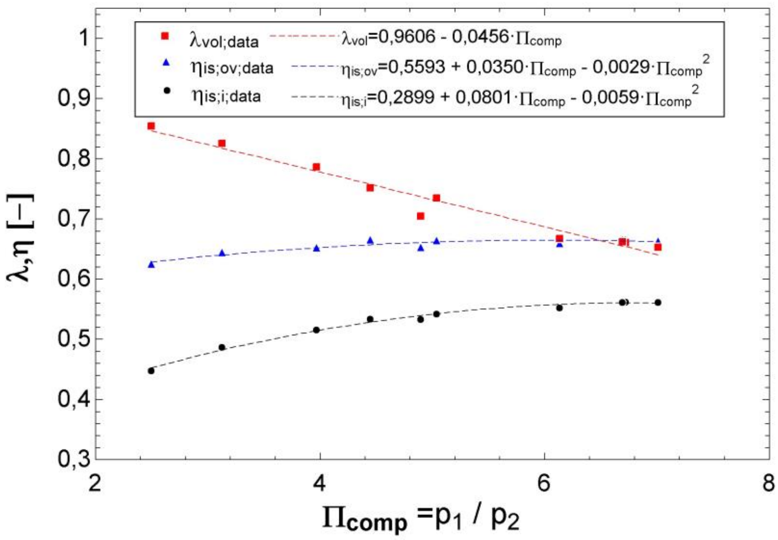


Figure 4.5: Compressor efficiencies calculated from manufacturer's data [47] and used interpolation curves

The most significant modification is required for the evaporator. In order to decrease the refrigerant mass in this component, especially during defrost operation, the fin-and-tube heat exchanger is replaced by a multi-port extruded tube (MPE) heat exchanger, based on [37]. As in the previous cases, suction gas superheat of 5 K is controlled via the expansion valve. The used geometry is depicted in Figure 4.6. In order to consider the refrigerant mass in distributor and headers, the volume equivalent to 0.5 m of the suction line filled with liquid refrigerant at the corresponding pressure level is included. Increasing the suction line inner diameter to 32 mm leads to flow velocities between 9 and 17 m/s, the liquid line inner diameter of 16 mm is unchanged and leads to velocities between 0.8 and 0.9 m/s.

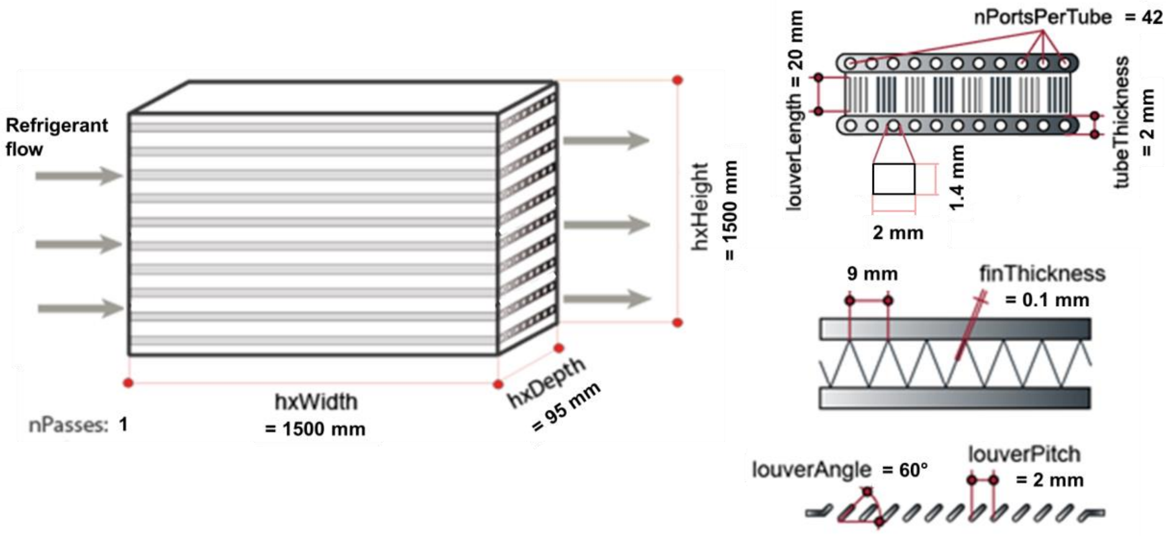


Figure 4.6: Geometry of the MPE evaporator, modified screen capture from the TIL Suite [43]

The following correlations supplied by the TIL Suite are used: Refrigerant heat transfer based on [48], [49] and [50], refrigerant pressure drop based on [51], air-side heat transfer based on [54] and a fin efficiency of 1. Pressure drop on the air-side is neglected.

4.2.5 Results of the charge optimized R290 system

Table 4-4 lists the refrigerant mass in the investigated components, the compressor speed, the heating capacity and the COP of the charge optimized R290 system.

Table 4-4: Results of the improved R290 system

Operating point	**Refrigerant (R290) mass in**			**compressor speed [rpm]**	**Heating capacity [kW]**	**COP [-]**
	Evaporator (out of which distributor and headers) [kg]	**Condenser [kg]**	**1m liquid line (d_i=16 mm) [kg]**			
H1	0.31 (0.22)	0.20	0.09	2335	30.0	2.54
H2	0.31 (0.22)	0.20	0.09	1610	30.0	3.62
H3	0.34 (0.21)	0.20	0.09	1130	30.0	4.41
H4*	0.38 (0.21)	0.20	0.09	937	30.0	5.05
DF (defrost)	1.30 (0.18)	0.05	0.10	1450	not applicable	not applicable

The utilization of a MPE evaporator enables the reduction of refrigerant mass in the component from 0.75 to 0.38 kg during heating operation and from 1.94 to 1.30 kg during defrost operation compared to the baseline fin and tube design (see Table 4-3). An alternative to reversing the refrigerant circuit for defrosting would be a hot gas defrost. This could even further reduce the refrigerant mass in the evaporator during defrost.

The achieved COPs are in the range between 2.54 and 5.05. Compared to the R410A system (see Table 4-2), the R290 reaches a higher COP at large temperature lifts (operating point H1) while the R410A system shows a slightly higher COP at low temperature lifts (operating point H4).

5. Conclusions

The main aim of this research was to investigate differences in the chemical and physical behavior between R290 (propane) and R410A which is a state-of-the-art refrigerant for heat pumps. R290, which has been used in air conditioners or refrigerators for a long time, has also proven to be a feasible alternative to R410A as a refrigerant for heat pumps, although the market share is still very low. While compared to widely used R410A, the Global Warming Potential (GWP) of R290 is negligible, the flammability is a risk that has to be handled. A high number of international, EU and national Austrian standards exist, which deal with the handling of flammable substances. Partly they are compulsory, partly it is strongly recommended to follow these.

There is a number of other large differences in the physical and chemical properties between R290 and R410A. The pressure level of R290 is significantly lower than for R410A, the critical temperature is a lot higher: p_{crit} = 42.5 bar, t_{crit} = 96.7 °C for R290 and p_{crit} = 49 bar t_{crit} = 71.3 °C for R410A. Moreover, the enthalpy of vaporization of R290 is higher, but the lower density leads to a lower volumetric refrigeration capacity. Higher heat transfer values for R290 can be fostered by decreasing heat transfer areas. To avoid larger pressure drops when using R290 larger diameters for the refrigerant piping are necessary.

These numerous differences make it obvious that a vapour compression cycle suitable for R410A cannot be at the same time optimized for R290. Larger modifications have to be taken into consideration. Most importantly due to the lower volumetric refrigeration capacity of R290, a compressor with larger displacement is necessary. Other measures include larger piping diameters and the sizing of the expansion device needs to be verified.

Within this research, an optimized circuit design for R290 was calculated using a Modelica-based TIL Suite software package to model the heat pump in the Dymola environment.

With these optimizations, it turns out that in a wide range of temperature lifts, R290 shows a higher COP, especially for higher values, making it especially an energy efficient alternative for unrefurbished or partly refurbished buildings as well as for domestic hot water heat pumps or diverse non-household applications. Moreover, the behavior of the system changes e.g. in the compressor speed which is significantly reduced and the refrigerant mass in the evaporator and condenser is much lower.

Mainly because of safety reasons, technologies aiming to reduce the refrigerant charge show great potential and will in future continue to be of great importance. Many components as compressor, condenser, receiver, expansion valve and evaporator have to be considered. An interesting technology in this respect is the falling film evaporator which allows the refrigerant not to evaporate from a liquid pool, but from a thin liquid film.

While at low temperature lifts the R290 system is almost as efficient as the R410A system, the R290 system outperforms the R410A system at higher temperature lifts. Although much more research was put into R410A heat pump systems until now, R290 seems to be a promising alternative not only for systems at lower temperatures as refrigerators and air conditioners, but also for a wide range of applications for heat pumps. As for the European Union, Regulation 517/2014 requires a significant drop in the total GWP of all refrigerants brought on the market, the replacement of R410A with a GWP of 2088 by R290 with a GWP of 3 can be the decisive step towards fulfilment of these requirements. Nevertheless, questions regarding high temperature systems and systems with high refrigerant loads in sensible areas still have to be investigated.

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