14th IEA Heat Pump Conference
15-18 May 2023, Chicago, Illinois

Performance and safety analysis of charge reduced brine to water heat pumps using R290

Timo Methler[a,*], Hannes Fugmann[a], Clemens Dankwerth[a], Christian Sonner, Katharina Morawietz[a], Lena Schnabel[a]

[a]Fraunhofer Institute for Solar Energy Systems ISE, Heidenhofstraße 2, 79110 Freiburg, Germany
* timo.methler@ise.fraunhofer.de

Abstract

The European implementation of the Montreal Protocol, the F-gas Regulation, restricts the use of refrigerants with high GWP. Thus, low-GWP refrigerants, in particular R290 (propane), are becoming the standard in the growing heat pump market. Within this contribution the results of the research project "Low Charge 150 g" are shown. The focus is on the reduction of charge needed in heat pumps, favorably below 150 g propane. As part of this project different components were combined in brine to water refrigerant circuits and experimentally evaluated. Conclusions and correlations will be described based on ErP standards and a multitude of measured values. All evaluation focus on the behavior based on charge variation and are used to create four distinct operation modes of a heat pump. Additionally, a test bench for refrigerant distribution testing was built to analyze R290 concentration development in controlled indoor scenarios. All measurements are done with compact plate-to-plate refrigerant circuits. The heating capacity reaches up to 12.85kW using 124 g R290. The current best solution achieves a specific heating capacity <10 g/kW with a SCOP (ErP) >4.5.

Keywords: propane; R290; heat pumps; refrigerant; charge reduction; safety

1. Introduction

Heat pumps have become the technology of choice for society and policy makers in the European region and increasing in popularity worldwide. Commonly used refrigerants have been regulated recently and they will be regulated/ banned in the near future. Different regions worldwide follow individual timelines. These timelines were agreed upon by participants of an assembly in Montreal in 1987, commonly referred to as "the Montreal protocol" [1]. The European implementations have led the EU-market to an increasing focus on R290 as refrigerant. R290 (propane) has a low GWP of 0.02 [2]and very good thermodynamic properties but is highly flammable and therefore needs special attention regarding safety. The reduction of charge in the system is an elegant solution to decrease the hazard represented by the system. The following contribution will show the basic concept of charge reduction, visualize the impact, and show taken measures.

2. Infrastructure

All measurements presented below were taken at Fraunhofer ISE. To ensure standardized testing, conditioning modules, were used to simulate source and sink and are referred to as secondary modules/circuits. These conditioning modules are standardized equipment in the laboratory and have the following adjustable/ controllable parameters: fluid temperature, pressure drop and mass flow. The conditioning module for the sink side is filled with water and the module on the source side with a mixture of ethylene/glycol and water with a freezing temperature of -20 °C. For evaluation, the heating and cooling capacities on the secondary sides were calculated by the measurement of the mass flow employing a coriolis sensor, combined with in- and outlet

* Corresponding author. *E-mail address:* timo.methler@ise.fraunhofer.de

temperature measurement, using submerged PT100 sensors and their applicable specific heat capacities. The electrical power consumed by the driver and compressor was measured between the driver and the power grid. Therefore, all frequency converter losses are included in the results. To estimate the power necessary for external pumps on the secondary side, differential pressures across the heat exchangers on the secondary side were monitored by pressure differential sensors. Controllers were used to control the inlet temperatures to the heat pump. The temperature difference between in- and outlet of the two plate heat exchangers on the secondary sides, were controlled by adjusting the mass flows of secondary fluids. The source side was set to 3 K and the sink side to 5 K temperature difference.

3. Test procedure

All prototypes measured by Fraunhofer ISE were built as the flow chart in Fig. 1. The refrigerant circuits (RC) were built in the workshops at Fraunhofer ISE and named following a versioning system RC-version. All manufacturing guidelines are coordinated with partners in the project´s advisory board and with component manufacturer, as well as the component manufacturers.

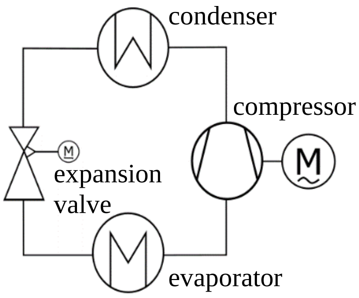

Fig. 1. Simplified Refrigerant Circuit Flowchart

The components used, were all selected to be able to provide 8 kW heating capacity in their upper rating range. Generally, all pipes were insulated by 1cm foam insulation. The heat exchangers were set up to operate in counter flow arrangement.

All measurements are taken as charge variations since the main focus lies on the low charge behavior of refrigerant circuits. For this purpose, a special charging station was developed and constructed for automated charging and safe discharging of up to three refrigerant circuits mounted at the same time processed individually. To control the direction of the R290 mass flow the charging station can be heated or cooled. In-depth explanation of the charging station can be found in the article of [3].

For the measurements the charging unit charged an initial amount of refrigerant to the circuit (for instance 80 g). With the initial amount several operation points were measured, which includes several different parameters set by the secondary modules, frequency converter and EEV. Those operation points were measured for 30 minutes in stable conditions, respectively. After running through this measurement matrix of different operation points the charging unit automatically charged 10 g extra refrigerant and the operation points of the measurement matrix were measured again. The procedure continues until high pressure protection is tripped, this commonly occurred between 200 g to 250 g of refrigerant R290 in the circuit. Within the refrigerant circuit the following parameter are recorded: Temperature and pressure in suction and discharge line, temperature in liquid and injection line, pressure difference across the heat exchangers and amount of refrigerant charge. The collected data is compiled in a combined database and enhanced by additional documentation including all pipe lengths, all pipe diameter, insulation type, insulation diameter, pictures, component list, oil type, oil amount, sensor connection length and diameter for differential pressure sensors. The additional base information is used in some following corrections mentioned below. During operation the suction side super heat of the refrigerant circuit is controlled with an electronic expansion valve (EEV), the reference temperature sensor in the suction line, was implements as submerged PT100 sensor. All set values are added to the measurement database and are used in the evaluations below.

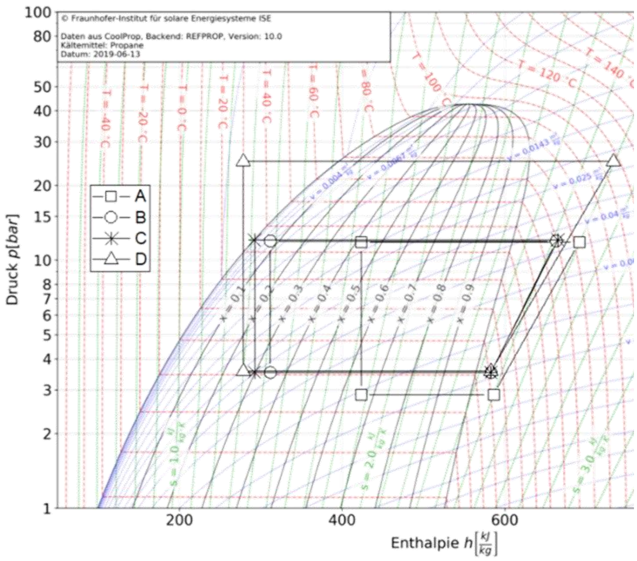

Fig. 2. Log (p) -h diagram visualizing heat pump operation of **RC8-21** at B0/W35/SSH10/F40 for the special attention points A-D, as marked in Fig. 3.

4. Operation with different charge

Fig. 3 shows five stacked graphs ((a)-(e)) with a common x-axis, which displays a full charge variation measurement for refrigerant circuit RC8-21 at operation point B0/W35/SSH10/F40 (brine inlet temperature: 0 °C; water flow temperature: 35 °C; suction super heat: 10 K; frequency: 40 %). When considering the full range of a charge variation measurement as described in Chapter 3, there are four major operation points, where different thermal and fluid dynamic effects have significant impact on the circuit. These four operation points are marked with **A-D** in Fig. 3. Point **A** is under charged with refrigerant. The necessary super heat is not achieved. In point **B** the desired super heat is achieved, but the system did not reach the maximum efficiency and heating capacity. Point **D** is over charged with refrigerant. Efficiency drops significantly. Most common operation is point **C**, which is widely regarded as baseline heat pump cycle. The presence of sub cool is considered essential for the baseline cycle.

Fig. 2 shows point **C**'s cycle in a log(p)-h diagram marked with stars. Operation **C** is expected to have 0-2 K sub cool, which is difficult to measure reliably in the tight physical arrangement. Therefore, the team is using alternative measured values to determine marginal sub cool. The behavior of the EEV (shown in Fig. 3 (b)), is used as main indicator for marginal sub cool. The EEV's task is the expansion of the refrigerant, this can be considered as forced pressure drop. Pressure drop is strongly dependent on fluid velocity, which in term is strongly dependent on density. For this reason, the EEV behavior shift from being strongly charge depended to largely independent, is considered the turning point of marginal sub cool. For this reason, the EEV opening is used to define the "minimal sub cool" points. This is also considered the minimal optimal charge in [3]. The additional charge, with point **C** as reference, does not significantly impact the EEV and therefore no significant density/ velocity changes in front of the EEV are occurring. This shows the existence of sub cool, it does not quantify sub cool reliably but considering fluid dynamics sub cool has to be present. Fig. 3 (d) shows condensation temperature and liquid line temperature; subtracting them from each other results in the sub cool of the system. With increasing charge increasing sub cool is present in the system. Additional charge will accumulate inside the condenser, if no collector is present, the additional inventory reduces the available area for condensation and desuperheating. The loss of available area is balanced with a forced increase in temperature difference and therefore condensation temperature. This is particularly prominent at operation **D**, where overfill has occurred. Fig. 3 (d) shows the significant increase in condensation temperature and therefore pressure. Fig. 3 (b) on the other hand shows the reaction of the EEV to create additional pressure drop the cross-sectional area of the EEV has to be reduced. The significance of the change in operation is also visualized in Fig. 2 where operation **D** is indicated as

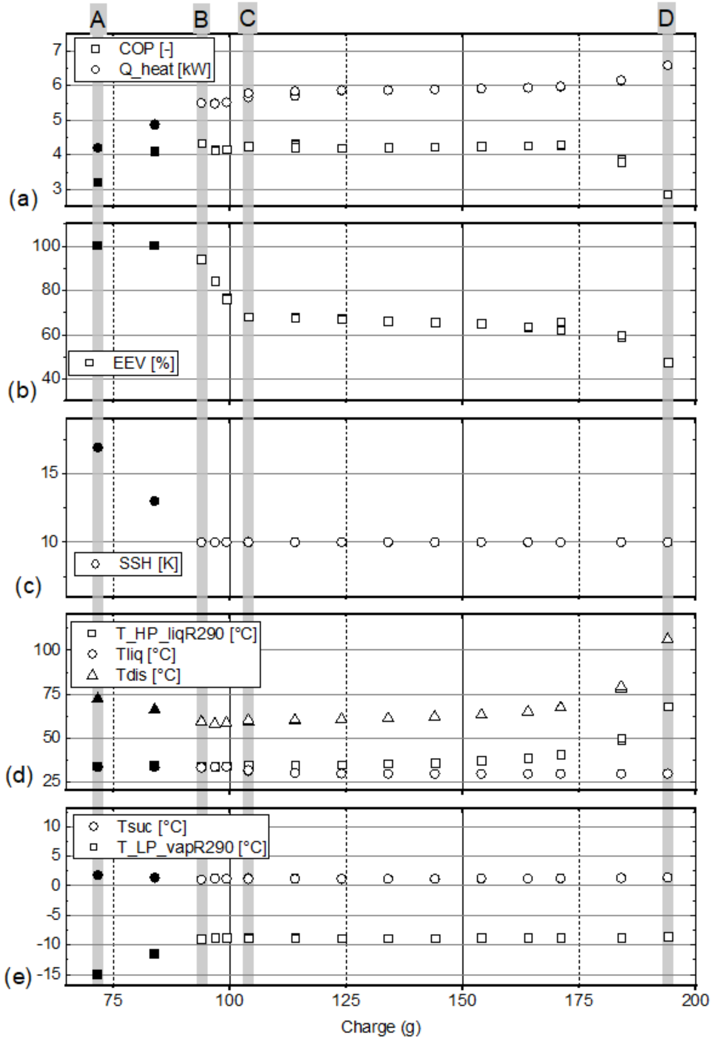

Fig. 3. ((a)-(e)). Charge variation of **RC8-21** at operation point B0/W35/SSH10/F40, 5 stacked graphs showing, (a) COP and heating capacity, (b) EEV opening in %, (c) suction side super heat, (d) high pressure propane condensation temperature as well as condenser outlet and discharge temperature, (e) suction gas temperature and low-pressure propane evaporation temperature. Marked for special interest are charges A-D representing **A** extreme undercharge, **B** undercharge, **C** minimal optimal charge and **D** overcharge operation

triangles. Additional side effects are the increased pressure ratio to be covered by the compressor. This leads to an additional increase in discharge temperature and drop down of COP shown in Fig. 3 (d) and (a) respectively.

When reducing charge, with **C** as reference point, the reduction will have different impacts. First effect will be the reduction of sub cool below 0 K sub cool. This will force the condenser to have two-phase outlet, marked in Fig. 3 as operation **A/B**. This will change the requirements for the expansion valve drastically and any expansion valve made for liquid entry state will reach its physical limit quickly, which is the major difference between operation **A** and **B**. As discussed earlier due to the higher velocities with two-phase flow and a considered constant mass flow. Mass flow can be considered constant, for points **B-D,** due to constant rotational speed of the compressor in combination with the constant pressure and temperature at the compressor inlet shown in Fig. 3 (d) and Fig. 2. Operation **A** does have altered mass flow compared to the operation **B-D**, due to the reduced inlet pressure and increased super heat which leads to lower densities at compressor inlet. This works to reduce the velocity increase at the EEV, which is still dominated by the decrease in density due to two-phase operation. The suction temperature displayed in Fig. 3 (e) shows the constancy of suction side temperature over the full charge range, due to the consistent secondary side source temperature. At operation **A**, where the decreased density and with it required additional EEV opening, exceeded the operation limits of the EEV. As a result, the low pressure is reduced lower than necessary for 10 K SSH, this leads to uncontrolled increase super heat in the circuit and is the dominant reason for the decreased pressure at the compressor inlet. Both operation states are visualized in Fig. 2 **B** as circles with the EEV at maximum operation envelop and **A** as squares with the EEV operated outside of its operation envelop. During their investigations the team Palm et. al. in 2006 found the same correlations for COP and temperature developments over charge, these are shown and discussed in [4]. Additional visualization can be provided by employing IR pictures of the cross section of condensers. The concept of using IR pictures for distribution evaluation was used and described nicely in [5]. In this case the same concept can be used to show the inventory level in condensers rising with charge during the previously described charge variation. For this visualization, a charge variation of refrigerant circuit **RC8-17** at B0/W24/SSH10/F10 has been selected to show charge inventory in the condenser. Operations **B**, **C** and **D** are marked and selected similar as RC8-21 discussed before (see Fig. 4 ((a) - (c))). For circuit **RC8-17** the IR pictures of the condenser cross section for operations **B**, **C, D** and an additional point are shown in Fig. 5. Refrigerant flow is downstream from top right to bottom right of the heat exchangers. The first picture of Fig. 5 (**B**) represents an underfilled system with a total charge of 114 g R290. Except the hot gas region in the upper region of the heat exchanger the IR picture indicates a homogenous temperature profile starting 26 on the y-axis to the bottom part of the heat exchanger. The homogenous temperature profile reflects the process of condensing the refrigerant. Pure liquid phase is not reached. At the outlet of the heat exchanger the refrigerant is still in two phases.

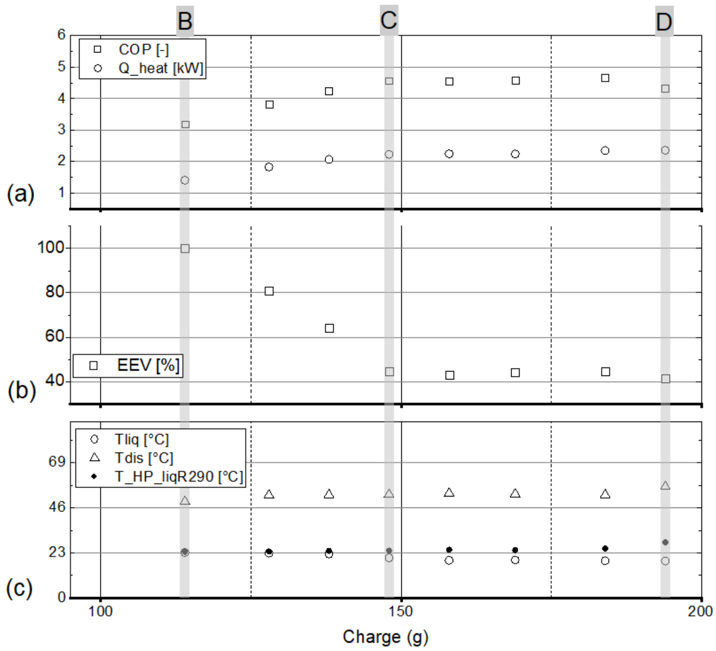

Fig. 4. ((a) - (c)). Charge variation of **RC8-17** at operation point B0/W24/SSH10/F10 stacked graphs showing, (a) COP and heating capacity, (b) EEV opening in %, (c) temperatures in discharge and liquid line as well as condensing temperature. Marked for special interest are charges **B-D** representing **B** undercharge, **C** minimal optimal charge and **D** overcharge operation

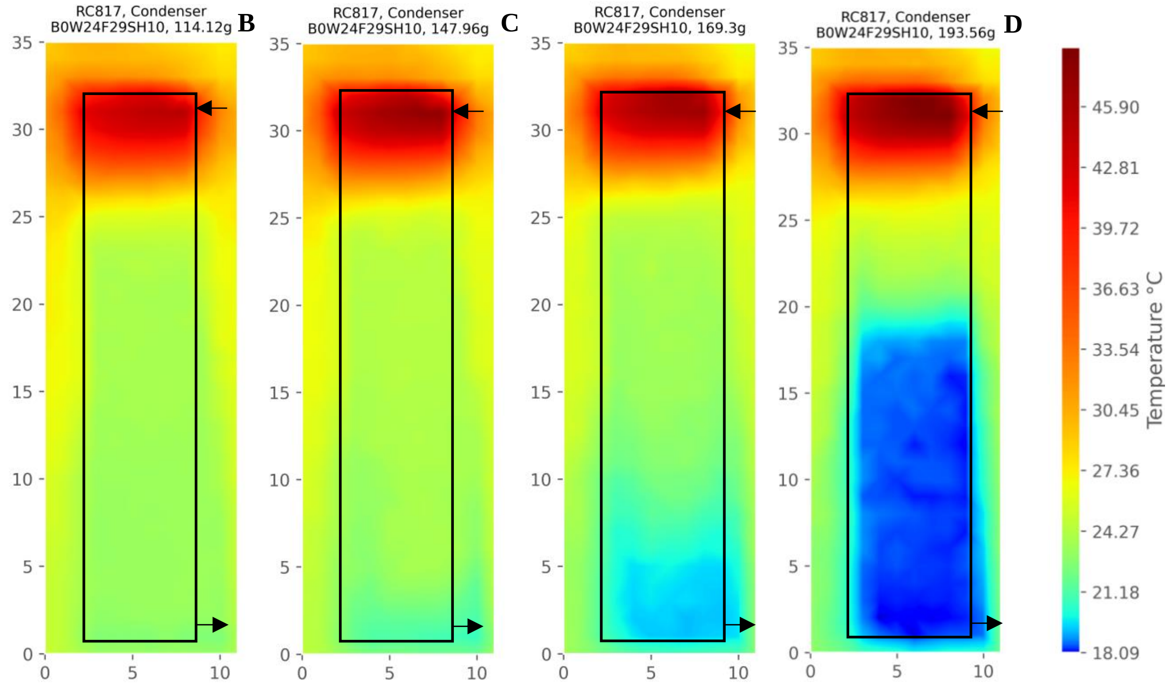

Fig. 5. IR pictures of condenser RC8-17 for operation points **B**, **C**, **D** and one additional operation point at 169 g, heat exchanger outline is marked for all images. Inlet R290 is located top right of the heat exchanger image, whereas the outlet is located in the bottom right. Water is running in counter flow the opposite direction.

The second picture in Fig. 5 (**C**), shows RC8-17 at the minimal optimal charge operation with 147 g R290. The upper region of the heat exchanger shows almost no change in temperature, compared to the image of operation **B**. Whereas the bottom region demonstrates a slight change in temperature level. This indicates a completed condensation and starting of sub cooling of the refrigerant at the bottom of the heat exchanger. This is reflected in Fig. 4 (c) where the condensation temperature (T_HP_liqR290) and measured outlet (T_liq) are shown. The temperature differential represents the measured sub cool. Fig. 5 shows an additional image for 169.3 g of charge to visualize the development of the sub cool region, visible at the bottom of the heat exchanger. The final picture of Fig. 5 (**D**) refers to an overfilled system with a total charge of 193 g R290. As explained earlier for overfilled systems, the discharge temperature rises. The increase in discharge temperature can be seen at the top region of the heat exchanger in a displayed dark red color. Furthermore, with additional charge, the condenser works as a collector. All unnecessary charge stays in the condenser in front of the EEV. The sub cooling increases and the area used for condensing the refrigerant is getting smaller. This can be seen in the blue bottom part of the heat exchanger. The earlier discussed behavior of charge varied refrigerant circuits is applicable to all circuits measured during the project, which were not changed by adding additional components. Therefore, qualitatively visualized process in Fig. 2 applies.

5. Design rules for charge reduced heat pumps

To design low charge heat pumps the following design rules should be in mind to keep the amount of refrigerant low and use the charged refrigerant as efficient as possible.

5.1. Heat exchanger manifolds

Heat exchangers have connection ports/ manifolds/ headers to distribute or collect the refrigerant to/ from the plates and channels. Those ports are often designed to be very simple and easy to manufacture. They are not optimized regarding refrigerant efficiency and often they are holding some amount of refrigerant/ oil which is not used in the circuit. For example the manifold and outlet port of a plate heat exchanger condenser are commonly punched with the same tool as the secondary side ports. This leads to manifold diameters 2-3 times larger than necessary, and therefore to charge inventories 4-9 the needed mass and volume. Manifolds could be designed much smaller to safe unused refrigerant, for this reason port design is an important part of charge reduced design.

5.2. Oil reduction/ absorption

The oil, commonly used in refrigerant circuits, absorbs 5-18 weight% (depending on the oil) of the refrigerant. The knowledge of adsorption can be measured and is widely known, a good example of behavior can be found in Ginies work of 2016. One example of the work is shown in Fig. 6 [6]. These absorption values occur during heat pump operation. The amount of refrigerant which is absorbed by the oil depends on several factors. The type of oil and refrigerant is important, as well as the physical conditions of the oil (temperature and pressure). The main amount of oil in a refrigerant circuit is present in the oil sump in the compressor. Therefore, it makes a difference if the oil sump is on the suction side or discharge side of the compressor.

The team did not elaborate on the benefits or negatives of either solution and found no clear indication for which system is charge optimal. The general rule in regards of oil remain, avoiding oil traps, reducing oil sump, considering the oil type, will all reduce or impact the required total amount of refrigerant.

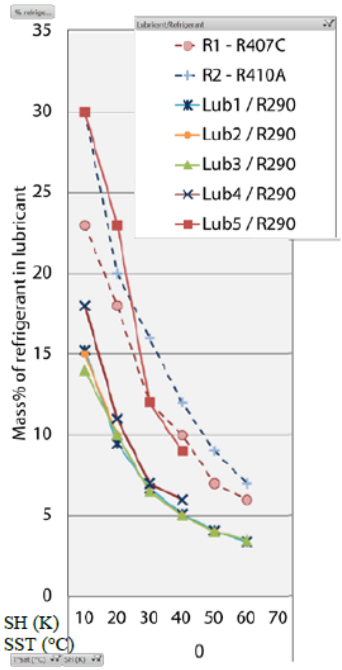

Fig. 6. Oil adsorption overview R290 with different lubricants, partial representation of work published in 2016 by Ginies et. al. [2]. SH= super heat; SST = suction side temperature

5.3. Liquid line

The liquid line should be designed as short and thin as possible. Due to the high density of the refrigerant in the liquid line (see top left Fig. 2) it has a high effect on the total charge of the system.

5.4. Operation strategy

For reverse operation via 4-way-valve the condenser and the evaporator are switched in the refrigerant circuit. This is used for deicing or active cooling mode. In those operation modes much more refrigerant is needed due to higher volumes in the evaporator (evaporator fills up with liquid refrigerant). For highly charged reduced refrigerant circuits other deicing modes are recommended and passive cooling could be an alternative.

5.5. Super heat optimization

Different suction super heat values require different amounts of refrigerant for their minimal optimal charge operation, marked as C in Figure 7. The higher the super heat the lower the required refrigerant charge. Though the suction super heat has a high impact on the efficiency of the unit. The lower the super heat the higher the efficiency.

In some scenarios to achieve a certain maximum refrigerant charge it can be useful to use the suction super heat to regulate the required refrigerant charge. All prototypes during the project had a charge delta between 5 K SSH and 15 K SSH of 20 g to 70 g, main reason for the widespread is the charge absorbed in the oil, due significant impact on oil sump/ suction gas temperature, as well as different inner volumes in evaporators, where the varied super heat has a significant impact on the average density in the evaporator.

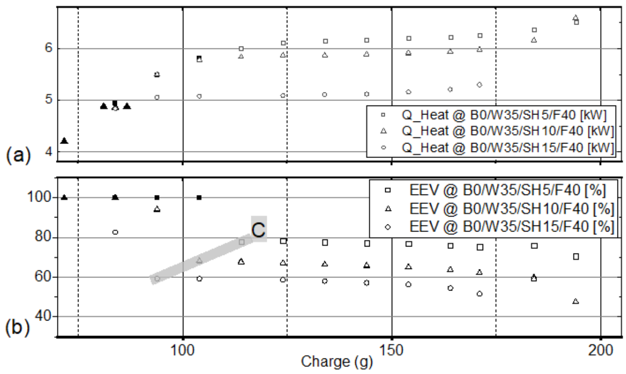

Fig. 7. (a,b) charge variation for **RC8-21** showing SSH 5, 10 and 15 K operations, shared x axis for all graphs, (a) showing heating capacity and (b) shows EEV opening, marked minimal optimal charge region as discussed earlier

5.6. Heat exchanger

A major amount of refrigerant charge in refrigerant circuit stays in the two heat exchangers. The larger the inner volume of the heat exchanger the more charge is needed. In the design process should be kept in mind, that the selection of the heat exchangers has a large impact on the final charge. Specifically additional plates due to safety margins should be considered carefully when designing a charge reduced system. Other options for charge reduction are asymmetric heat exchangers, which enable low secondary side pressure drops but low inner volumes on the primary refrigerant side. A recent publication of Will et. al. showed increased necessary charge if any maldistribution is present. The investigation was done purely theoretically and simulation based, never the less all five most common models for two phase charge estimation were in agreement. [7]

5.7. Heat exchanger

Normally filter dryer are used in the liquid line in refrigerant circuits. The filter dryer is holding liquid refrigerant with a high density. For very compact systems the share of the total refrigerant charge can be quite high. To avoid some refrigerant from the filter dryer in the liquid line, the filter dryer can be shifted to the suction line of the refrigerant circuit where the refrigerant is in gas phase and has a lower density. Due to high gas velocities the filter dryer needs to be larger than in suction line. Nevertheless, some refrigerant can be saved, the team saved 50 % charge during pre-evaluations. Though, the filter dryer in suction line has additional pressure losses which harm the efficiency of the complete unit, the actual impact will vary on a case-to-case bases.

Another option is to renounce the filter dryer completely. The amount of saved refrigerant is the highest. The manufacturing process and needs to be a very careful to have no particles in the refrigerant circuit. And the evacuating needs to be very accurate to remove almost every humidity from the unit.

6. Safety analysis outlook

To evaluate the safety of propane heat pumps during and after a leakage, knowledge of the propane concentration in its surroundings is essential. Of interest are the behavior of concentration development over time in the casing of the refrigerant circuit (heat pump) as well as in the installation room around it. To be able to measure distribution of propane, inside and outside of the casing, a test installation room (see Fig. 8) has been installed at Fraunhofer ISE. The dedicated room enables the team to test the distribution development over time in a controlled environment. The leaked mass of propane is not limited; however, the leakage mass flow rate is limited. This flow rate represents small leakages to rupture leaks varying between 0.001 g/s to 2 g/s, respectively; and the leakage position can be varied within the casing, focusing on critical positions such as brazing joints, any specific component necessary to evaluate.

The test installation room is built with sandwich panels with a volume of approximately 21 m³ (3,5 m x 2,5 m x 2,4 m), representing a typical size of a cellar. The installed test room includes, a gas warning system, a ventilation system and measurement equipment. The ventilation system can be used for cleaning the installation room during or after a leakage test. Measurement results with desired resolution are recoded with more than 30 gas sensors, temperature sensors and differential pressure sensors, which are installed in the test installation room. The gas sensors (Dynament propane gas sensors with ATEX certificate) detect propane concentration by infrared

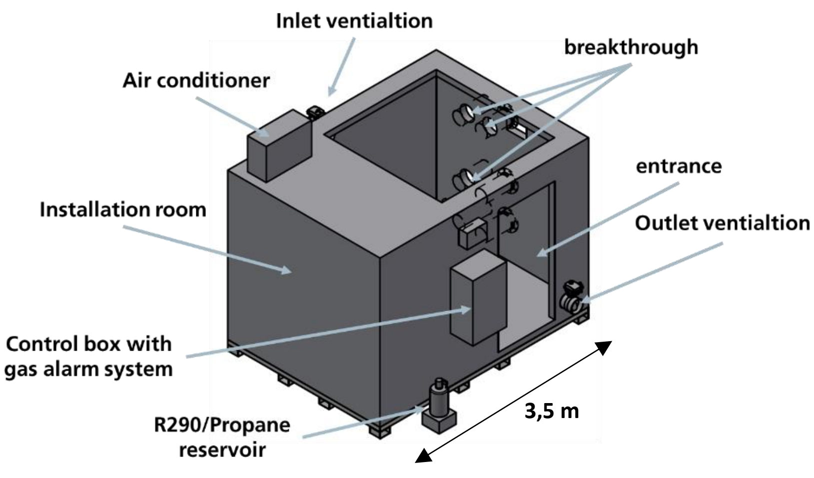

Fig. 8. Installation room for leakage testing of heat pumps

measurement principle and have a response time of ~1.2 s. The gas sensors are installed within the casing and within the test installation room around the casing. As part of the project “Low Charge 150” three types of casings have been defined for testing: (1) without casing / very permeable casing, (2) with a casing that is individually designed, based on the refrigerant mass in the circuit, (3) with a casing, that has a fixed volume according to the “white goods” volume with 60 cm x 60 cm footprint. Casing (2) and (3) are built to have custom side panels to be able to accommodate new designs and component-based additions to housings. The dedicated measurement campaign for “Low Charge 150” will start beginning of 2023. The desired outcome will be, whether a casing can be designed to include a passive safety concept intrinsically. A further goal is the definition of an upper limit of propane with a fixed casing volume and this passive safety concept. The main benefit of such a concept would be to eliminate additional active safety devices in the casing or in the installation room.

7. Conclusions and outlook

The described behavior in Section 4 has been present in more than 20 refrigerant circuits measured during the project “Low Charge 150” and all circuits with no additional components. The thermodynamic relations between the individual components and the thermodynamic behavior with charge reduction has been very consistent and conclusive. Overall charge reduction has a long chain of dependencies attached. The authors summarize the full chain as follows.

Reduced charge (below the minimal optimal charge point) leads to gas parts in the liquid line **(saves charge)**. This reduces the available enthalpy delta in the condenser **(reduce heating/ COP)**. The gas phase in the liquid line leads to additional non desired expansion. This in term leads to higher super heats. Additional super heat reduces absorption in the compressor oil **(saves charge).** Lower evaporation temperatures/pressure created by the additional expansion increases the necessary compression ratio **(increases electric use/ reduces COP)**. The rise in suction side super heat and the additional pressure ratio increases the hot gas temperature. Rising hot gas temperature leads to the less efficient gas-based heat transfer in the condenser **(reduce COP)**. A similar conclusive argument chain forms for overfilling which is explained in section 4, but does not represent special interest for charge reduction.

To summarize the described effects and relations the Fraunhofer ISE derives seven rules for designing charge reduced heat pumps

1. Oil reduction
2. Minimal liquid line
3. Heat exchanger manifolds/ ports
4. Operation strategy
5. Super heat strategy
6. Heat exchanger inner volume/ distribution
7. Minimal / suction side filter dryer

These rules are described in detail above in section 5. Most if not all of the rules can be respected when selecting components. The fundamental relations also give strong areas of attention for future developments. Testing and implementing new oils with lower absorption of refrigerant are a strong contributor to charge reduction, as well as the general reduction of oil. On the thermal dynamics side the quality of fluid-distribution in heat exchangers has an impact on the efficiency as well as on charge efficiency.

Acknowledgements

The study presented in this paper received funding from the German Federal Ministry of Economic Affairs and Climate Action (BMWK) under the grant agreement number FKZ 03EN4001A (LC150).

The cooperation between Fraunhofer ISE and UPV and the ideas for this work were inspired, supported and supervised by Prof. José Miguel Corberán Salvador. He served as Professor in the Department of Applied Thermodynamics and Director of the Institute for Energy Engineering at the Polytechnic University of Valencia (UPV), Spain and died on July 7, 2022 at the age of 65. We will continue the work with his attitude towards professionalism in refrigeration, cooperation and humanity.

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