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Experimental Evaluation of a charge reduced Heat Pump Module using 150g of Propane

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ABSTRACT

A low refrigerant charge brine-to-water heat pump circuit was evaluated addressing a heating capacity of 5 - 10 kW and using propane as refrigerant. Propane is a natural refrigerant, has a low GWP (3) and attractive thermodynamic properties. Due to safety aspects the refrigerant charge was limited to 150 g, resulting in 0.45 kg of CO_2 equivalent.

The reduction of charge was achieved by minimized volumes of the internal components: the condenser and evaporator were chosen with an asymmetric plate profile, the liquid line was designed as short and thin as possible, the filter dryer in the liquid line was shifted to the suction line and the amount of compressor oil was reduced to its minimum.

The charge reduced heat pump circuit was tested with different heat exchangers and two different compressors at various points of operation. This paper presents the results of the experimental evaluation and potentials for further optimization.

Keywords: charge reduction, propane, heat exchanger, heat pump

1. INTRODUCTION

The installation and use of heat pumps is increasing worldwide. In parts of Europe, the heat pump has become the dominant heating solution in new buildings. Due to the increasing awareness towards global warming and the need to reduce the use of fossil fuels and environmentally harmful gases, the market share of heat pumps is expected to rise steeply. Heat pumps use electricity for heating purposes with significantly higher efficiency than purely electrical resistance heaters. Combined with electricity from renewable sources, heat pumps can provide building heating without using any fossil fuels. Currently most heat pump systems employ refrigerants with a high global warming potential (GWP) due to security advantages regarding flammability and toxicity. The global community unanimously banned refrigerants with an ozone depletion potential (ODP) by the Montreal protocol [13] and is now gradually restricting the maximum GWP for refrigerants. In Europe the F-gas Regulation [9] is implemented to phase out the use of refrigerants with a high GWP. Such regulations encourage and force the shift towards refrigerants with very low GWP, such as natural refrigerants, in a timely manner. Propane is a well-known refrigerant with excellent thermal dynamic properties and a minimal impact on the environment (GWP = 3, ODP = 0). Furthermore, no regulations, except for safety, are expected for propane as refrigerant, making it predestined for use in heat pumps. Propane heat pumps research and development focuses on developing ***security systems*** [4][5], optimizing ***performance*** [2], reducing ***charge*** [1, 3] and ***comparing*** various ***refrigerants*** i.e. evaluating the different perspectives and making comparisons to other, mainly synthetic, refrigerants [10, 15][12].

The aim of the presented work is to create a heat pump using commercially available components only achieving 5 - 10 kW heating capacity with a maximum charge of 150 g propane. This paper will show the first results of a charge-reduced brine-to-water propane heat pump.

2. MOTIVATION AND BACKGROUND

In 2001 and later in 2007, Fernando et al. [7] presented first experimental results of a low-charge propane heat pump. The comprehensive results and findings are part of his PhD thesis [14] on a brine-to-water heat pump with a heating capacity of 5 kW at -3 °C/35 °C and a charge of 230 g. [6] In 2018, Andersson [1] presented a study on a heat pump with heating capacity of 10 kW using a refrigerant charge of less than 150 g propane (B7/W40) with a non-hermetic compressor. Both studies investigated heat pump concepts, in which unique prototype components as well as an automotive scroll compressor were installed. The employed automotive compressor, however, was neither rated for the use with propane nor was it designed as a hermetic compressor. Another shortcoming was that the source temperature fluctuated strongly because the test environment was too small. The work presented by Andersson provided a baseline for the reduction of charge as well as an identification of the preliminary areas of attention for future research.

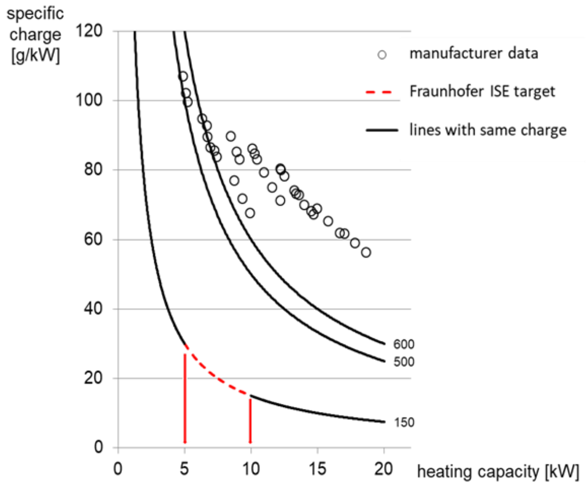


Figure 1: Specific charge as a function of the heating capacity of available brine-to-water heat pumps

There are a few propane-charged systems commercially available. Most of them are air-to-water systems for outdoor locations. Outdoor systems are more common because of less security restrictions and concerns. Due to the chance of leaking, indoor propane systems could create a hazardous environment. Outdoors any leaked propane would quickly dissipate, thus creating no hazard. Most manufacturers do not employ propane heat pumps for indoor applications due to the high security requirements and the accompanying financial costs. A brief market analysis of available brine-to-water heat pumps from the beginning of 2019 is shown in Figure 1. The dots represent available brine-to-water systems. The black lines show the lines with same charge. The red line marks Fraunhofer ISE's target of a heating capacity between 5 and 10 kW with 150 g propane. As seen in the graph, this target results in a specific charge of 15 to 30 g/kW, a reduction by a factor of four in comparison to market available heat pumps.

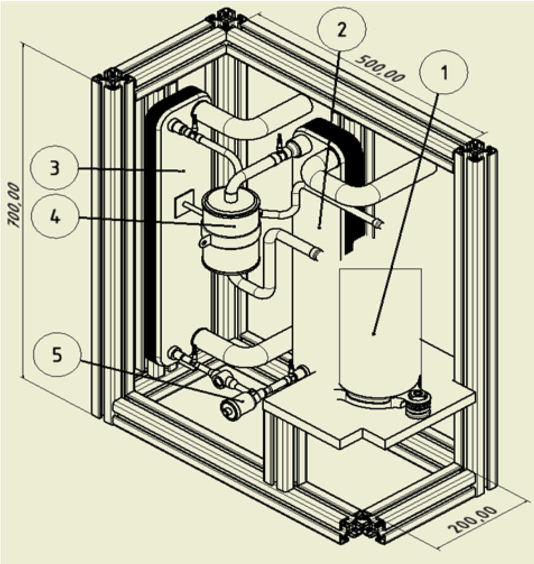
A more elaborate market study with a wider range of application of air-to-water and brine-to-water heat pumps was done by Palm [11] as part of the LifeFront report [8]. These reports are comparable to the market research done preliminary to this low-charge heat pump development project.

3. EXPERIMENTAL SETUP

3.1. Main refrigerant circuit

The refrigerant circuit was designed and built as shown in Figure 2**.** As a side effect of minimizing the pipe length and diameters, the overall dimensions were reduced. With dimensions of roughly 700x500x200 mm, the refrigeration circuit is small in comparison to other units with the same capacity. The main components used for the heat pump are mentioned in Figure 2.

During the measurements all changes to the refrigerant circuit have been documented with version numbers. The compressor has been changed during the project duration and from now on will be referred to as system 1 and system 2 respectively. Minor changes result in numbering such as 2.1 ➔ 2.2. This will be clarified if necessary in relevant places.



No.	Component
1	Compressor
2	Evaporator: Alfa Laval ACH72- 16AH- F
3	Condenser: Alfa Laval CB65- 16AH- F
4	Filter dryer: ALCO PCN 008925
5	Expansion device: Parker SDR2

Figure 2: Main components in the refrigeration circuit.

3.2. Measurement procedure

Most measurements were taken in a setup in which charge was varied. First, the initial charge was filled into the heat pump. Operating points were recorded by running the circuit until steady state conditions were reached. With another 30 minutes in steady state conditions mean values during the last few minutes were recorded and calculated. The transition to the next operational point was initiated by adding additional 10 g of propane to the system and the described loop was repeated to complete the series of measurements.

The temperatures for the secondary circuits were chosen based on typical test parameters in the heat pump sector. For the source temperature -10 °C, -7 °C, 0 °C, 12 °C were chosen. These values refer to the inlet temperature of the evaporator. For the sink 35 °C, 45 °C, 55 °C and 65 °C were chosen as outlet temperatures of the condenser. For both of the secondary hydraulic circuits, a constant temperature difference between inlet and outlet of 5K and 3K respectively was maintained as specified in EN14511. Measured data points are designated as “BX/WY” which derives from “brine with temperature X” and “water with temperature Y.” The brine always characterizes the source temperature and water the sink temperature.

3.3. Test environment

To ensure standardized testing, the secondary conditioning modules, providing the secondary fluids, were used to simulate source and sink and are referred as secondary modules. These conditioning modules are standardized equipment in the laboratory and have the following adjustable parameters: flow temperature, pressure difference and mass flow. The conditioning module for the sink is filled with water and the module for the source side with a mixture of ethylene glycol and water with a freezing temperature of -30 °C. Both modules are connected to the building cooling system to ensure operation at all times. All testing was done in a monitored environment with gas detection system and forced ventilation. In case of a propane leakage, the power supply is switched off and the ventilation enhanced.

4. RESULTS AND DISCUSSION

The heat pump cycle was analyzed by simulation and by thermodynamic and visual analysis of the experiments. As simulation software IMST-ART was used, the simulations were based on the available geometric data from the datasheets provided by the manufacturers, excluding compressor volume and oil volume due to stability issues and insufficient available oil data to calculate accurate absorption of the refrigerant in the oil. The simulation predicted the demonstrator would be able to fulfill the goal of a heating capacity of 5 - 10 kW.

4.1. Measurement results

Figure 3 shows the experimental results for the two different compressors (System 1 and 2) with their heating capacity and the coefficient of performance (COP) for different refrigerant charges at B0/W35 and two different compressor speeds.

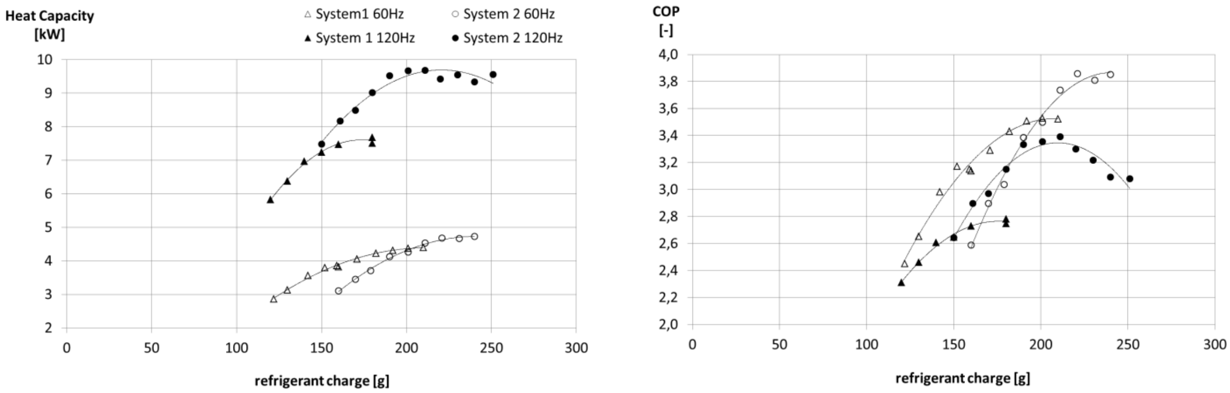


Figure 3: Heating capacity (left) and COP (right) as a function of refrigerant charge comparing both compressors, all operation points are B0/W35

The total range tested was from 30 Hz to 120 Hz in 10 Hz increments. The suction superheat (SSH) has been set to 10 K; nevertheless it was not reached for all operation points due to the limited operational window of the electronic expansion valve (EEV). The SSH varied between 5 and 22 K. The deviation from 10 K is only significant for operation points with the lowest and highest charge. The right figure shows the COP values corresponding to the heating capacities of the left figure. As highest values a heating capacity of nearly 10 kW and a COP of 3.8 was reached. For the heating capacity it can be seen that system 2 reaches a higher heating capacity than system 1 at high frequencies. For lower frequencies the behavior of the two systems is opposite. Until 200 g of charge system 1 has a higher heating capacity. Beyond 200 g, system 2 has a higher heating capacity. Both systems reach a higher heating capacity with higher charge.

In the right figure it can be seen that both systems achieve a higher COP for lower frequencies and both systems achieve their highest COP in a range around 200 g charge. For lower frequencies, system 1 achieves higher COPs. For higher frequencies it is the other way round.

In order to design an entire heat pump system, stable operation must be ensured at all possible operating points. Moreover the behavior at all operation points must be well understood in order to give accurate information about performance. For this purpose different parameter variations were tested. In this paper a variation of source temperature and suction superheat will be mentioned, as can be seen in Figure 4.

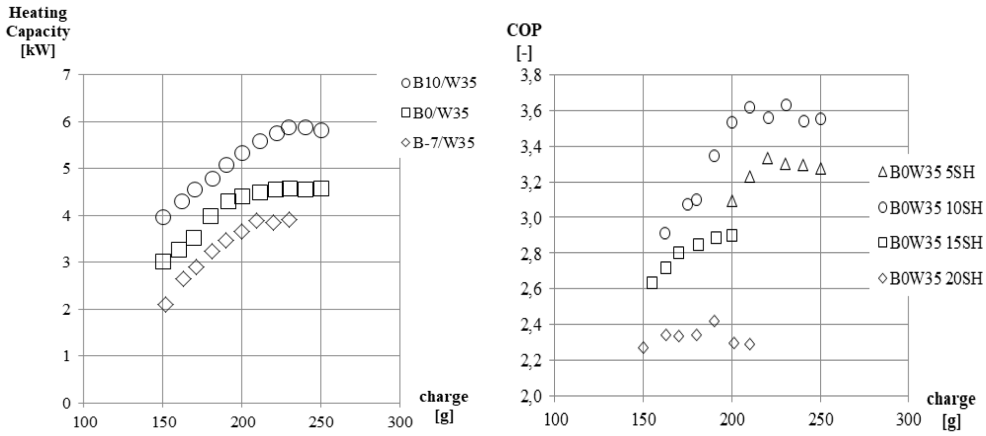


Figure 4: Heating capacity as a function of charge, source temperature variation, system 2, SSH 10 K (left) and COP as a function of charge, comparing superheat 5 K, 10 K 15 K, 20 K, all operation points are B0/W35 (right)

The left graph of Figure 4 shows the heating capacity as a function of source temperature and refrigerant charge. A reduction of the source temperature leads to lower amount of charge necessary to reach the maximum capacity. Possible explanations are different characteristics of oil absorption and changes in the refrigerant density. The figure also shows the significant impact of the source temperature on the provided heating capacity. Comparing 10 °C and -7 °C the maximum heating capacity drops from 5.9 kW to 3.9 kW, corresponding to a difference of 34 %.

The second parameter varied is the suction superheat (SSH), as can be seen in the right graph of Figure 4. The superheat does have a significant impact on efficiency. The highest efficiencies around 3.6 were achieved during measurements with an SSH of 10 K.

The results show the sensitivity of the charge reduced system. Due to the wide range of tested charge, the EEV is not sufficiently sized in either direction to cover all points of operation. Considering the constant source input temperature and therefore a correlating evaporation temperature, varying superheats have an impact on the suction gas temperature. The varying suction gas temperature has again an impact on the suction gas density and therefore on the mass flow into the compressor. For high superheats a large area of the evaporator is used as a gas heater, which lowers the efficiency of the heat exchanger and therefore the entire unit. Any or all of these reasons are possibly part of a high COP optimum at 10 K SSH as well as the shifting saturation points. These issues will be addressed in further investigations.

Optical analysis

In addition to the thermal performance evaluation thermography pictures were taken from the condenser as shown in Figure 5 (middle) and photographs were taken from the evaporator as shown in Figure 5 (right). The pictures were taken from the face side view onto the heat exchanger plates. Both heat exchangers have 16 plates. The refrigerant flows along the vertical axis of the pictures, running from the inlet to the outlet as marked in Figure 5 (middle, right). The thermography pictures were taken with a direct view of the plate heat exchangers with no insulation applied. Therefore the pictures were not taken of a flat surface, thus impairing the resolution and accuracy of the images. It was attempted to normalize the emissivity, which was set to 1 by applying a thin chalk layer. Based on the relative temperature distribution a maldistribution in the condenser can be assumed.

The evaporator could not be analyzed employing thermography, due to condensed water and ice formation on the surface. Therefore, the distribution of the evaporator had to be analyzed by optical information, such as ice formation on its surface. An example of the ice formation is shown Figure 5 (right). Only the first few plates show ice formation on the outside of the evaporator. A maldistribution can be concluded. Only the first plates are reached by a significant mass flow of refrigerant.

Due to the maldistribution on both heat exchangers the pressure ratio is higher than necessary. Especially the evaporator showed a very high pressure drop. This results in a higher power consumption of the compressor which again leads to lower COPs of the unit. Both heat exchangers as well as the distribution system of the evaporator will undergo further investigation.

5. CONCLUSION AND OUTLOOK

The low charge heat pump has successfully shown the potential of charge reduction and the results provide a solid base to accurately define further research areas.

Two main aspects are relevant for the further work:

1. The identification and determination of design rules and criteria which take the total charge and/ or the specific charge into account. There is tradeoff between the optimal COP and the optimal specific charge, as shown in Figure 5 (left). The efficiency maximum is located in the region of 170-200 g of refrigerant, whereas the specific charge minimum is located in the region of 150 g refrigerant. The results given before indicated that 150 g of refrigerant is too low for a stable performance for this unit.
2. Identification of performance maps for components working with low charge systems. Using a minimum charge results probably into strongly underperforming operation points of the heat

exchanger (Figure 5 (middle, right) or EEVs. This strongly influences the overall efficiency. Further research needs to identify these operational limits and should develop solutions with low influence on the refrigerant charge.

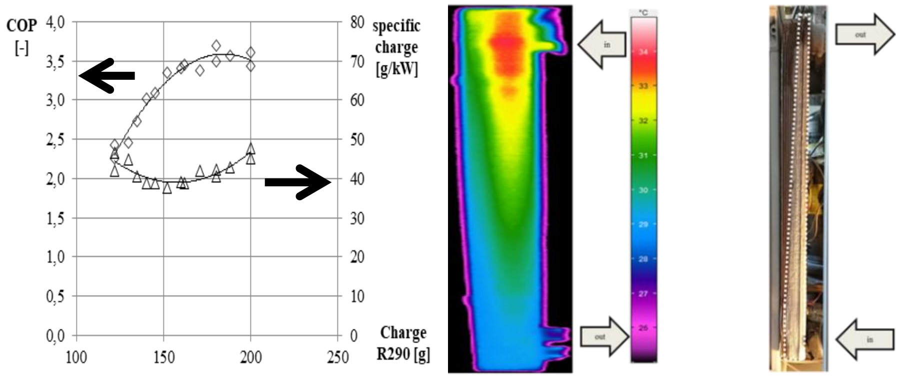


Figure 5: COP and specific as a function over total charge, at B0/W35, 60Hz and SSH 10K, system 1 (left), thermography pictures of the condenser at B0/W35, 60Hz and SH10K, system 1 (middle), ice formation on evaporator at B0/W35, 60Hz and SH10K, system 1 (right)

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