REFRIGERANT CHARGE PREDICTION FLUCTUATIONS IN THE SCOP CAMPAIGN OF A BRINE-WATER HEAT PUMP

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Abstract: *The refrigerant amount reduction is a matter of increasing importance because of the rise in prices of current hydrofluorocarbons and the safety reasons of the leading natural alternatives. There is no space for the refrigerant deposit in vapour compression cycles with refrigerant charge optimised; consequently, it is essential to know which refrigerant charge will make the vapour compression cycle work with the best performance during a year.*

In this work, a simulation of the different tests of the seasonal COP campaign for a brine-water heat pump is performed using the software IMST-ART. With the results, it can be possible to understand the different behaviour of the heat pump during the year and guess the best criteria to establish the optimum refrigerant charge.

Keywords: Refrigerant charge amount, optimisation, SCOP, natural refrigerants, design conditions

1. INTRODUCTION

In Europe, the decarbonisation of cities has become a matter of interest in recent years. With the Europe roadmap to 2050 [1] and the objectives for 2030 [2], there will be changes in the systems used for heating, col, and domestic hot water production. Currently, households' heating and sanitary hot water account for 78.4 % of the total consumption [3], 75 % of it produced using non-renewable sources [4]. Heat pumps present one of the possible solutions to reduce gas emissions in households. If the seasonal performance factor (SPF) is higher than 2.5, part of the heating produced is considered obtained from a renewable source [5]. It is a carbon-free alternative if no refrigerant leakage is produced and powered by renewable electricity.

Refrigerant leakage or liberation is one of the major problems concerning heat pumps and has provoked restrictions and regulations about which refrigerants are allowed. The first forbidden group was chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) due to their ozone depletion potential [6]. Currently, hydrofluorocarbons are getting restrictions in the amount commercialised [7], and the main alternatives that work at a suitable pressure for the heating application are at least mildly flammable or toxic [8].

These refrigerants have security concerns, and for this reason, some safety systems can be added or have the refrigerant charge inserted in the range permitted (150g or calculated accordingly to the nature of the refrigerant) [9]. For this reason, refrigerant charge reduction became a matter of interest in recent years [10]–[13] and refrigerant charge prediction too.

There is no place for refrigerant deposit inside the refrigerant circuit in systems with minimum refrigerant charge. Therefore, the fluctuation of charge needed due to variations of external conditions over a year will impact the overall behaviour and reduce the heat pump's efficiency. This change in the required refrigerant has been reported before [14]. Still, the impact in the SCOP campaign or knowledge about which test condition the performance should be sacrificed most due to the overall effect is missing.

This work has studied the fluctuation of refrigerant needed through the most significant test conditions during an average year for heating at low temperatures in an average climate. Then, the effect of using the biggest and the lowest refrigerant charge is also shown.

2. METHODOLOGY

As mentioned before, this work's interest is to understand the variations in refrigerant charge required in a heat pump used for domestic heating, along the year due to the external condition changes.

A custom version of IMST-ART v4.0 [15] will be used for this study.

2.1. HEAT PUMP DEFINITION

The heat pump used is a ground-source heat pump designed to have enough heating capacity for a standard family in Europe (11kW) [15] with a slightly higher refrigerant charge amount of propane than the allowed by EN-378 [9] without considering any extra security measures (150g).

Table 1. Compressor definition.

Type	CC	Oilt type	Oil amount (dm^3)
Rotary	30.6	POE	0.4

The selected heat pump components can be seen in Table 1 and Table 2. The compressor technology is a variable speed rotary with POE oil and a compression chamber of 30.6 cm^3. The heat exchangers chosen are asymmetric to reduce the refrigerant amount required. The number of plates has been selected so that the pinch point in the refrigerant outlet of the heat exchanger is near 4K for the most demanding condition (B0W35 @ 120 rps). With this consideration, the definition of the brazed plates heat exchanger can be found in Table 2.

Table 2. Heat exchanger definition.

	Type	Ref. internal volume (dm^3)	Height (cm)	Width (cm)	Number of plates
Condenser	Asymmetric	0.18	31.8	76.2	22
Evaporator	Asymmetric	0.41	37.6	11.9	22

2.2. Definition of the case studies

The heat pump has been analysed in the different points used to characterize the seasonal coefficient of performance (SCOP) according to the standard EN-14825 [16].

In this case, the heat pump is selected to work under average climate conditions for low-temperature application, i.e., used to cover the heating demand, of a family, in centre Europe using the radiant floor as a terminal unit. The conditions used for the simulation can be seen in Table 3.

Table 3. Different conditions of each study.

	Partial load(%)	Test name	Heating capacity needed (kW)	T w in (°C)	T w out (°C)	T b in (°C)	T b out (°C)
A	88	B0W34	9.68	27.9	34	0	-3
B	54	B0W30	5.94	26.3	30	0	-3

	Partial load(%)	Test name	Heating capacity needed (kW)	T w in (ºC)	T w out (ºC)	T b in (ºC)	T b out (ºC)
C	35	B0W27	3.85	24.6	27	0	-3
D	15	B0W24	1.65	22.9	24	0	-3
E(TOL)	100	B0W35	11	28	35	0	-3

The outlet water temperature has been selected to be with variable temperature and fixed mass flow: However, in the evaporator, the inlet temperature is constant, as it is ground-source, but the temperature difference has been considered fixed.

These conditions were used in all simulations results. There have been performed four different case studies:

- 1: Working with the optimal refrigerant charge at each point.
- 2: Working in all points with the maximum refrigerant charge obtained in 1.
- 3: Working in all points with the minimum refrigerant charge obtained in 1.
- 4: Working in an intermediate refrigerant charge amount between 2 and 3.

At this point, it is clear that the performance results of campaign one will be the best ones and serve as a reference. All the simulations, but the most refrigerant demanding, will work overcharged in the second group. All test conditions, but the least refrigerant demanding, will be underfilled in group three. And in the last group, in some conditions, the system will be overcharged, and in others, the system will be underfilled.

The definition of optimum charge was the one that allowed the system to work with a controlled value of superheat (SH), calculated at the evaporator outlet, of 5K and a degree of subcooling (SC), calculated at the condenser outlet, of 5K as well. In the other case studies, a minimum value of SC of 4K has been considered to be sure that the calculation of the refrigerant charge in the condenser is correct. A minimum value of SH of 5K, too, to have a reasonable value that a hypothetical expansion valve could control.

3. RESULTS AND DISCUSSION

The main results of this work are the refrigerant charge needed in the ground-source heat pump of the different points studied and how the refrigerant charge affects the performance results. More concretely, how overcharging/undercharging the system in different test conditions affect the overall yearly energy consumption of the heat pump.

3.1. Optimum refrigerant charge

This study presents the heat pump results without any limit on refrigerant charge. The test conditions are expressed in Table 3 and consist of a ground source-heat pump producing heating at low temperatures in an average climate according to the standard. Different results can be seen in Table 4.

Table 4. Results of the optimised conditions.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T cond (ºC)	T evap (ºC)	T discharge (ºC)	SC (K)	SH (K)	Ref charge (g)
A	105	9.63	3.95	37.4	-7.1	58.3	5	5	166.5
B	65	5.78	4.80	32.6	-5.6	50.0	5	5	174.0
C	45	3.80	5.10	29.8	-5.0	47.5	5	5	186.3
D	30	2.39	5.18	28.0	-5.0	46.6	5	5	198.2
E	120	11.08	3.71	38.7	-7.7	61.3	5	5	164.4

Point D doesn't accomplish the heating capacity required of the test conditions because the compressor speed was limited by its minimum velocity (30rps).

As Table 4 shows, as the partial load decreases, the refrigerant amount required is higher, with a maximum difference of 34g between the most demanding point and the least demanding. With these results, the SCOP obtained would be 4.54.

3.2. Different strategies

The first study is the overcharged one. For this study, all tests are simulated with 198.3g of propane. The results of these simulations can be seen in Table 5.

Table 5. Results of the overcharged conditions.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T cond (ºC)	T evap (ºC)	T discharge (ºC)	SC(K)	SH(K)	Ref charge(g)
A	105	10.07	3.71	43.5	-7.2	65.8	15.3	5	198.2
B	65	5.87	4.55	35.5	-5.6	53.7	9.0	5	198.2
C	45	3.82	4.91	31.2	-5.0	49.6	6.5	5	198.2
D	30	2.39	5.18	28.0	-5.0	46.6	5	5	198.2
E	120	11.75	3.45	46.8	-7.8	71.0	18.5	5	198.2

As expected, working with overcharged conditions make the system store the extra refrigerant charge in the condenser as a subcooled liquid. This effect reduces the two-phase area of the heat exchanger, reducing the overall heat transfer coefficient and increasing condensing pressure. Additionally, as the SC increases, the condensing pressure is obliged to increase to avoid a temperature cross between the refrigerant and secondary fluid.

Seeing the second study results in Table 6, when the refrigerant charge is reduced, the SC is reduced until there is no more reduction possible, and there is a need to increase the SH to compensate the needs. Increasing the SH reduces the refrigerant charge in the evaporator and reduces the oil solubility by heating the compressor. In this case, the impact on the COP is not as significant as in the previous point.

Table 6. Results of the undercharged conditions.

Test condition	Compspeed (rps)	Heating capacity (kW)	COP	T cond (ºC)	T evap (ºC)	T discharge (ºC)	SC(K)	SH(K)	Ref charge (g)
A	105	9.57	3.93	37.2	-7.2	58.3	4	5.2	164.4
B	65	5.70	4.76	32.3	-5.6	50.6	4	5.8	164.4
C	45	3.65	4.96	29.0	-5.0	48.7	4	6.4	164.4
D	30	2.05	4.52	27.0	-5.0	53.5	4	10	164.4
E	120	11.08	3.71	38.7	-7.8	61.3	5	5	164.4

The last case, seen in Table 7, shows an intermediate case of previous ones. In this case, the lack or excess of refrigerant charge of every point only mean a reduced impact in performance results, reducing the heating capacity slightly from case study 2 and COP from the optimum case. It could be considered a compromised solution for selecting the refrigerant charge to add to the system. Still, other options can be used, depending on the criteria and which aspect is most important.

Table 7. Results of the medium charge conditions.

Test condition	Compspeed (rps)	Heating capacity (kW)	COP	T cond (ºC)	T evap (ºC)	T discharge (ºC)	SC(K)	SH (K)	Ref charge (g)
A	105	9.88	3.93	39.0	-7.2	60.3	9.9	5	180.3
B	65	5.81	4.77	33.0	-5.6	50.6	6.1	5	180.3
C	45	3.79	5.16	29.2	-5.0	46.8	4.39	5	180.3
D	30	2.28	5.04	27.1	-6.5	47.8	4.15	6.5	180.3
E	120	11.46	3.70	40.8	-7.8	63.9	11.5	5	180.3

In Table 8, a summary of the main results is shown. It can be seen that the impact of overcharging the system in the SCOP is higher as it affects more the COP of the singular points. Also, the points where this overcharge affects have more weight when doing the year's simulation, affecting more hours during a year. However, overcharging make the heating capacity increase. If the same heating capacity was compared instead of the same compressor speed, there is a possibility that the result would be different, improving the global performance.

Table 8. Results summary.

Case study	Refrigerant charge (g)	Declared heating capacity (kW)	SCOP	Specific charge (g/kW)
Optimum	164.4-198.2	11	4.54	-
Overcharged	198.2	11.75	4.09	16.87
Undercharged	164.4	11	4.3	15.135
Medium charge	180.3	11.46	4.32	15.73

As mentioned before and shown in Table 8 when underfilling, the SCOP is not that affected, and the minimum specific charge is obtained, sacrificing a bit of heating capacity. If a middle term of heating capacity and specific capacity wants to be considered, the results of medium charge could be a possible solution.

4. CONCLUSIONS

A simulation of different cases in different test conditions has been done using IMST-ART obtaining results of refrigerant charge needed by the system and performance results. The next results have been obtained for this ground-source heat pump producing heating at average climate:

- As the heating load required is reduced, the refrigerant charge required increases. From 166.5g to 198.2g in this specific case. (31.7g, 16% from the maximum).
- Overfilling the system the whole year, make the SCOP being reduced gravely.
- Having an average charge amount could be a compromise solution between heating capacity and SCOP.
- Underfilling with a good SH control algorithm obtains the best solution in terms of specific charge.

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