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Optimal Refrigerant Charge Determination based on SCOP Maximization with IMST-ART Simulation Tool

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ABSTRACT

In the past, the refrigerant inventory of vapour compression cycles was not a question under study because traditional refrigerants were widely available. There were no limitations on its use due to environmental, safety or economic reasons. The vapour compression cycles were usually provided by a liquid receiver or a deposit placed in the refrigerant circuit to ensure optimum cycle performance at any working condition and mitigate the adverse effects of the refrigerant charge fluctuations or leakages to the ambient.

The refrigerant charge reduction is a matter of increasing importance, not only because of the rise in prices of hydrofluorocarbons but also due to safety concerns of the leading natural alternatives such as hydrocarbons. For this reason, refrigerant charge optimization became crucial during the design and test phases of heat pumps.

There is no space for the refrigerant deposit in vapour compression systems with optimized refrigerant charge. Consequently, it is essential to know which refrigerant charge will make the vapour compression cycle work with the best performance during a whole year.

In this work, a simulation campaign covering the different tests conditions for the seasonal coefficient of performance (SCOP) for brine-water and air-water heat pumps 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 for each type of heat pump mentioned.

1. INTRODUCTION

Due to the environmental impact of the gas emissions, the fluids used in refrigeration systems are under restrictions and regulations. The first forbidden group was chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) due to their ozone depletion potential (UN, 1989). The refrigerants that are currently used the most, hydrofluorocarbons (HFCs), are getting restrictions about the amount that can be commercialized (Regulation (EU) No 517/2014, 2014), making their prices increase and the search for an alternative a necessity. The main options to replace these refrigerants that work at suitable pressure conditions are at least mildly flammable or toxic (McLinden et al., 2017), which have safety concerns.

Due to these safety problems or the increase in the price of refrigerants, refrigerant charge reduction has become a matter of interest in recent years (Corberán et al., 2008; Hrnjak & Hoehne, 2004; Poggi et al., 2008).
In low-charge systems, the refrigerant receiver elimination is the first modification (Andersson et al., 2018), so the fluctuation of refrigerant charge needed during the year will affect the performance if a certain amount of refrigerant charge is added to the system. Additionally, when leakages take place, it will affect the behavior of the whole year as well, since there is no extra amount to cover this issue.

2. METHODOLOGY

This work aims to see the variation of refrigerant charge needs during a year for a ground-source heat pump and an air-source heat pump. These variations take place due to the external conditions variation or the heating capacity needed. A custom version of IMST-ART v4.0 (Corberán & Gonzálvez-Maciá, 2009) has been employed to perform this study.

2.1Heat pump definition

Two heat pumps were used in the software, one air-water heat pump (AWHP) and one brine-water heat pump (BWHP). Both of them use the same compressor, they work with R290 and the selection criteria of the components have been similar. In the case of the AWHP, the declared heating capacity at low-temperature (35ºC) is 9kW while the declared capacity of the BWHP is 11kW, both in the range of the needs of a single-family house in Europe (Lund, 2001).

Table 1: Compressor definition.

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

In Figure 1 it is shown a scheme of each heat pump. The AWHP extract heat from the air ond provide it to the building through a water loop, while the BWHP it is thought to extract the heat from the ground. It has been considered two option of terminals in the building to use this heat: (i) radiant floor, which would have a nominal temperature of 35ºC and (ii) fancoils with a nominal temperature of 55ºC.

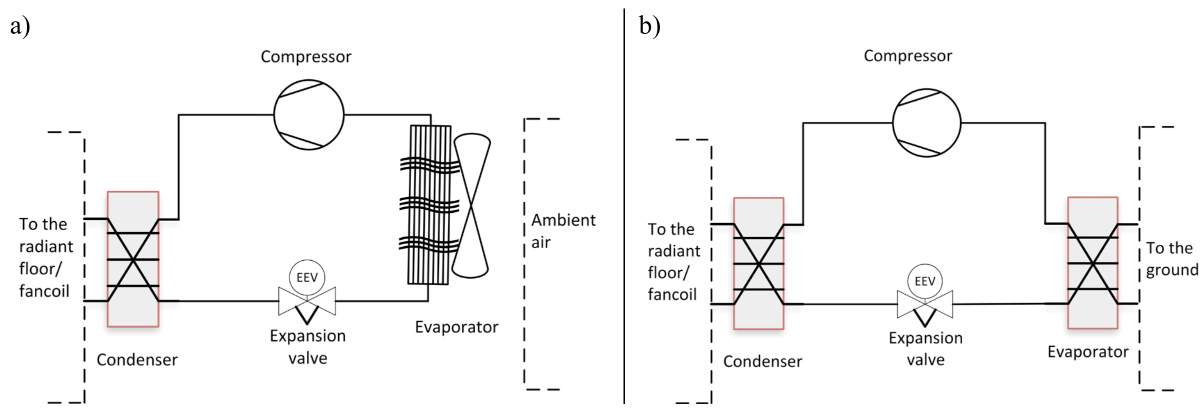

Figure 1: Scheme of different heat pumps studied. a) AWHP b) BWHP

The components for both heat pumps can be seen in Table 1 and Table 2. The compressor in both cases is a variable speed rotary compressor with a compression chamber of 30.6cm3 and POE lubricant. The heat exchanger type selected is a brazed plates heat exchanger (BPHE) for the heat exchanger between liquid and refrigerant and a fins and tubes heat exchanger (FTHE) for air-refrigerant. In the case of BPHE, asymmetric heat exchangers have been chosen with a certain number of plates that make the approximation between refrigerant and secondary side of 4K. In the case of the FTHE the criterion has been similar: an evaporation temperature of -20ºC when air is at -10ºC and 5K of superheating for heating at low-temperature.

Table 2: Heat exchanger definitions.

		Type	Ref. internal volume (dm^3)	Height (cm)	Width (cm)	Number of plates/circuits
BWHP	Condenser	Asymmetric	0.146	31.8	7.6	18
	Evaporator	Asymmetric	0.653	37.6	11.8	34
AWHP	Condenser	Asymmetric	0.146	31.8	7.6	18
	Evaporator	FTHE	1.195	80	50	8

2.2 Case studies

Both heat pumps have been analyzed in different test campaigns described in the standard EN 14825 (European Committee for Standardization, n.d.). These test campaigns are typically used to characterize the seasonal coefficient of performance (SCOP) to know the efficiency of a heat pump during a typical year.

In this study, the test campaigns are described in the standard as heating production for average climate at low-temperature and medium-temperature, respectively. These test conditions will be used for both AWHP and BWHP and are defined in Table 3 and Table 4.

Table 3: Test conditions for low-temperature applications.

	Partial load (%)	Heating capacity required (kW) AWHP/BWHP	$T_{w,out}$ (ºC)	T_{air} (ºC)	$T_{b,in}$ (ºC)	$T_{b,out}$ (ºC)
A	88	7.9/10.1	34	-7	0	-3
B	54	4.9/6.2	30	2	0	-3
C	35	3.2/4.0	27	7	0	-3
D	15	1.4/1.7	24	12	0	-3
E (TOL)	100	9/11.5	35	-10	0	-3
T w in is calculated using the mass flow obtained in point E having 30ºC of water temperature inlet.						

Table 4: Test conditions for medium temperature applications.

	Partial load (%)	Heating capacity required (kW) AWHP/BWHP	$T_{w,out}$ (ºC)	T_{air} (ºC)	$T_{b,in}$ (ºC)	$T_{b,out}$ (ºC)
A	88	7.7/9.3	52	-7	0	-3
B	54	4.7/5.7	42	2	0	-3
C	35	3.1/3.7	36	7	0	-3
D	15	1.3/1.6	30	12	0	-3
E (TOL)	100	8.8/10.6	55	-10	0	-3
T w in variable using the mass flow obtained in point E having 47ºC of water temperature inlet.						

The water flow has been chosen to be constant with variable outlet temperature in the condenser. However, in the evaporator side of BWHP, the temperature difference between the inlet and outlet on the brine side has been selected as constant and the temperature outlet, being 0ºC and -3ºC in all the conditions of BWHP. In the FTHE, the air temperature inlet is dictated by the standard, and the face velocity has been selected as 1.5 m/s.

Additionally, in each simulation campaign, there have been performed three 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.

For the optimal refrigerant charge, case 1, values of 5K of subcooling (SC) and 5K of superheat (SH) were imposed, and this case study was thought to be the reference case.

For other studies, in cases 2 and 3, a minimum value of 3K of SC and a minimum of 5K of SH were imposed.
Case study 2 was limited in some cases by the discharge temperature. In case this happened, a limit of 120ºC of discharge temperature was imposed, resulting in a lower refrigerant charge than the maximum. If maximum discharge temperature was reached during case study 2 of a given campaign, the refrigerant charge adjusted for not overpassing the temperature limit was used for all the case studies.

3. RESULTS AND DISCUSSION

In this work, there are two main results. The first one is the optimal refrigerant charge amount needed. This is obtained from case study 1, affected by the test conditions (compressor speed, source/sink variation, etc.). With the other results, the authors aim to study how seasonal performance is affected by overcharging or undercharging the system when the heat pump doesn't have a refrigerant deposit in the circuit.

3.1AWHP

The first results shown are from the AWHP. In this case, the source temperature and the external temperature change at each test condition. Additionally, the heating capacity and compressor speed and impulsion temperature are a function of the external temperature.

3.1.1 Optimal case: In this heat pump, the main results of the reference case for heating at low and medium temperatures are shown in Table 5 and Table 6. In air-water, the fluctuation of charge between the different test conditions is non-negligible, being 78g for low-temperature applications and 76g for medium temperature applications, which correspond to 35% of the total refrigerant charge amount in both cases. Seeing the variations in the test conditions, the less heating capacity is demanded, the more refrigerant charge is needed, due to the change in external condition variation, which provokes a rise in the evaporation temperature.

In Table 7 can be seen the refrigerant distribution at the different test conditions for heating at medium temperature considering the optimal case. All the components increase the refrigerant charge needed when the heating capacity required decreases. When the condensation pressure decreases, owing to the secondary fluid temperature reduction, the quality inlet of the evaporator decreases, which explains the increase of refrigerant charge in the evaporator. This pressure also affects the compressor's refrigerant charge, reducing its oil solubility. However, as the oil temperature decreases, the effect is opposite and more significant, making the solubility higher with lower compressor speeds. Lastly, in the condenser, the percentage of the total area used for the subcooling increases as the mass flow decreases because the condenser is oversized for this condition.

Table 5: Results of the optimum case of AWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	7.91	2.70	37.5	-19.5	71.4	5	5	151.7
B	55	4.73	4.31	32.9	-5.6	50.8	5	5	185.6
C	35	3.41	5.27	29.9	0.9	45.1	5	5	207.4
D	30	3.41	7.14	26.9	6.2	38.7	5	5	223.5
E	120	7.33	2.51	38.2	-22.1	75.6	5	5	145.3

Table 6: Results of the optimum case of AWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	7.70	2.15	53.5	-18.3	91.7	5	5	144.7
B	55	4.58	3.50	43.8	-5.0	64.9	5	5	174.9
C	35	3.30	4.49	37.8	1.2	56.1	5	5	195.9
D	30	3.34	6.58	31.8	6.3	45.3	5	5	214.4
E	120	7.15	1.96	56.2	-20.7	99.5	5	5	138.4

Table 7: Refrigerant charge distribution in AWHP at medium-temperature for the optimal case.

	Evaporator	Condenser	Compressor	Total
Pont E	34.5	30.7	73.2	138.4
Point A	38.6	30.0	76.1	144.7
Point B	48.8	33.8	92.3	174.9
Point C	55.8	41.4	98.7	195.9
Point D	67.0	41.7	105.7	214.4

Comparing both test campaigns, Table 5 and Table 6, medium-temperature tests require less refrigerant charge than the low-temperature tests. This difference increases when the compressor speed is reduced, being about 5% of the refrigerant needed for condition D.
In point D, the heating capacity is higher than the required (Table 3 and Table 4), but the compressor was at its minimum velocity. At this point, the heat pump would work, switching on/off.
In this campaign, the hypothetical optimal SCOP is 4.17 for low-temperature production and 3.45 for medium-temperature production.

3.1.2 Overcharged case: When the system is overcharged, the most affected points are test conditions A and E. As expected from the beginning, this extra refrigerant charge is mainly accumulated in the condenser as liquid, increasing the subcooling and ultimately, increasing the compressor's discharge pressure. When this pressure is increased, there are other effects, such as the increase of discharge temperature and other solubility equilibrium conditions in the oil in the crankcase.

Since the charge increase was significant, more than 50% in some cases, the discharge pressure increased to a point where the temperature could be too hot for the oil. For this reason, the refrigerant charge has been selected in order to limit discharge temperature to 120ºC. The results of this study are shown in Table 7 and Table 8.

Table 8: Results of the overcharged case of AWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	8.79	2.22	61.1	-19.3	104.9	33	5	190.3
B	55	4.75	4.28	33.3	-5.6	51.5	6	5	190.3
C	35	3.37	5.35	29.0	0.8	44.1	3	5.3	190.3
D	30	3.35	7.21	25.9	5.8	38.5	3	6	190.3
E	120	8.25	2.04	64.5	-21.8	114.5	35	5	190.3

Table 9: Results of the overcharged case of AWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	8.25	2.07	62.4	-18.5	105.4	19.4	5	177.3
B	55	4.61	3.50	44.1	-5.1	65.3	6	5	177.3
C	35	3.25	4.47	37.3	1.0	55.9	3	5.5	177.3
D	30	3.25	6.47	31.3	5.7	46.0	3	6.2	177.3
E	120	7.76	1.85	68.5	-20.8	118.5	22	5	177.3

Even though the limitation mentioned before, at test conditions where the system requires less refrigerant charge amount, the SC reaches values of 35K. At medium temperature, this limitation is stronger because the sink is hotter, and therefore all the discharge temperatures were nearer to the limit at optimal charge conditions. Also, test conditions C and D had to work undercharged due to this temperature limitation, and the SC was reduced to 3K.
The SCOP of this study is 3.48 for low-temperature and 3.19 for medium-temperature.

3.1.3 Undercharged case: When the system is undercharged, the first component that starts noticing the refrigerant charge reduction is the condenser, but when it reaches a specific value, the other components begin also losing performance. In this case, it resulted in an increase in SH, reducing the evaporation pressure. These changes, like what happened with the overcharge, make the compressor work in another condition, and also, the solubility value is different due to changes in pressures and temperature. Table 9 and Table 10 show the results for this condition.

Table 10: Results of the undercharged case of AWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	7.85	2.69	37.3	-19.5	71.0	3.6	5	145.3
B	55	4.50	4.14	32.3	-6.7	54.1	3	8.5	145.3
C	35	3.15	4.89	29.0	-1.5	48.7	3.3	8.5	145.3
D	30	3.08	6.31	25.9	3.0	42.8	3	9	145.3
E	120	7.33	2.52	38.2	-22.1	75.6	5	5	145.3

Table 11: Results of the undercharged case of AWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	120	7.65	2.15	53.3	-18.4	93.4	4.2	6.8	138.4
B	55	4.33	3.33	43.4	-6.7	69.1	3	8.6	138.4
C	35	2.98	4.06	37.2	-2.0	61.5	3.4	9	138.4
D	30	2.91	5.15	31.0	2.0	51.4	3	10	138.4
E	120	7.15	1.96	56.2	-20.7	99.5	5	5	138.4

This undercharged condition seems less aggressive with the value of COP; however, the heating capacity has been clearly reduced. Additionally, when leakages occur, the effect if the system is already undercharged may result in a considerable loss of performance.

The SCOP of the undercharged situation is 3.76 for low-temperature and 3.06 for medium-temperature.

3.2 BWHP

The same studies have been performed for the BWHP. In this case, the refrigerant fluctuation during the year is considerably reduced because of the stability of the secondary fluid temperatures in the evaporator side. This stability makes the heat pump work in similar evaporation temperature and compressor suction conditions similar during the year, reducing the global fluctuation. In this case, the maximum change is 16g at low temperatures and 11g at medium temperatures. It is also curious how, in the medium-temperature campaign, the observed trend in refrigerant charge is not followed, and point C is the one with the lowest refrigerant charge demand.
As expected, with the lower variation of the refrigerant charge, the effect of overcharge and undercharge is also more minor because the amount needed to be added or subtracted is lower.
In the BWHP, the SCOP values of the optimal, overcharged, and undercharged campaigns are 4.64, 4.53, and 4.67, respectively for low-temperature and 3.5, 3.43 and 3.54 for medium-temperature.

Table 12: Results of the optimum case of BWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	9.94	3.91	39.7	-5.7	60.2	5.0	5.0	189
B	70	6.36	4.66	34.2	-5.1	51.9	5.0	5.0	192.2
C	50	4.31	5.05	30.6	-5.0	48.1	5.0	5.0	193.5
D	30	2.39	5.17	28.0	-5.0	46.7	5.0	5.0	203.2
E	120	11.52	3.68	41.3	-6.0	63.2	5.0	5.0	187.5

Table 13: Results of the optimum case of BWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	9.24	2.82	55.7	-5.3	79.5	5.0	5.0	192.4
B	65	5.57	3.59	44.9	-5.0	65.2	5.0	5.0	187.4
C	45	3.67	3.93	38.6	-5.0	59.9	5.0	5.0	186.3
D	30	2.33	4.33	33.3	-5.0	54.7	5.0	5.0	197.4
E	120	10.64	2.61	59.0	-5.6	84.4	5.0	5.0	191.1

When overcharging the system, Table 14 and Table 15, the same behavior observed in AWHP occurs: the SC increase, making the condensation pressure increase and reducing COP but increasing heating capacity.

Table 14: Results of the overcharged case of BWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	10.28	3.85	42.5	-5.7	63.6	11.8	5.0	203.2
B	70	6.46	4.55	36.0	-5.0	54.1	8.1	5.0	203.2
C	50	4.34	4.85	32.3	-5.0	50.5	7.0	5.0	203.2
D	30	2.39	5.17	28.0	-5.0	46.7	5.0	5.0	203.2
E	120	12.00	3.64	44.6	-6.1	67.1	13.3	5.0	203.2

Table 15: Results of the overcharged case of BWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	9.46	2.86	56.4	-5.4	80.4	8.3	5.0	197.4
B	65	5.67	3.57	46.1	-5.0	66.8	7.9	5.0	197.4
C	45	3.70	3.81	40.4	-5.0	62.5	7.1	5.0	197.4
D	30	2.33	4.33	33.3	-5.0	54.7	5.0	5.0	197.4
E	120	10.86	2.65	59.5	-5.6	85.0	7.8	5.0	197.4

In this platform (BWHP), going to the undercharged situation, Table 16 and Table 17, mainly affects the SC level in conditions with low compressor speed. In conditions with low heating capacity, as the water mass flow has been set as constant, with low heating capacity, there is a lower temperature difference between the supply and return. With lower temperature difference and lower SC, the temperature approach is reduced and with it, the condensation pressure and pressure ratio decrease, increasing the COP at this point.

Table 16: Results of the undercharged case of BWHP for low-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	9.87	3.89	39.5	-5.6	59.9	3.9	5.0	187.5
B	70	6.28	4.65	33.7	-5.0	51.3	2.9	5.0	187.5
C	50	4.28	5.09	30.0	-5.0	47.3	3.8	5.0	187.5
D	30	2.37	5.47	26.1	-5.0	43.8	2.9	5.0	187.5
E	120	11.52	3.68	41.3	-6.0	63.2	5.0	5.0	187.5

Table 17: Results of the undercharged case of BWHP for medium-temperature.

Test condition	Comp speed (rps)	Heating capacity (kW)	COP	T_{cond} (ºC)	T_{evap} (ºC)	T_{dis} (ºC)	SC (K)	SH (K)	Ref charge(g)
A	105	9.11	2.79	55.4	-5.6	79.8	3.9	5.5	186.3
B	65	5.55	3.58	44.7	-5.0	65.0	4.5	5.0	186.3
C	45	3.67	3.93	38.6	-5.0	59.9	5.0	5.0	186.3
D	30	2.33	4.42	32.5	-5.0	53.4	4.1	5.0	186.3
E	120	10.49	2.59	58.6	-5.7	84.6	3.6	5.5	186.3

3.2 Discussion

Figure 1 shows the values of relative COP and relative Heating capacity. These are calculated by dividing the actual value of COP by the one obtained in the same conditions in the optimal campaign (SC=5K and SH=5K). As mentioned before, overcharging and undercharging the heat pump affects the COP in AWHP more than in the BWHP. This effect is mainly seen in test conditions A and B for overcharging and D and E when undercharging. Two different motives cause this difference: the internal volume of the evaporator is greater in the AWHP, and the source temperature remains constant in BWHP, while in AWHP has a significant variability.

The same differences can be seen in the heating capacity, but the heating capacity increases when overcharging the system.

In BWHP, similar results can be seen, but in some test conditions, overcharging or undercharging the system positively impacts the COP due to the temperature difference in the secondary fluid.

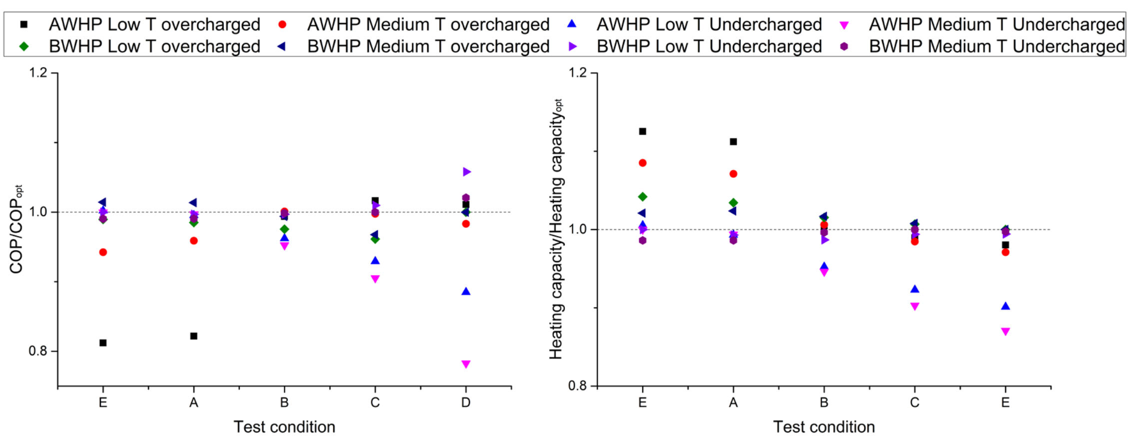

Figure 2: COP and heating capacity comparison.

Analyzing global results, shown in Figure 2, it can be seen firstly the same effect mentioned before AWHP shows a higher impact in COP than BWHP. Then, focusing on AWHP, at low-temperature overcharging affects more than undercharging and at medium temperatures, the opposite effect is seen. This is because the temperature difference in the water-side at medium-temperature (8K) is greater than at low-temperature (5K). Also, the limit imposed by the discharge temperature is nearer to the optimal case at medium-temperature than low-temperature.

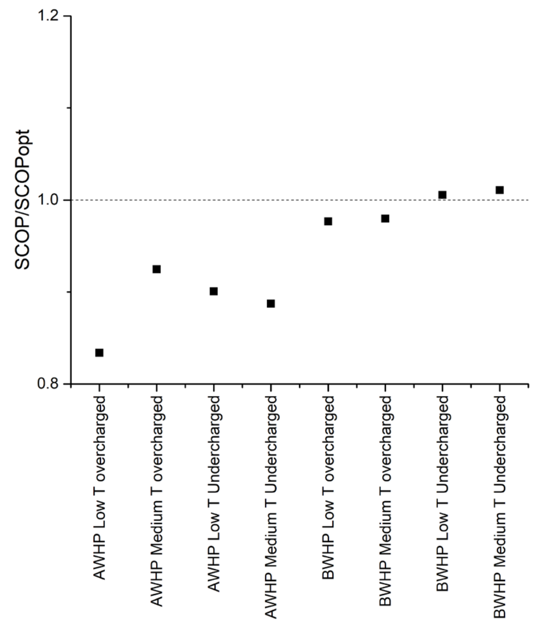

Figure 3: SCOP comparison.

In BWHP the most notable result is that the minimum charge is better than the defined optimum. However, this point is probably not functional as a reference refrigerant charge because of the effect of the possible refrigerant leakages.

4. CONCLUSIONS

From this work, the following conclusions can be highlighted:

- External condition variations affect the refrigerant needed to work with normal SC and SH levels in a great manner. For this reason, having a constant temperature source as ground or water will reduce the fluctuations, allowing the heat pump to work correctly during the whole year without a refrigerant receiver.
- In AWHP the fluctuation of charge during the year can cause a significant deterioration of the COP. In this case, the refrigerant receiver may not be removable.
- In a test where less heating capacity is required, the refrigerant charge is higher. The compressor's oil solubility increases because of the oil temperature reduction, the SC condenser area increases and the inlet quality in the evaporator increases as well.
- For BWHP it can be thought to work with undercharged conditions due to its good performance. However, at this point, if a little leakage appears in the circuit, the performance will worsen faster. It may be desirable to work with an intermediate refrigerant charge to have a safety margin in case of leakage.

For future work, it should be interesting to add to the study the effect of the frost and how the refrigerant charge affects to the COP, as it affects the evaporation temperature and the wall temperature, frost generation can be enhanced due to overcharging or undercharging the system. Also, it would be interesting to study the transient during the start-up and when the cycle is inverted to defrost the evaporator. This process needs typically a bit extra charge to not work under extreme conditions.

NOMENCLATURE

T	Temperature	(ºC)
Subscript		
w	water	
b	brine	
out	outlet	
in	inlet	
cond	condenser	
evap	evaporator	
dis	discharge	
Abbreviations		
CFC	Chlorofluorocarbons	
HCFC	Hydrochlorofluorocarbons	
HFC	Hydrofluorocarbons	
AWHP	air-water heat pump	
BWHP	brine-water heat pump	
BPHE	Brazed plates heat exchanger	
FTHE	Finned-tubes heat exchanger	
SCOP	Seasonal coefficient of performance	(-)
SC	Subcooling	(K)
SH	Superheating	(K)
Ref	Refrigerant	
Comp	Compressor	

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