Experimental Characterization Of A Condensing Units' Performance Under A Controlled Refrigerant Leakage

Etienne HADDAD, Assaad ZOUGHAIB
MINES ParisTech, PSL Research University, CES - Center for Energy efficiency of Systems,
5 rue leon blum, 91120 Palaiseau, France
etienne.haddad@mines-paristech.fr

ABSTRACT

The European regulation has set a series of limitation to reach the objective of reducing the high Global Warming Potential (GWP) gases emissions. Among these limits, the ban of Hydrofluorocarbons (HFC) refrigerants having a GWP higher than 2500 starting 2020 and a stringent leakage control on refrigeration equipments. Many continuous leakage detection technologies exist but they are not applicable in many cases such as open space or outside equipments. A software based leak detection technique may be interesting in these cases. This technique is based on machine data monitoring and data derived models. In order to be more precise and avoid outlier alarms, the sensitivity of usually measured variables such as temperatures and pressures should be correlated to refrigerant charge level.

In this paper, an experimental campaign is performed on a condensing unit under a controlled leakage. The tested condensing unit having a piston compressor of 16 m3/h is cooling a secondary glycol loop. It has a thermostatic expansion device and a liquid receiver. The condensing unit is first charged in a way to overfill the liquid receiver. A leakage is simulated by connecting the receiver's outlet to a micro valve that delivers the refrigerant to an external empty cylinder continuously weighted.

Different usually measured variables are recorded. These are pressures and inlet and outlet temperatures of condenser and suction temperature. Compressor electricity consumption and the produced cooling capacity are also monitored.

Results show the sensitivity of the high pressure and the subcooling to the charge level until the liquid receiver starts to be partially filled. The cooling capacity is also impacted.

1. INTRODUCTION

Refrigeration systems play a major role in the global electrical energy consumption. In 2015, the refrigeration sector accounted for 17% of the overall electrical energy used worldwide (IIR, 2015). Therefore, it is important to optimize the energy consumption of this sector. Also, the refrigerants used in these equipments contribute directly to the global warming. Therefore, the refrigeration sector impacts the global warming directly and indirectly and this impact should be mitigated. That can be done by improving the overall energy efficiency of the systems and by reducing refrigerant leakage.

Refrigerant leakage detection is done by direct and indirect methods. Direct methods consist of measuring the concentration of the refrigerant or its presence in the environment surrounding the system. We can cite the halogen leak detector (Rasmussen and Thorud, 2007), bubble detector, fluorescent detector and others. Direct methods were shown unable to detect small leakages which can degrade the system's efficiency over time. Indirect methods were then introduced.

Indirect methods consist of measuring system parameters that were shown to be affected by the system's refrigerant charge. We can find in the literature:

- Refrigerant charge was shown to have an impact on the sub-cooling and superheating of the system, and real time monitoring systems showed to be capable of detecting refrigerant leakage by observing these parameters (Tassou and Grace, 2005).
- Rossi and Braun (1997) also showed a correlation between refrigerant charge and sub-cooling degree.
- Yoo, Hong and Kim (2017) showed a relation between refrigerant charge and temperature differences between inlet air and mid-point of heat exchanger. They stated that it is possible to detect leakages using this method.

It was also shown that refrigerant charge is correlated with the overall system's efficiency.:

- Cho *et al.* (2005) showed that every refrigeration system has its own optimal charge.
- Kim *et al.* (2007) showed that the cooling capacity and the compressor's electrical consumption increase with increasing refrigerant charge, however there exists an optimal COP that corresponds to a certain optimal charge. This was also confirmed in further studies done by Kim *et al.* (2014).

We can conclude that optimal refrigerant charge has been an interesting topic due to the direct correlation between it and the overall efficiency of the system. Therefore, refrigerant leakage has a non-negligible impact on the performance of a refrigeration system.

The present paper aims to find an experimental correlation between certain measured parameters and the refrigerant charge of a refrigeration system. Since the studied system contains a liquid receiver, we will completely fill the liquid receiver to bypass it. We will then proceed to overcharge the system observe the effect of reducing this charge on the machine parameters.
This work is part of a much larger scope where we aim to develop an indirect leakage detection system based on a hybrid method coupling machine learning and physical modelling of the refrigeration systems.

2. SYSTEM AND TEST DESCRIPTION

2.1. Experimental setup

The experimental setup is constituted of a temperature controlled room (climatic room) representing outdoor conditions and a 40% propylene glycol loop representing the cooling load. This experimental setup is shown in Figure 1.
The 40% water-glycol loop is equipped with: a water-glycol storage tank of 200 L, a heating device and a circulating pump to control the water-glycol flow rate and temperature. Water-glycol temperature is measured at the inlet and outlet of the condensing unit's evaporator using 4 thermocouple sensors (2 on the inlet and 2 on the outlet) with an accuracy of ±0.5°C. An electromagnetic flow meter with an accuracy of ±1% is used to measure the water-glycol volumetric flow rate.
The condensing unit is installed in the climatic room which is equipped with a cooling coil permitting to cool and control the room's temperature by compensating the heat rejected by the unit (on the condenser side). Air temperature is measured at the condenser air inlet using a PT100 temperature sensor with an accuracy of ±0.3°C.
Figure (1) shows the experimental setup of the condensing unit and the glycol water loop.

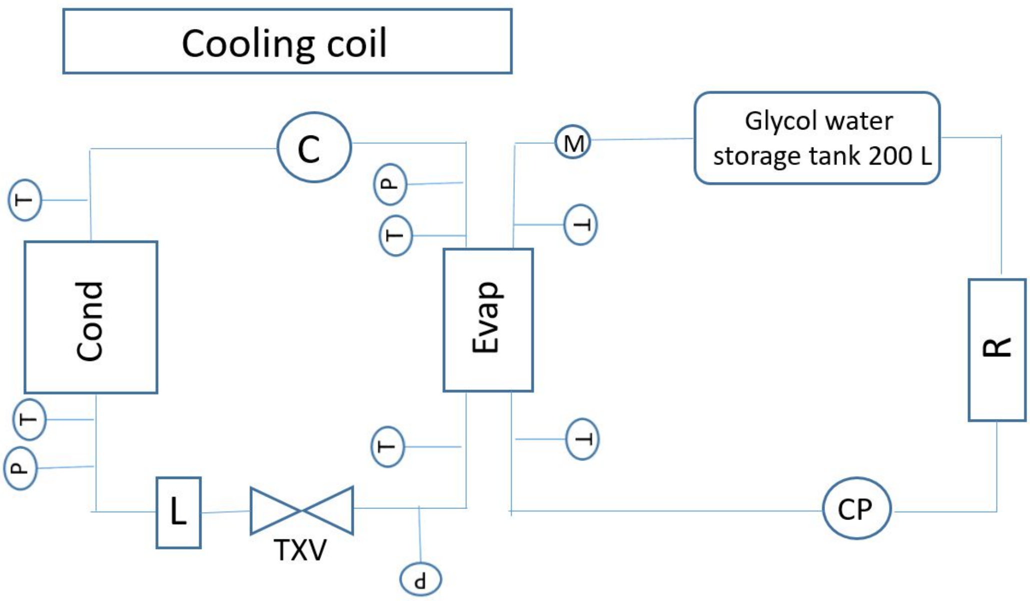

Figure 1: Experimental setup for the climatic room, the water-glycol loop and condensing unit.
C: Compressor; CP: Circulation Pump; Cond: Condenser; Evap: Evaporator; L: liquid receiver; M: flowmeter; P: Pressure sensor; R: Heating Coil; T: Temperature sensor; TXV: Thermostatic Expansion Valve

2.2. Description of the system and performed tests

R-134a is used as the working fluid during the tests.

According to its manufacturer, the condensing unit has the following characteristics:

- Reference refrigerant: R-134a
- Nominal refrigerant charge: 8.6 kg
- Nominal refrigerating capacity: 5.22 kW (at 20°C suction gas temperature, 1°K liquid sub-cooling, 32°C ambient air temperature and -10°C evaporating temperature)
- Nominal COP = 2.49

For the measurements of temperature and pressure on the working fluid, thermocouples with an accuracy of ±0.5°C and pressure sensors with an accuracy of ±15kPa are installed at inlets and outlets of the condensing unit components.

Since our system has a liquid receiver, changing the refrigerant's charge in the system will have no effect on the system's parameters unless the liquid receiver is completely filled (overcharged) or completely empty (undercharged). In this paper, we will aim to overcharge the system by completely filling the liquid receiver thus bypassing it.

Before proceeding with the overcharging tests, we compared experimental results of cooling capacities and system COP with those of the system manufacturer's datasheet. These tests allowed verifying the test methodology and the sensors' reliability. The tests were done at 29°C, 38°C and 43°C ambient temperature, and -2.5°C evaporating temperature (figure 2).

After validating our measurement methodology, we proceeded to overcharge the system. We started by slowly increasing the refrigerant charge in the condensing unit until we started seeing changes in certain parameters measured. The focus will be mainly on the high pressure the sub-cooling degree, the cooling capacity and the system COP, that have shown sensibility to overcharging in previous studies.
Once these parameters start showing clear variations (meaning the liquid receiver is completely filled), we added an extra refrigerant charge in the system. The tests then started and a leakage was simulated by a valve that allows to extract a controlled refrigerant amount from the receiver to a cylinder continuously weighted. The balance used is a digital balance with a ± 5 grams accuracy.

We then proceeded to discharge the system at five steps:

- First step: 575 g overcharged
- Second step: 170 grams discharged
- Third step: 170 grams discharged
- Fourth step: 160 grams discharged
- Fifth step: 75 grams discharged

At each step, we waited for the system to stabilize and then recorded values for 20 minutes with 5 seconds intervals between recordings, totalling around 240 points recorded for each step. We obtained five stabilized sets of points (including the reference charge point).

The tests were done at a constant ambient temperature of 38°C.
During all tests, the evaporator's inlet water-glycol temperature is controlled in order to maintain an inlet glycol temperature of 13.7°C. All tests are conducted with 0.291 l/s glycol-water volume flow rate.

2.3. Results and discussions

For all realized tests, the cooling capacity is calculated by two means:

- The first relatively to water-glycol mass flow rate and water-glycol temperature at the inlet and outlet of the evaporator according to eq. (1).

$$Q_{cold} = \dot{m}_w c_{p,w}\left(T_{in,glycol} - T_{out,glycol}\right) \quad (1)$$

- The second relatively to the refrigerant's mass flow rate at the evaporator and refrigerant enthalpies at the inlet and outlet of the evaporator according to eq. (2).

$$Q_{cold} = \dot{m}_{f,evap}(h_{out,evap} - h_{in,evap}) \quad (2)$$

The cooling capacity calculated by using eq. (1) presents an underestimation of 10% compared to the cooling capacity calculated using eq. (2). An uncertainty study (using the Monte Carlo statistical method), taking into account the accuracy of the sensors, showed that the error on the cooling capacity using the first method is around 20%, and the error using the second method is around 5%. Eq. (2) gives closer results to the ones announced by the condensing unit's manufacturer, and presents a lower uncertainty, it will be used for all further calculations in this paper.

The system's power consumption is measured using a Wattmeter. The value measured includes the compressor's power consumption and the fan power consumption.

System coefficient of performance COP_{system} is calculated according to eq. (3). COP_{system} is the ratio of the cooling capacity to the system power consumption, which includes the condenser's fan power consumption.

$$COP_{system} = \frac{Q_{cold}}{W_{comp}+W_{fan}} \quad (3)$$

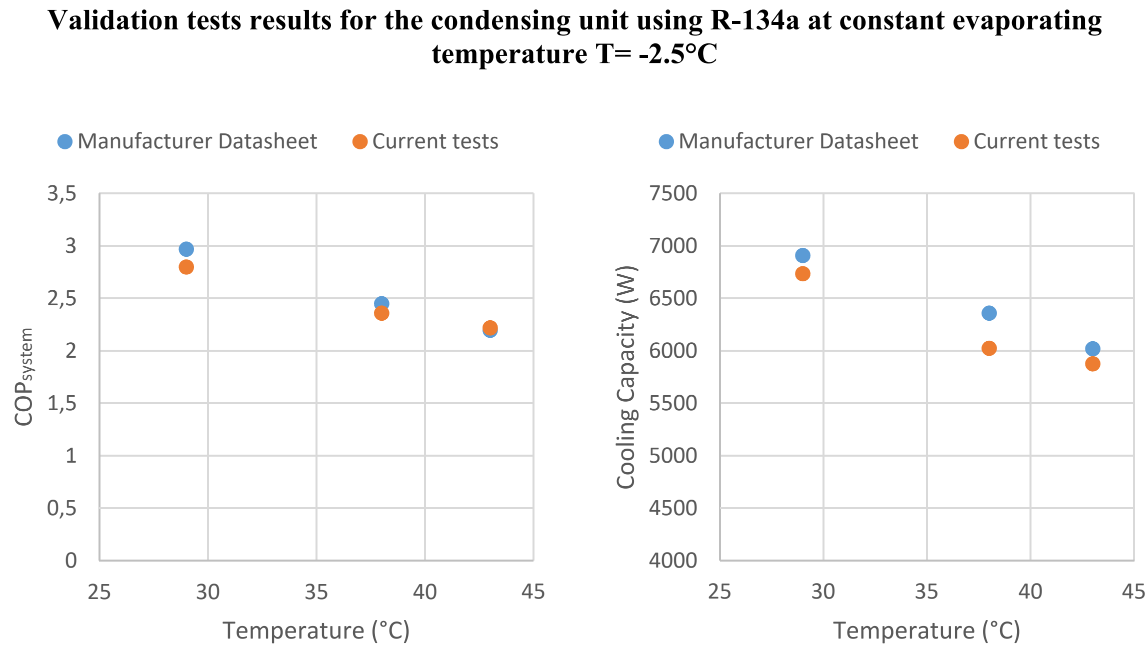

Figure 2: COP_{system} and cooling capacity (W) comparison of current tests with R-134a with manufacturer datasheet at T_{evap}=-2.5°C (before overcharging)

These tests allowed verifying the test methodology by comparing the determined performance with R-134a with the manufacturer's datasheet.
The results in Figure 2 show that, at the same evaporating temperature, the test results show very close trends and values compared to the manufacturer declared performances at different ambient temperatures. For the cooling capacity, the maximum difference is 5% while for the COP_{system}, the maximum difference is 5.7%. Therefore, the experimental protocol presented in this paper is validated and we can proceed to test the sensibility of certain parameters to the charge variation.

Different parameters evolution at constant inlet glycol temperature T= 13°C

Table 1 details the average values of different parameters during the tests at different charges. 575g is the initial reference charge. 0 g is the equivalent charge after the discharge.

Discharge	575g	170 g (405g total)	170 g (235g total)	160 g (75 g total)	75 g (0 g total)
Ambient Temperature	39.39°C	38.91 °C	38.33 °C	37.89 °C	37.2 °C
High Pressure (bar)	16.7	15.27	14.36	13.71	13.4
Low Pressure (bar)	3.05	3.02	3.01	3	2.99
Sub-cooling (K)	18.816	15.25	12.56	10.836	9.44
Q condensation (W)	10531.826	10541.87	10480.283	10445.806	10541.45
Q evaporation (W)	7817.43	7914.98	7913.215	7920.98	8038.55
COP_{system}	2.758	2.88	2.944	2.9948	3.064

Table 1: Evolution of different parameters at different charges

We notice that the low pressure remains almost constant. The following graphs show the evolution of the high pressure, the sub-cooling, the cooling capacity and the system COP as a function of the charge.

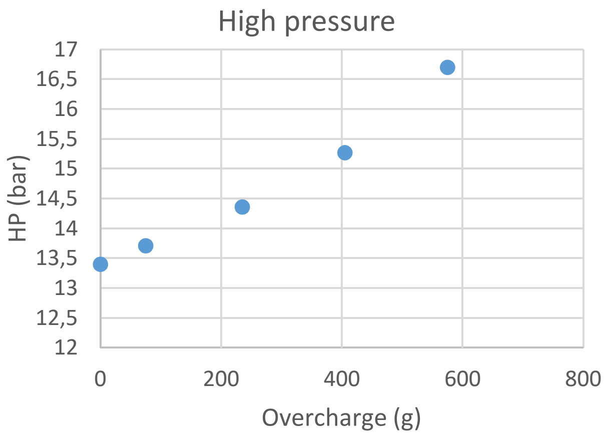

Figure 3: Evolution of the high pressure in function of the charge

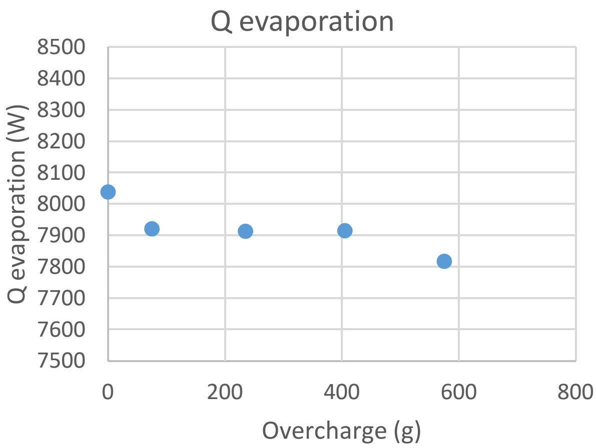

Figure 4: Evolution of the cooling capacity in function of the charge

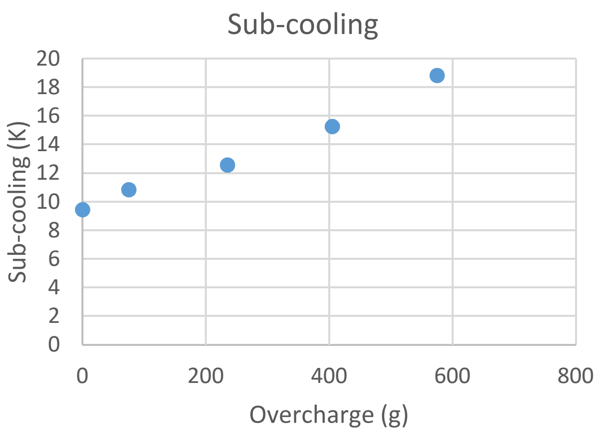

Figure 5: Evolution of the sub-cooling in function of the charge

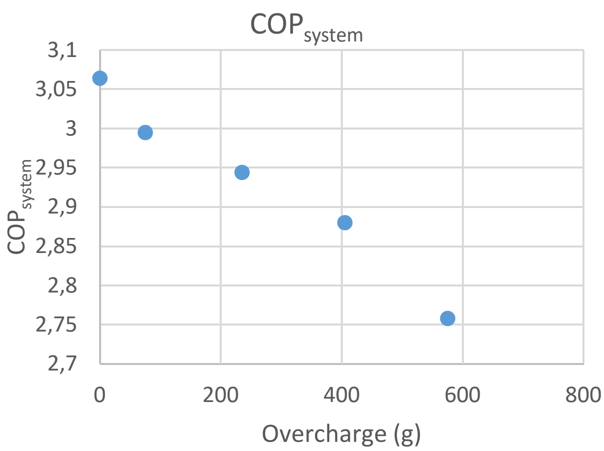

Figure 6: Evolution of the system COP in function of the charge

We observe that the discharge pressure and the sub-cooling increase with the overload. This is because the added fluid will be stored in the condenser. This additional fluid charge will cause the increase of the pressure at the condenser, which will increase the density of the vapour part, therefore giving more volume to the liquid part. The increase in high pressure is mainly due to the smaller surface area of the condenser dedicated to condensation, the remaining surface area filled with liquid will exchange heat between the liquid and air, therefore increasing the sub-cooling degree.

The discharge pressure decreases from 16.7 bar to 13.4 bar with the discharge of 575 g. However, this variation is not linear. We can note that the rate at which the high-pressure changes, increases at higher charge. For example, discharging 170 g (from 575 g to 405 g) decreased the high pressure from 16.7 bars to 15.27 bars. Discharging 235 g (from 235 g to 0 g) decreased the high pressure from 14.36 bars to 13.4 bars. As for the sub-cooling, it decreased in a near linear manner, from 18.816 K to 9.44 K with the discharge of 575g.

Discharging 575 g caused an increase of the cooling capacity from 7817.43 W to 8038.55 W, which is equivalent to an increase of 2.75%. It should be noted that the ambient temperature varied by 2 °C between the first and final test, which may also impact the cooling capacity. Therefore, we cannot directly correlate the increase in cooling capacity to the discharge.
The system COP increased from 2.758 to 3.0642 with the 575 g discharge. This equates to a 9.98% increase. Even with a decrease of 2 ° C in ambient temperature, this drop in COP can be strongly correlated with the discharge that causes less consumption at the compressor level. This is because the compressor has to do less work to compress the fluid to the new lower pressures that are decreasing with discharge.

Uncertainty study

The pressure and temperature sensors used during the tests have a certain uncertainty. The pressure sensors have an accuracy of ± 15 kPa and the temperature sensors have an accuracy of ± 0.5 °C. The wattmeter used to measure the power consumption has an accuracy of ± 1%. This causes an uncertainty on the cooling capacities and the system COP calculated during the different tests. Using the Monte-Carlo statistical method, we present this uncertainty in Table 2.

Overcharge (g)	575	405	235	75	0
Q cooling (W)	7817.43	7914.98	7913.215	7920.98	8038.55
Error	245,610385	239,744604	257,705751	264,413638	273,124562
COP_{system}	2,75157292	2,88402068	2,9593982	3,02081294	3,09248161
Error	0,08644221	0,09455366	0,09891474	0,09735605	0,10138736

Table 2: Uncertainty study on the cooling capacity and the system COP

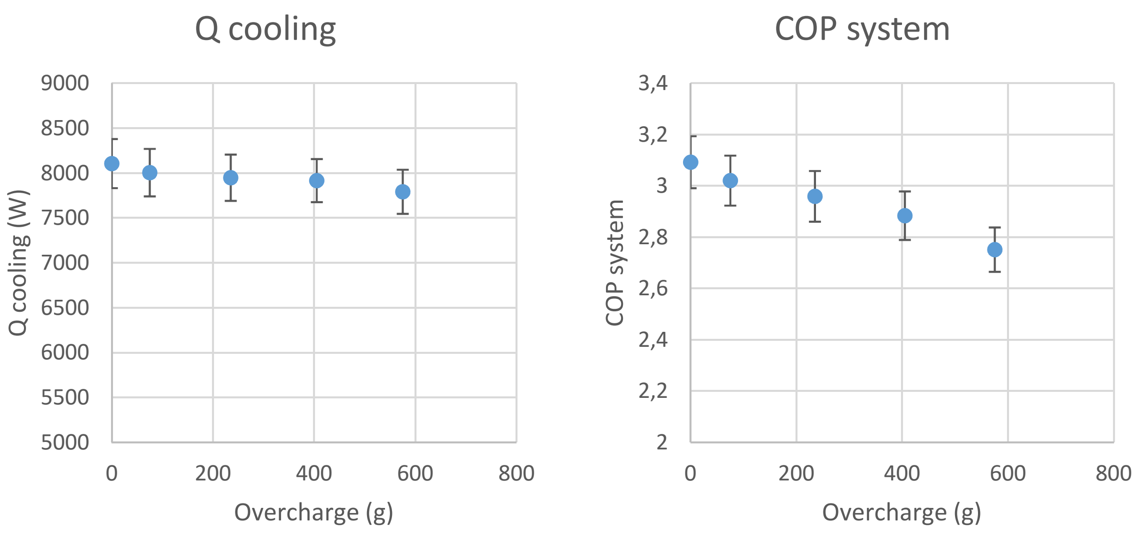

Figure 7: Cooling capacity and COP system with error bars

According to this uncertainty study, we can conclude that the error on the cooling power shows that there is no clear relation between the overcharge and the cooling capacity. However, even with the error bars, we can see a decreasing trend of the system COP the more we overcharge.

3. Conclusion

In this paper, we tested the effect of charge change (leakage) on the high pressure, the sub-cooling degree, the cooling capacity and the system COP. The studied system is a condensing unit cooling a glycol water loop. The tests were done at 38 °C and the working fluid used is R-134a. We discharged the system in four successive steps: 170g, 170 g, 160 g, and 75 g, totalling a 575 g discharge. We observed a decrease in the high pressure that decreased at a higher rate at the beginning. From 0g to 575g discharge, the high pressure decreased from 16.4 bars to 13.7 bars. The sub-cooling degree decreased from 18.82 K to 9.44 K. The system COP increased from 2.75 to 3.06. The cooling capacity did not show any clear relation to the overcharge and remained constant. The uncertainty study taking into account the sensors' accuracy confirmed the conclusions made.
This study is part of a larger scope, where we aim to use machine learning combined with system modelling, to detect leakages by measuring parameters that are sensible to variations of the working fluid's charge. We concluded in this paper that there is a direct relation between the working fluid's charge and each of the high pressure, the sub-cooling, and the compressor's power consumption. Further studies testing an undercharged system could show a relation between the working fluid's charge and other parameters.

4. Nomenclature

Q_{cold} = cooling capacity (W)
$\dot{m}_w$ = water glycol mass flow rate (kg/s)
$c_{p,w}$ = water glycol specific heat capacity (J/kg/K)
$T_{in,glycol}$ = water glycol temperature at the inlet of the evaporator (°C)
$T_{out,glycol}$ = water glycol temperature at the outlet of the evaporator (°C)
$\dot{m}_f$ = working fluid total mass flow rate (kg/s)
$\dot{m}_{f,evap}$ = working fluid mass flow rate at the evaporator (kg/s)
$h_{in,fluid}$ = working fluid enthalpy at the inlet of the evaporator (J/kg)
$h_{out,fluid}$ = working fluid enthalpy at the outlet of the evaporator (J/kg)
W_{comp} = compressor power consumption (W)
W_{fan} = fan power consumption (W)
COP = coefficient of performance
COP_{system} = system's coefficient of performance

5. References

Cho, H. *et al.* (2005) 'Effects of refrigerant charge amount on the performance of a transcritical CO2 heat pump', *International Journal of Refrigeration*, 28(8), pp. 1266–1273. doi: 10.1016/j.ijrefrig.2005.09.011.

IIR (2015) 'The Role of Refrigeration in the Global Economy'. 29th Informatory Note on Refrigeration Technologies.

Kim, D. H., Park, H. S. and Kim, M. S. (2014) 'The effect of the refrigerant charge amount on single and cascade cycle heat pump systems', *International Journal of Refrigeration*, 40, pp. 254–268. doi: 10.1016/j.ijrefrig.2013.10.002.

Kim, J. H. *et al.* (2007) 'Circulation concentration of CO2/propane mixtures and the effect of their charge on the cooling performance in an air-conditioning system', *International Journal of Refrigeration*, 30(1), pp. 43–49. doi: 10.1016/j.ijrefrig.2006.06.008.

Rasmussen, H. and Thorud, S. (2007) ‘Using a refrigerant leak detector to monitor waste gases from halogenated anesthetics’, *J. Am. Assoc. Lab. Anim*, (46), pp. 64–68.

Rossi, T. M. and Braun, J. E. (1997) ‘A statistical, rule-based fault detection and diagnostic method for vapor compression air conditioners’, *HVAC&R Research*, 3, pp. 19–37.

Tassou, S. A. and Grace, I. N. (2005) ‘Fault diagnosis and refrigerant leak detection in vapour compression refrigeration systems’, *International Journal of Refrigeration*, 28(5), pp. 680–688. doi: 10.1016/j.ijrefrig.2004.12.007.

Yoo, J. W., Hong, S. B. and Kim, M. S. (2017) ‘Refrigerant leakage detection in an EEV installed residential air conditioner with limited sensor installations’, *International Journal of Refrigeration*, 78, pp. 157–165. doi: 10.1016/j.ijrefrig.2017.03.001.