Test of condensing units' performance and operating limits using different refrigerants

Etienne HADDAD(*), Samer SAAB(**), Assaad ZOUGHAIB(*)
(*)MINES ParisTech, PSL Research University, CES - Center for Energy efficiency of Systems, 5 rue leon blum, 91120 Palaiseau, France
(**) Chemours France S.A.S., Villers-Saint-Paul, Rue Frédéric Kuhlmann, 60871 Rieux, France
etienne.haddad@mines-paristech.fr
samer.saab@chemours.com

ABSTRACT

The European regulation has set a series of limitation to reach the objective of reducing the Global Warming Potential (GWP) emissions. Among these limits, the ban of Hydrofluorocarbons (HFC) refrigerants having a GWP higher than 2500 starting 2020. To replace these refrigerants, a new family of refrigerants was invented: hydro-fluoro-olefins (HFOs), also new systems were developed using CO2.

In this paper, we present the outcome of a testing campaign aiming to compare the performance, operating range and cost of two condensing units of a rated capacity of 8kW cooling a secondary glycol loop. One of these machines, initially operated with R-404A, is operated with low GWP blends containing HFO while the other is using CO2.

The comparison includes traditional HFC refrigerant (R-404A), 3 different commercial HFOs containing blends and transcritical CO2. The comparison is made using different ambient temperatures to have a fair comparison.

A special focus is done analysing the glide impact on the condensing unit behaviour and performance. The glide matching blend (R-454C) shows an increased performance while the other refrigerants are penalized.

Finally, HFOs containing blends were found more efficient and capable of operating at a wider temperature range compared to CO2.

1. INTRODUCTION

Lately, R-404A has been widely used in condensing units and refrigeration systems. Nowadays, EU regulations aiming to limit the GWP of used refrigerants are affecting systems using R-404A. Having a GWP of 3922, R-404A will not be able to cope with the EU regulation No 517/2014 that bans the use of refrigerants having a GWP>2500 (Mota-Babiloni *et al.*, 2015).

Studies have shown that new blends of refrigerants having low GWP are capable of replacing R-404A. We can find in the literature:

- A retrofit test of R-404A showed the possibility of adopting R-407F (GWP=1825) and R-410A (GWP=2088) in a condensing unit operating at medium temperature [-5;10]°C, and R-407F in a condensing unit operating at low temperature [-25;-15]°C (Bortolini *et al.*, 2015).
- Another suitable replacement of R-404A in an indirect supermarket refrigeration system is R-449A refrigerant blend (GWP=1397) (Makhnatch *et al.*, 2017). Measurements showed a decrease in the energy consumption of 4% when using R-449A instead of R-404A (Vaitkus and Dagilis, 2017).
- A retrofit test showed that R-442A (GWP=1888) refrigerant blend presents an 8% increase in cooling capacity and a 12% increase in COP compared to R-404A (Oruç, Devecioğlu and Ender, 2018).

Besides looking at the replacement refrigerant's low GWP, the overall energy efficiency of the system should be taken into account: if the use of a new refrigerant results in a lower overall energy efficiency of the system, it is likely to be more harmful than beneficial based on net global warming impacts (Calm, 2008). The cited studies tested replacement refrigerants having relatively low GWP however at long term the GWP reduction should be more drastic (<150).

This is why, alongside low GWP refrigerant blends, carbon dioxide (R-744) is argued to be a good replacement for CFCs and HCFCs (Lorentzen, 1994). It has many advantages such as non-toxicity, low cost, zero ozone depletion potential and a GWP=1 (Haida *et al.*, 2018). It is also an abundant fluid in nature (Hu *et al.*, 2015).
However, Mitsopoulos *et al.* (2019) showed that R-744 has a 15.23% higher yearly electricity consumption than R-404A.

The present paper aims to test very low GWP replacement candidates for R-404A in a condensing unit. The replacement fluids tested are R-454C (GWP=148), R-454A (GWP=239), R-455A (GWP=148) and R-744 (GWP=1). R-454C, R-454A and R-455A belong to the mildly flammable class A2L.

First tests are performed on the condensing unit using R-404A. These tests determine a reference to draw conclusions concerning R-404A replacement candidates based on COP and cooling capacity values. R-454C, R-454A and R-455A refrigerant blends are tested on the same condensing unit as R-404A. R-744 is tested on a different condensing unit due to its much higher operating pressures which impose a different condensing unit's configuration. The compressor's discharge temperature will also be studied.

2. SYSTEM AND TEST DESCRIPTION

2.1. Experimental setup

The experimental setup is constituted of a temperature controlled room (climatic room) representing external 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.
R-404A, R-454C, R-454A and R-455A are tested on the same condensing unit (A) since they have similar properties and comparable operating pressures. As for R-744, its properties (operating pressures, densities…) are not comparable to those of the other fluids, so it must be tested on a different condensing unit (B) that has a different configuration.
Figure (1) and Figure (2) show the experimental setups using condensing unit (A) and condensing unit (B) respectively.

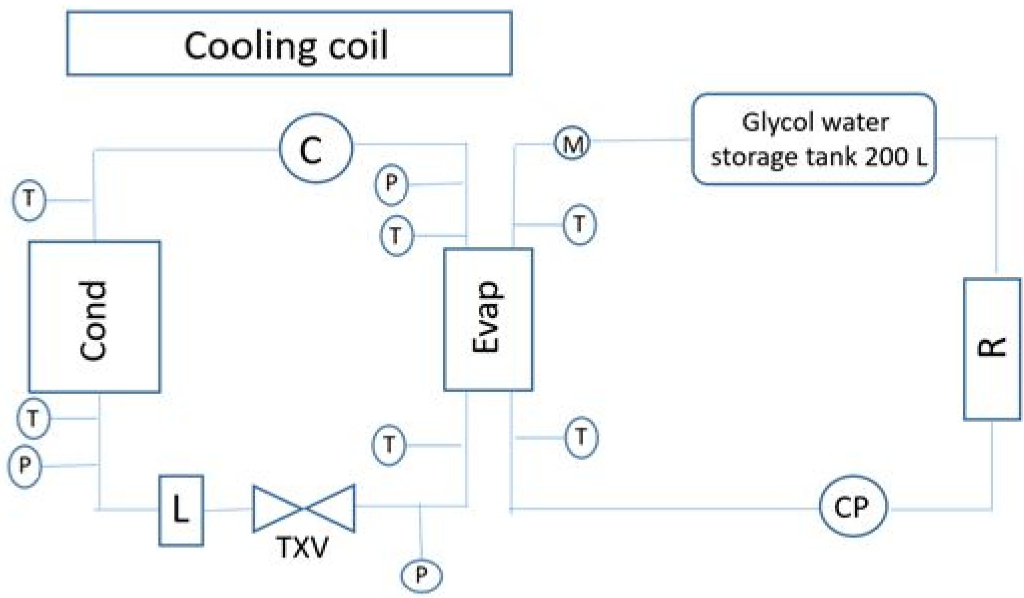


Figure 1: Experimental setup for the climatic room, the water-glycol loop and condensing unit (A).
C: Compressor; CP: Circulation Pump; Cond: Condenser; Evap: Evaporator; L: liquid receiver; M: flowmeter P: Pressure sensor; R: Heating Coil; T: Temperature sensor; TXV: Thermostatic Expansion Valve

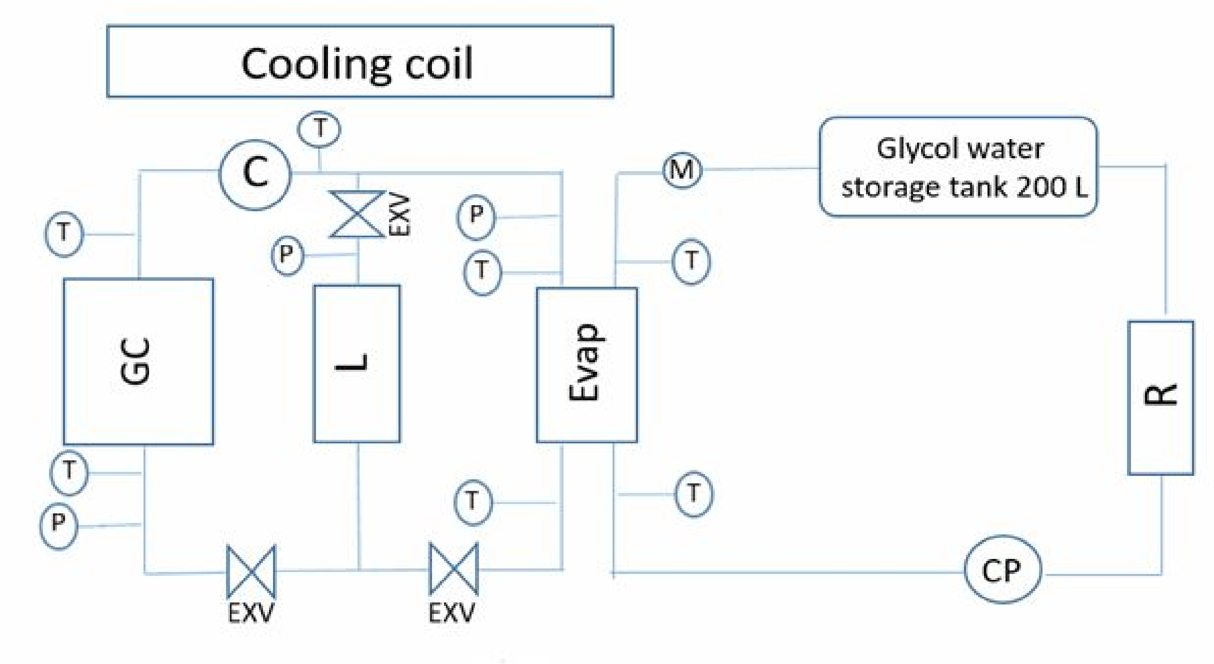


Figure 2: Experimental setup for the climatic room, the water-glycol loop and condensing unit (B).
C: Compressor; CP: Circulation Pump; Evap: Evaporator; EXV: Electronic Expansion Valve; GC: Gas Cooler L: Liquid receiver; M: Flowmeter P: Pressure sensor; R: Heating Coil; T: Temperature sensor

2.2. Description of the system and performed tests

a) According to its manufacturer, the condensing unit (A) has the following characteristics:
- Reference refrigerant R-404A
- Refrigerant charge 7.5 kg
- Nominal refrigerating capacity 8.12 kW (at 20°C suction gas temperature, 1°K liquid subcooling, 32°C ambient air temperature and -10°C evaporating temperature)
- Nominal COP = 2.96

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's components.

Barve and Cremaschi (2012) and Park and Jung (2007) suggested that during retrofit tests, the expansion device should be adjusted for each fluid in order to optimize the COP and control the superheat. They showed that these adjustments could be considered a minor change to the system. Therefore, during the tests on condensing unit (A), the expansion valve was adjusted with each refrigerant by the use of the setting spindle on the valve in order to reduce high refrigerant mass flow fluctuations observed during some of the tests and which was due to hunting phenomenon and to have comparable superheating at the evaporator outlet (7 K).

For safety reasons, tests on condensing unit (A) were limited to 43°C ambient air temperature.
For R-404A and for each of tested low GWP refrigerant blends, the tests are conducted for 7500g refrigerant charge (nominal charge for the equipment).

For each refrigerant, the condensing unit (A) is tested at full load with a constant average evaporating temperature of -10°C, at the following ambient air temperatures: 15°C, 25°C, 32°C, 38°C, and 43°C.

These tests also allowed to validate the measured performance level according to the manufacturer's R-404A datasheet.

b) According to its manufacturer, the condensing unit (B) has the following characteristics:
- Reference refrigerant R-744
- Refrigerant charge 13 kg
- Nominal refrigerating capacity 8.69 kW (at 10K suction gas superheat, 32°C ambient air temperature and -10°C evaporating temperature)
- Nominal COP = 1.47

For the measurements of temperature and pressure on the working fluid, temperature and pressure sensors are built in the condensing unit at inlets and outlets of the gas cooler, the liquid receiver and the compressor. A pressure sensor with an accuracy of ±15kPa was also installed at the evaporator's inlet to double check the evaporating temperature.

For safety reasons, tests on condensing unit (B) were limited to 38°C ambient air temperature. At this ambient temperature, the refrigerant's temperature at the gas cooler's inlet was 127 °C.

The condensing unit (B) is tested at full load with a constant average evaporating temperature of -10°C, at the following ambient air temperatures: 25°C, 27°C, 32°C, and 38°C.

Tests at these ambient temperatures resulted in a transcritical behaviour of R-744. Tests were not conducted at 15 °C due to limitations on the cooling coil power.
These tests also allowed to validate the measured performance level according to the manufacturer's R-744 datasheet.

During all tests on both condensing units, the evaporator's inlet water-glycol temperature is controlled in order to maintain an average evaporating temperature of -10°C. All tests are conducted with 0.267 l/s glycol-water volume flow rate.

2.3. Results and discussions

For all performed 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).

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

- 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).

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

In condensing unit (B) the total refrigerant's mass flow rate (after the compressor) is not equal to the refrigerant's mass flow rate at the evaporator. The fraction of the total mass flow rate that goes to the evaporator is equal to the liquid quality at the liquid receiver's inlet. A step by step reasoning is explained below:

- The enthalpy at the gas cooler's outlet is calculated from the pressure and temperature sensors placed at the gas cooler's outlet.
- The electronic expansion valve between the gas cooler and the liquid receiver will result in an isenthalpic expansion such that:

$$h_{out,GC} = h_{in,L} \tag{3}$$

- Having the enthalpy and the pressure (using the pressure sensor) at the liquid receiver's inlet, we can calculate the liquid quality x_{liq}.
- Finally, we can calculate the mass flow rate at the evaporator from the equation:

$$\dot{m}_{f,evap} = \dot{m}_f \times x_{liq} \tag{4}$$

An uncertainty analysis (not presented because of lack of space) shows that the cooling capacity calculated by using eq. (1) presents an uncertainty level of 20% while the cooling capacity calculated using eq. (2) has a 5% uncertainty. For this reason, eq. (2) will be used for all further calculations in this paper.

Both systems 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. (5). COP_{system} is the ratio of the cooling capacity to the system power consumption which includes the condenser's fan power consumption.

$$COP_{system} = \frac{\dot{Q}_{cold}}{\dot{W}_{comp} + \dot{W}_{fan}} \tag{5}$$

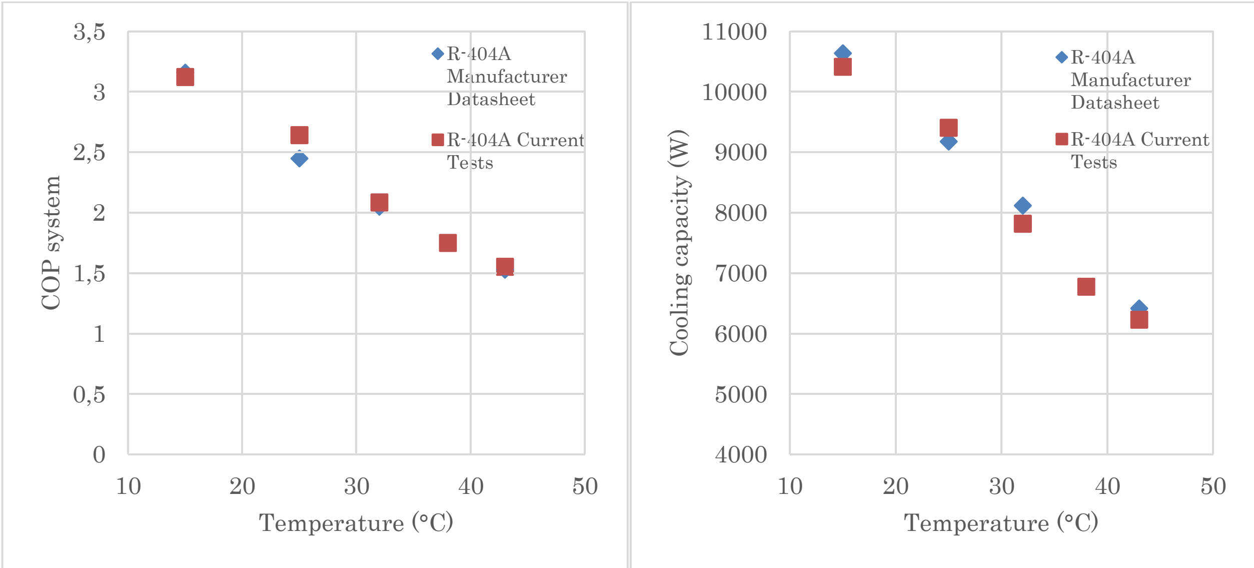


Figure 3: COP_{system} and cooling capacity (W) comparison of current tests with R-404A with manufacturer datasheet at T_{evap}=-10°C

These tests allowed verifying the test methodology by comparing the determined performance with R-404A with the manufacturer's datasheet.
The results in figure 3 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 3% while for the COP_{system}, the maximum difference is 5%. Therefore, the experimental protocol presented in this paper is validated for condensing unit (A) and we can proceed to the testing of R-454C, R-454A and R-455A.

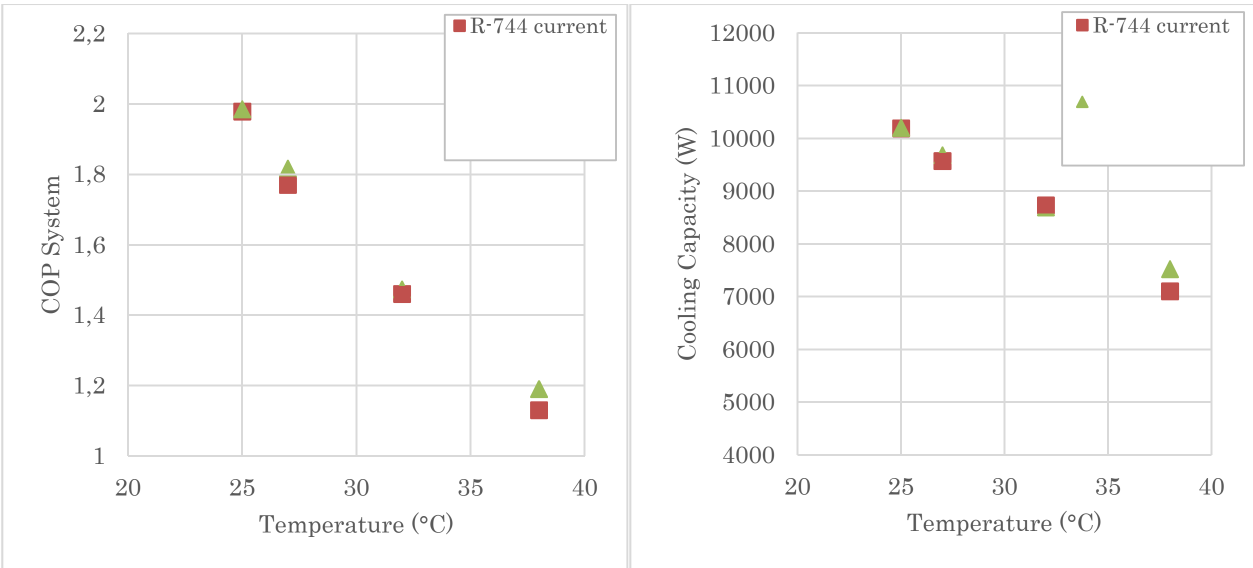
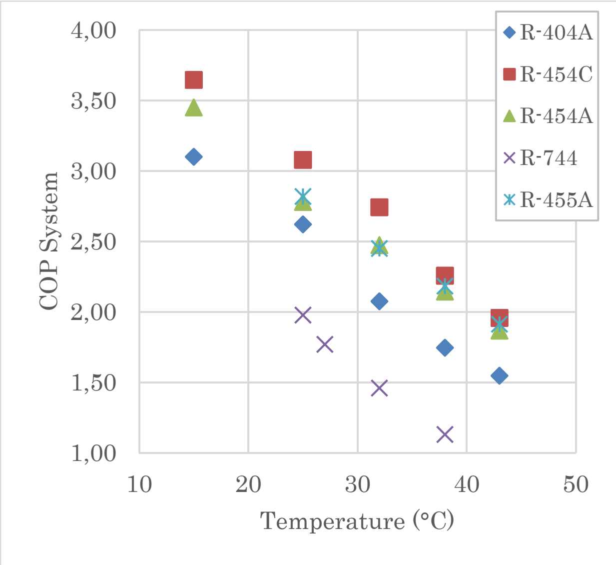


Figure 4: COP_{system} and cooling capacity (W) comparison of current tests with R-404A with manufacturer datasheet at T_{evap}=-10°C

These tests allowed to verify the experimental results on R-744 compared to the manufacturer's declared performances. For the cooling capacity, the maximum difference is 5.5% while for the COP_{system}, the maximum difference is 4.9%. Therefore, the experimental protocol presented in this paper is validated for condensing unit (B) and we can have an unbiased performance comparison between R-744 and the other tested fluids



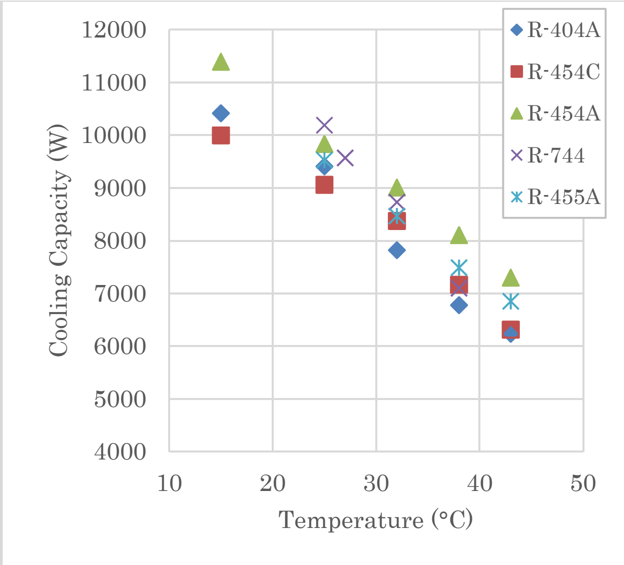


Figure 5: COP_{system} and cooling capacity (W) comparison at T_{evap}=-10°C

The results in figure 5 show that, at the same evaporating temperature, R-744 presents the lowest COP_{system} followed by R-404A, R-454A, R-455A and R-454C which represents the highest COP_{system} values. For the cooling capacity, R-404A and R-454C present comparable cooling capacities for the tested ambient temperatures. R-455A presents intermediate values. R-454A presents higher values than R-404A, R-454C and R-455A. R-744 tested in condensing unit B has a different behavior in term of cooling capacity, it has the highest cooling capacity at 25°C but intermediate cooling capacities at the other tested ambient temperatures.

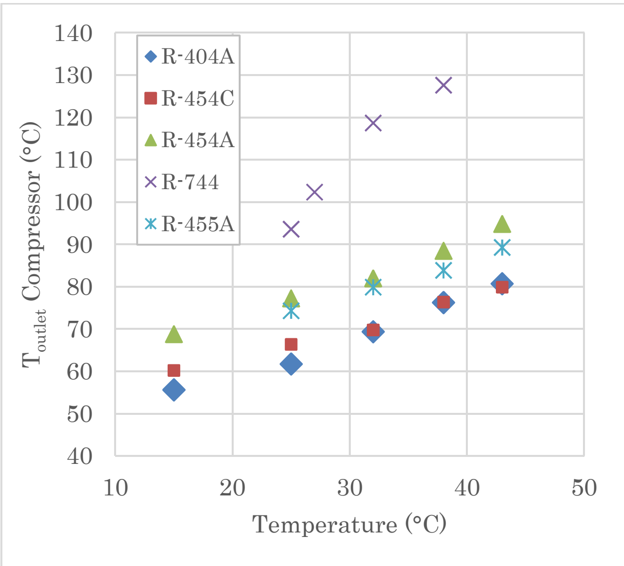


Figure 6: Compressor's outlet temperature at different ambient temperatures (°C)

The condensing unit (A)'s compressor's outlet temperature was measured using a skin thermocouple (type K) with an accuracy of ±0.5°C. The condensing unit (B)'s compressor's outlet temperature was measured using a built-in immerged temperature sensor. R-404A, R-454C, R-454A and R-455A present discharge temperatures lower than 100 °C as we can see in Figure 6. These temperatures are much lower than that of other low GWP refrigerant blends mentioned in other articles. R-32 showed a discharge temperature higher than 120°C at 40°C ambient temperature (Hanna, Ortego and Zoughaib, 2015). As for R-744, at 38°C ambient temperature, the compressor's discharge temperature is 127.7°C with an operating pressure of 101.7 bars. Temperatures and pressures at this level are extreme and could lead to compressor's faults and failure therefore we limited our tests to 38°C ambient temperature. Thus, an inconvenience of R-744 is its extremely high operating pressures and discharge temperatures which limits its application range.

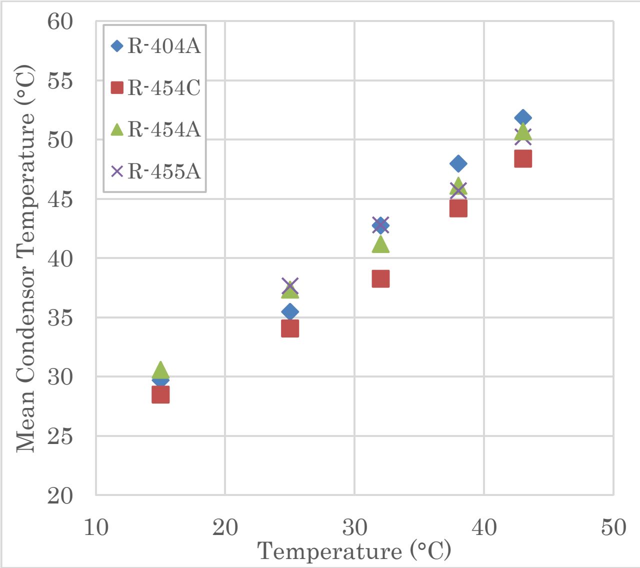


Figure 7: Mean condensing temperature at different ambient temperatures (°C)

The results presented in Figure 7 show that R-454C, R-454A and R-455A have a lower mean condensing temperature than that of R-404A. This, in part, can be explained by the temperature glide (4.5 K to 10 K) present in the low GWP refrigerant blends, and the heat exchanger's configuration presented in Figure 8.

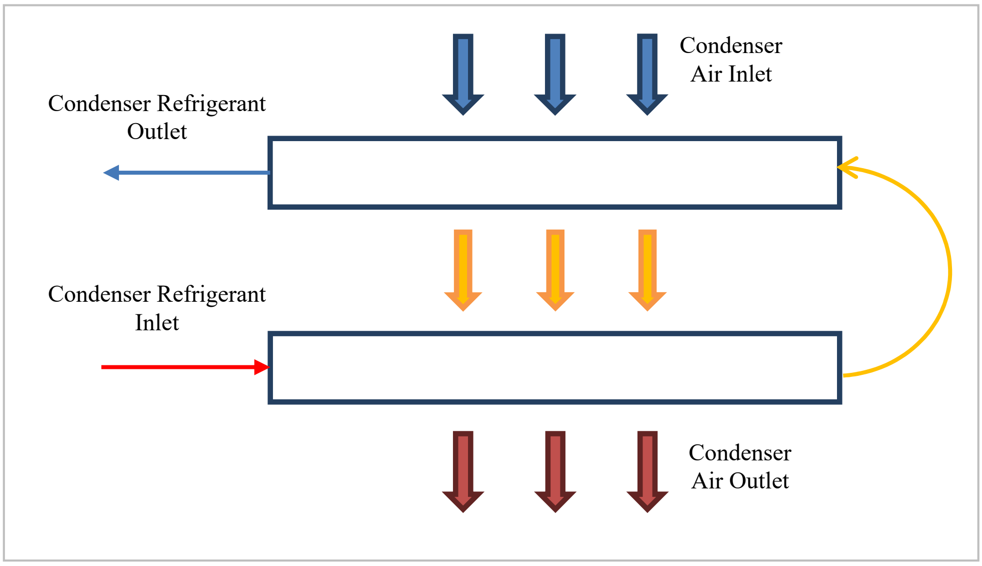


Figure 8: Tested condenser's configuration

The tested condenser is a two-pass counter cross current heat exchanger. This configuration was found to be highly beneficial in the case of working fluids having a moderate temperature glide (Bigot and Clodic, 2002). This resulted in higher COP_{system} when using the new low GWP refrigerant blends.

3. Conclusions

We will divide the results into two parts. We will start by comparing results of R-404A to R-454C, R-454A and R-455A. Then we will proceed to compare results of R-404A to R-744.
The results presented in this paper show that it is possible to replace R-404A in a condensing unit by R-454C, R-454A or R-455A without causing negative impacts on the COP_{system} and the cooling capacity. Under all conditions, these new refrigerants showed a higher COP_{system} than that of R-404A, while R-454C presented the highest COP_{system} among the

tested refrigerants. At constant mean evaporating temperature T=-10°C and ambient air temperatures of [15;25;32;38;43]°C, R-454C showed a similar cooling capacity than that of R404A, however R-454A and R-455A showed a higher cooling capacity than that of R-404A.

For the tests performed at constant mean evaporating temperature, R-454C showed a COP_{system} higher than that of R-404A by 17% to 33%, while the cooling capacity is more or less similar for both fluids. For the same evaporating temperature, R-454A showed a COP_{system} higher than that of R-404A by 6% to 23%. Also, R-454A showed gains in cooling capacity by 5% to 21% compared to R-404A. R-455A showed a COP_{system} higher than that of R-404A by 7.6% to 25.1% at the same evaporating temperature. R-455A also showed gains in cooling capacity by 2% to 11% compared to R-404A.

As for R-744, it showed losses in COP_{system} compared to R-404A by 33% to 54%. However, it showed a different behavior in terms of COP and cooling capacity when ambient temperature decreases. Indeed, the cooling capacity and COP vary steeper in function of the ambient temperature due to the particularity of the transcritical cycle.

This paper also studied the discharge temperatures for each refrigerant at different ambient temperatures. It was shown that for R-404A, R-454C, R-454A and R-455A, the discharge temperature is lower than 100 °C for all tested conditions. This temperature is considered safe for the compressor's components. However, tests realised on R-744 showed compressor's discharge temperatures ranging from 93.6°C to 127.7°C. These temperatures are considered high especially that the corresponding compressor's discharge pressures range from 76.46 bars to 101 bars. Thus, a condensing unit using R-744 should be handled carefully with extra safety measures. These high discharge temperatures also limit the maximum allowed ambient temperature.

The paper also showed that, for a condenser presenting a counter cross flow configuration, the average condensing temperature for refrigerants presenting a moderate temperature glide is lower than that of refrigerants that do not. This results in higher COP_{system} for R-454C, R-454A and R-455A compared to that of R-404A.

4. Nomenclature

GWP = global warming potential (kg_{eq} CO_2)
DB = dry bulb
$\dot{Q}_{cold}$ = 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)
$h_{out,GC}$ = working fluid enthalpy at the outlet of the gas cooler (J/kg)
$h_{in,L}$ = working fluid enthalpy at the inlet of the liquid receiver (J/kg)
$\dot{W}_{comp}$ = compressor power consumption (W)
$\dot{W}_{fan}$ = fan power consumption (W)
COP = coefficient of performance
COP_{system} = system's coefficient of performance
$T_{ambient}$ = ambient air temperature (°C)
x_{liq}= liquid quality at liquid receiver's inlet

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