Effects caused by the internal heat exchanger at the low temperature cycle in a cascade refrigeration plant

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ABSTRACT

This work analyses and quantifies the effects caused by the use of an internal heat exchanger (IHX) at the CO_2 subcritical cycle in an HFC134a/CO_2 cascade refrigeration plant that incorporates a gas-cooler at the exit of the low temperature compressor. Previous theoretical and experimental studies showed that the IHX reduces the refrigeration capacity and COP of the subcritical cycle, however, it has been seen that it also lowers the heat to be rejected at the condenser. This reduction, when the cycle is a part of a cascade system, allows reducing the heat load of the high temperature cycle, modifying the working conditions of the cascade plant. The modifications result in an increment of the overall coefficient of performance of the cascade system. The analysis here presented is based on the evaluation of an experimental HFC134a/CO_2 refrigeration plant, which has been analysed with and without internal heat exchanger in an evaporating temperature range from -40 to -30 ºC and in a condensing one from 30 to 50 ºC. The plant incorporates a gas-cooler at the exit of the CO_2 compressor. The experimental results confirm that the IHX slightly reduces the cooling capacity but it can increment the overall COP up to 3.7 %.

KEYWORDS

CO_2; subcritical; internal heat exchanger; cascade; experimental

HIGHLIGHTS

- The effects of an IHX in a CO_2 subcritical cycle with gas-cooler on a HFC134a/CO_2 cascade are analysed.
- The IHX reduces the cooling capacity between 1.1 to 2.4 %
- The IHX reduces the heat transfer in cascade heat exchanger between 4.4 to 5.5 %
- The IHX increases the COP of the cascade system up to 3.7 %

NOMENCLATURE

casc	Cascade heat exchanger
COP	Coefficient of performance
h	Specific enthalpy (kJ/kg)
HT	High temperature
HX	Heat exchanger
IHX	Internal heat exchanger
LT	Low temperature
$\dot{m}$	Refrigerant mass flow rate (kg/s)
N	Compressor speed (rpm)
P	Pressure (bar)
P_c	Electric power consumption (kW)
q_O	Specific cooling capacity (kJ/kg)
q_K	Difference of specific enthalpy in condenser (kJ/kg)
$\dot{Q}_{casc}$	Heat transfer rate in cascade heat exchanger (kW)
$\dot{Q}_{IHX}$	Heat transfer rate in internal heat exchanger (kW)
$\dot{Q}_K$	Heat transfer rate in condenser (kW)
$\dot{Q}_O$	Cooling capacity (kW)
S	Heat transfer area (m^2)
T	Temperature (ºC)
$\overline{U}$	Averaged overall heat transfer coefficient ($W/m^2{\cdot}K$)
$\dot{V}_G$	Compressor displacement (m^3/s)

GREEK SYMBOLS

Δ	Variation
η_v	Volumetric efficiency
ε	Thermal effectiveness
v	specific volume (m^3/kg)

SUBSCRIPTS

C	cascade heat exchanger
dis	Compressor discharge

env	Environment
exp	Expansion valve
g	Glycol, secondary fluid in evaporator
gc	Gas cooler
H	High temperature cycle
i	Inlet
ihx	Internal heat exchanger
K	Condensation
L	Low temperature cycle
O	Evaporation
o	Outlet
r	Refrigerant
suc	Compressor suction
w	Water, secondary fluid in condenser

1. Introduction

The new rules for the commercial refrigeration sector established by the F-Gas regulation in Europe [1], meaning the practical end of use of the R404A or R507A in medium and large centralized refrigeration systems from 2022 on, have expanded the use of cascade refrigeration systems using CO_2 as low temperature refrigerant, especially in supermarket refrigeration for low temperature [2]. Cecchinato et al. discussed the best options of using natural refrigerants in supermarket refrigeration [3] and integrating them with HVAC systems. Focussing on cascade systems with CO_2 as low temperature refrigerant, Sharma et al. [4] and Llopis et al. [5], using a theoretical approach, concluded that cascade systems using CO_2 as low temperature refrigerant are a good alternative to the current R404A direct expansion systems. They can present small reductions in COP but with high reductions of the Total Warming Equivalent Impact regards the classical systems, but with new designs, such as high efficient compressors or electronic expansion valves, they can overcome the firsts. This was experimentally verified by da Silva et al. [6] in a pilot supermarket plant, where a R404A/CO_2 cascade plant offered 22.3 % reduction of electricity consumption during a year of operation regards a conventional R404A direct expansion configuration.

Since the low temperature level of commercial refrigeration is around -30 ºC to maintain frozen products, CO_2 compressor manufacturers recommend the use of an internal heat exchanger (IHX) in the low temperature cycle to increase the suction temperature of the compressor. In fact, they recommend operating the compressors with a minimum superheat at suction of 20 K to avoid problems related with lubrication and to extend the useful life of the compressor. The IHX allows obtaining this increase in suction temperature providing at the same time a small subcooling of the liquid CO_2 at the exit of the condenser. This element has been widely analysed for plants working with HFC refrigerants [7-9] and for plants working with CO_2 in transcritical conditions [10, 11]. In general, its use is beneficial if the refrigerant has high heat capacity. For CO_2 in transcritical conditions its use has been experimentally corroborated to increase the energy performance of the plant.

However, the use of the IHX in CO_2 subcritical cycles has drawn little attention up to the moment. The unique available studies are the ones presented by Zhang et al. [12] and Llopis et al. [13]. Zhang et al. [12] evaluated theoretically the effect of the use of the IHX in CO_2 transcritical and subcritical cycles. For CO_2 subcritical cycles, they considered evaporating temperatures from -20 to -10 ºC and condensing temperatures from 10 to 20 ºC. Theoretically, they observed that the use of the IHX slightly lowered the COP and the capacity of the system, concluding that its use is not convenient for subcritical cycles. This theoretical results were corroborated experimentally by Llopis et al. [13] in a CO_2 subcritical cycle working with a semihermetic compressor and a gas-cooler at the exit of the compressor. For an evaluation range of -40 to -25 ºC of evaporating temperature and of -15 to 0 ºC of condensing temperature, they measured reductions in the capacity provided by the subcritical cycle ranging from 3.5 % at -40 ºC to 3.7 % at -25 ºC. They also measured reductions in the COP varying from -1.58 to -3.29 % at -25 ºC but small increases up to 0.45 % at -40 ºC. This experimental results are in agreement with the theoretical predictions of Zhang et al. [12], who does not recommend the use of the IHX in subcritical cycles from the point of view of energy efficiency. But as pointed out theoretically by Llopis et al. [13], the IHX also reduces the heat to be released in the condenser of the plant, and therefore, when the subcritical cycle is used as a part of a cascade system its effect must be analysed considering the whole cascade system. With a simplified approach, they pointed out that the reduction of the heat to be transferred in the cascade heat exchanger could provide increases of the overall COP of a cascade system, with a maximum increase of 3 %.

Accordingly, this paper is devoted to present and analyse the global effects of the use of an IHX in the subcritical cycle in a whole HFC134a/CO_2 cascade system. Using the previous experimental data presented by Llopis et al. [13], this paper focuses on the modifications induced in the cascade heat exchanger and in the high temperature cycle (HT) of the cascade system. First, we summarize the effects of the IHX in the CO_2 subcritical cycle. Then, we detail the modifications induced in the cascade heat exchanger and in the HT cycle. Next, we present the experimental COP of the overall cascade system in all tested conditions. Finally, we compare the operation of the plant with and without the IHX in the best performing energy conditions of the plant.

2. Experimental plant and measurement system

The experimental evaluation has been carried out with a HFC134a/CO_2 cascade refrigeration plant, presented previously by Sanz-Kock et al. [14], whose main scheme is detailed in Figure 1.

The subcritical cycle uses a semihermetic compressor with a displacement of 3.48 $m^3 \cdot h^{-1}$ at 1450 rpm and a nominal power of 1.5 kW (at evaporation of -35 ºC and condensation of -10 ºC) and incorporates a finned-tube gas-cooler with 0.6 m^2 heat transfer area in the refrigerant side driven with a fan with nominal electricity consumption of 75 W. Two brazed plate heat exchangers with a total heat transfer area of 3.52 m^2 are used as cascade heat exchanger, a 0.096 m^2 brazed plate heat exchanger as IHX, and a 2.39 m^2 brazed plate heat exchanger as evaporator. The expansion valve is electronic working as thermostatic control. A secondary fluid loop using an 84% by vol. tyfoxit-water mixture provides the cooling load to the plant. More details about the secondary loop can be found in Llopis et al. [15].

The high temperature cycle, working with HFC134a, is driven by a semihermetic compressor with displacement of 32.66 $m^3 \cdot h^{-1}$ at 1450 rpm and nominal power of 3.7 kW (at evaporation of -35 ºC and condensation of 30 ºC). It is a variable speed compressor operated with an inverter drive. A brazed plate heat exchanger of 2.39 m^2 is used as condenser and two electronic expansion valves working as thermostatic control the evaporating process in the cascade heat exchanger. The heat dissipation in condenser is done with a loop working with water [16].

The experimental plant is fully instrumented. The location of sensors is described in Figure 1. It incorporates 29 T-type thermocouples with an uncertainty of ±0.5 ºC. 7 pressure gauges are used in the subcritical cycle (3 of 0-60 bar ± 0.18 bar, 4 of 0-100 bar ± 0.3 bar) and 4 in the HFC134a cycle (2 of 0-10 bar ± 0.03 bar, 2 of 0-25 bar ± 0.075 bar). Two Coriolis mass flow meters (± 0.15 % of lecture) measure refrigerant mass flow rates and two magnetic flow meters (± 0.33 % of lecture) the secondary fluids volumetric flow. Two digital wattmeters (± 0.5 % lecture) measure the compressors power consumption. Data validation is made by comparing the heat transfer rates of the different fluids in the LT evaporator, in the cascade heat exchanger and in the condenser. Sanz-Kock et al. [14] details the data validation.

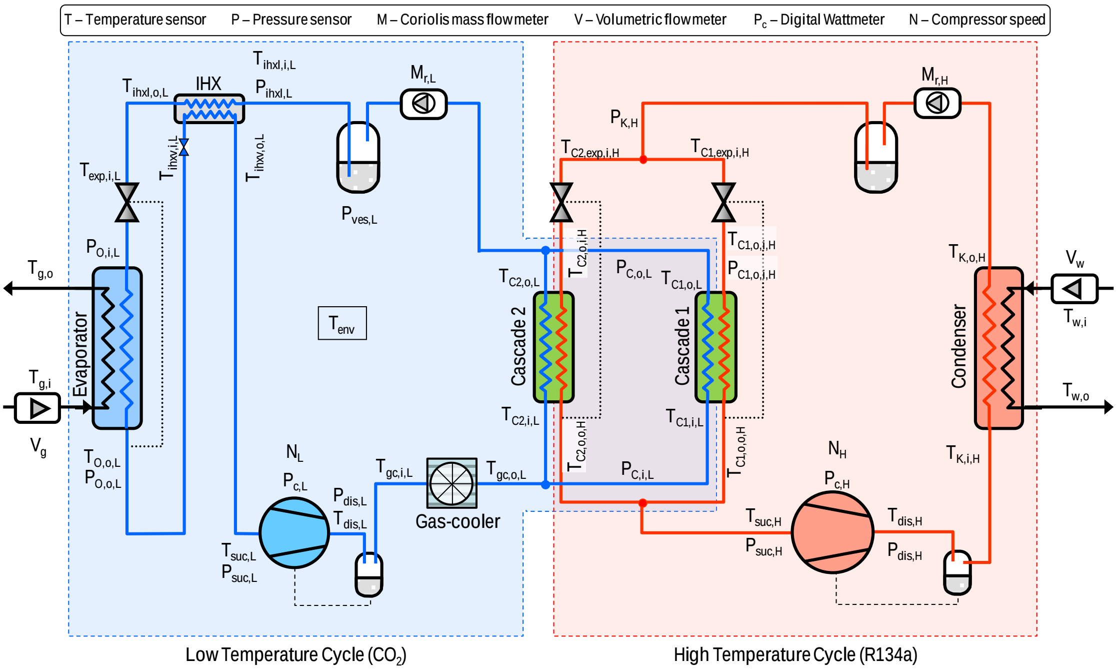

Figure 1. Experimental diagram of the test plant and measurements in each point

3. Experimental procedure

The test campaign used to evaluate the performance of the HFC134a/CO_2 cascade refrigeration system working with and without IHX covered evaporating temperatures of the LT cycle from -40 to -30 ºC and HT condensing temperatures from 30 to 50 ºC. Table 1 details the main reference parameters during the evaluation of the plant. For all the tests, the CO_2 compressor operated at its nominal speed of 1450 rpm and, for each test condition, the intermediate temperature level varied with HT compressor speed regulation. At least five steady-states (at different HT compressor speeds) were measured for each test condition, covering thus a wide range of intermediate conditions (LT condensing temperature / HT evaporating temperature). Each steady-state lasted at least 15 minutes, were the maximum allowed deviation in the evaporating and condensing temperatures was of ± 0.3 ºC. In the tests the gas-cooler of the LT cycle was always kept on.

4. Discussion of experimental results

The internal heat exchanger directly modifies the operating conditions of the low-temperature cycle, as analysed by Llopis et al. [13] for the same plant, however, it also indirectly influences the high-temperature cycle and thus the global operation of the cascade system. This affirmation can be observed in the temperature-entropy diagram of the cycles presented in Figure 2, where the measured cycles of the cascade system are presented for a LT evaporating temperature of -40 ºC and a HT condensing temperature of 50 ºC with and without the IHX for a speed of the HT compressor of 1007.2 rpm. In Figure 2, it can be observed that the use of the IHX introduces also modifications in the intermediate temperature level. Accordingly, first, we discuss the main modifications in the low-temperature cycle, then in the cascade heat exchanger and in the high-temperature cycle and finally we present the overall performance of the cascade system.

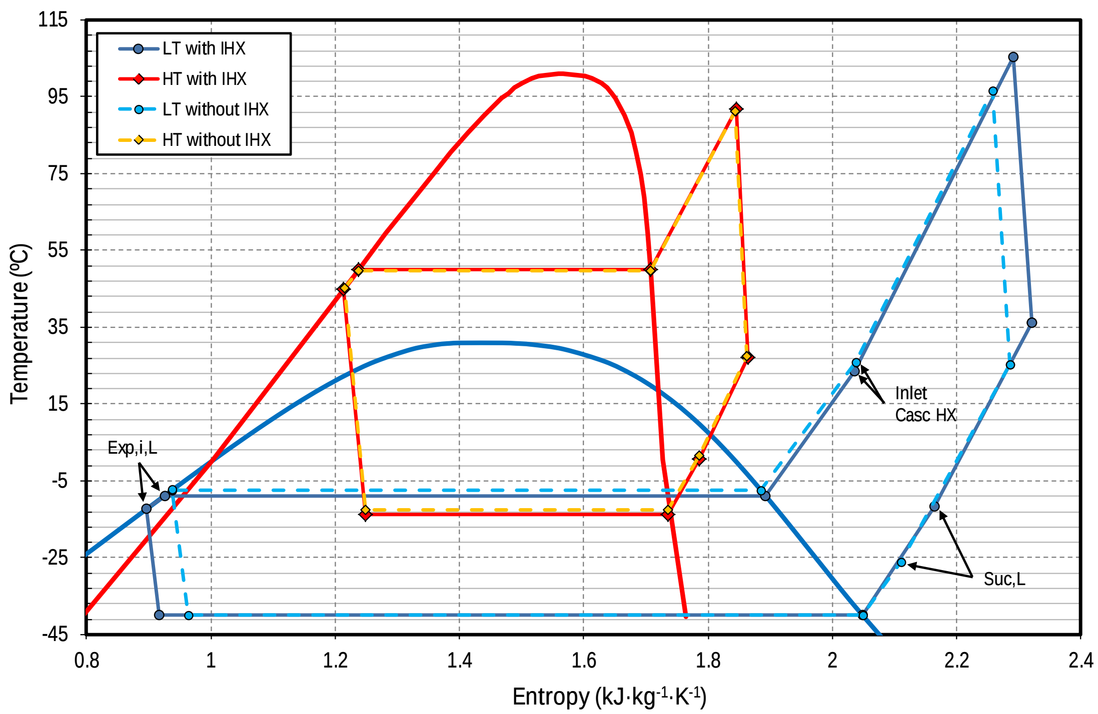

Figure 2. Ts diagrams of the cycles with and without IHX. ($T_{O,L}$=-40.0 ºC, $T_{K,H}$=50.0 ºC, N_H=1007.2 rpm)

4.1. Modifications in the low temperature cycle

The use of the IHX in the subcritical cycle of a cascade system brings about increments on the specific suction volume of the LT compressor, thus refrigerant mass flow rate in the LT cycle is reduced, Eq. (1); however the specific cooling capacity increases thanks to the subcooling achieved in the IHX. For CO_2 in subcritical operation, it was verified that the reduction of the mass flow was more important than the increase of the specific cooling capacity, resulting in reductions of capacity of the whole system, Eq. (2) [14]. Figure 3 compares the refrigerant mass flow rate of the LT cycle in different LT evaporating levels in a wide range of LT condensing temperatures. The average reductions of the LT refrigerant mass flow rate range from 4.7 to 6.2 %. Figure 4 presents the experimental capacity measured in all the tests, according to Eq. (2), where the inlet enthalpy to the evaporator has been evaluated considering the lamination process isenthalpic. Data regarding the three HT condensing levels is represented in Figure 4. The average reductions in capacity for all the test range, when using the IHX, ranged from 1.1 % at -40 ºC to 2.4 % at -30 ºC, they being lower than the reductions suffered by the mass flow rate. Obviously, from the point of view of the capacity, the use of the IHX will slightly reduce the capacity of the plant.

$$\dot{m}_{r,L} = \frac{\eta_{v,L}}{v_{suc,L}} \cdot \dot{V}_{G,L} \qquad (1)$$

$$\dot{Q}_{O,L} = \dot{m}_{r,L} \cdot \left(h_{O,o} - h_{O,i}\right) = \dot{m}_{r,L} \cdot q_{O,L} \qquad (2)$$

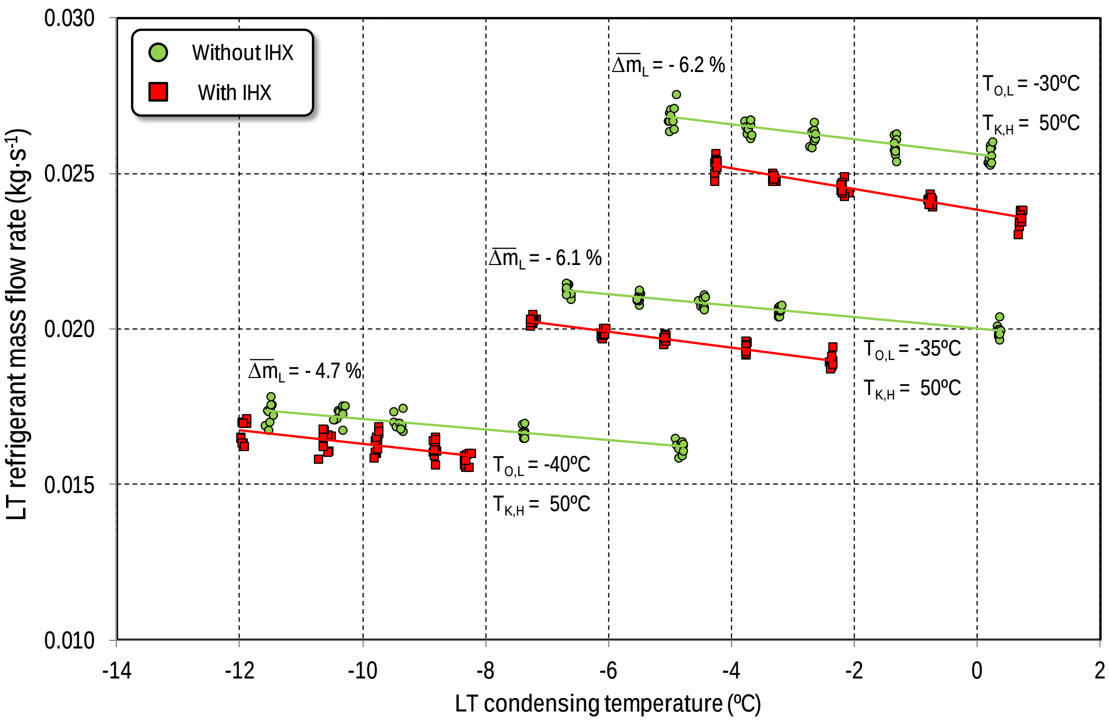

Figure 3. Low temperature refrigerant mass flow rate with and without IHX ($T_{K,H}$ = 50 ºC)

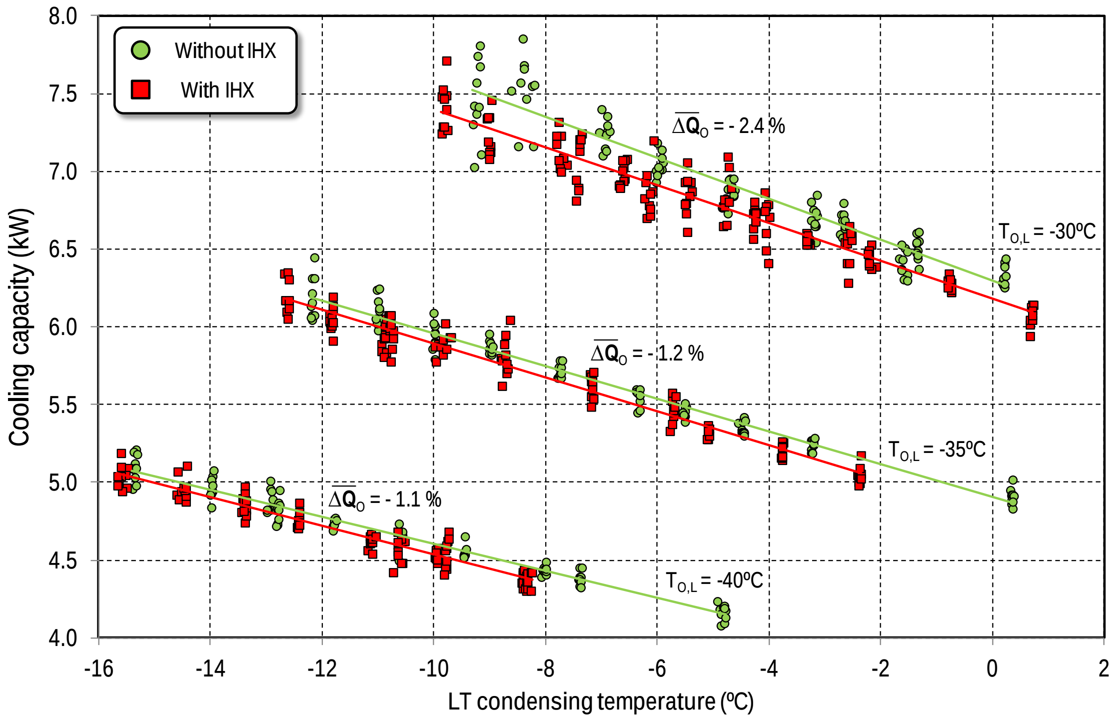

Figure 4. Cooling capacity with and without IHX

However, the reduction of the LT refrigerant mass flow rate has also influence in the cascade heat exchanger and thus in the HT refrigeration cycle. The use of the IHX reduces the mass flow but increases the LT compressor discharge temperature. Nonetheless, in this plant, the increment of the discharge temperature is avoided since it incorporates a gas-cooler at the exit of the compressor (Figure 1). The inlet temperature to the cascade heat exchanger depends on the thermal effectiveness of the gas-cooler, Eq. (3). For this plant in all the analysed range, its thermal effectiveness varied from 96.4% to 99.9%, and it was independent of the use of the (Table 1). Accordingly, the inlet temperature of the vapour to the cascade heat exchanger was also independent of the use of the IHX, being the inlet temperature of the CO_2 to the cascade heat exchanger similar to the environment temperature, as can be seen in Figure 2. The result is that when the IHX is used there is an effective reduction of the heat to be transferred in the cascade heat exchanger, caused by the reduction of the refrigerant mass flow in the LT cycle. Accordingly, the HT cycle is subjected to less cooling load, as presented in Figure 5. The average reductions of the heat to be transferred to the HT cycle ranged from 4.4 % to 5.2%. They slightly differ from the reductions of mass flow because the intermediate level (LT condensing or HT evaporating temperatures) is modified (Figure 2), as detailed in next subsection.

$$\varepsilon_{gc} = \frac{T_{gc,i} - T_{gc,o}}{T_{gc,i} - T_{env}} \tag{3}$$

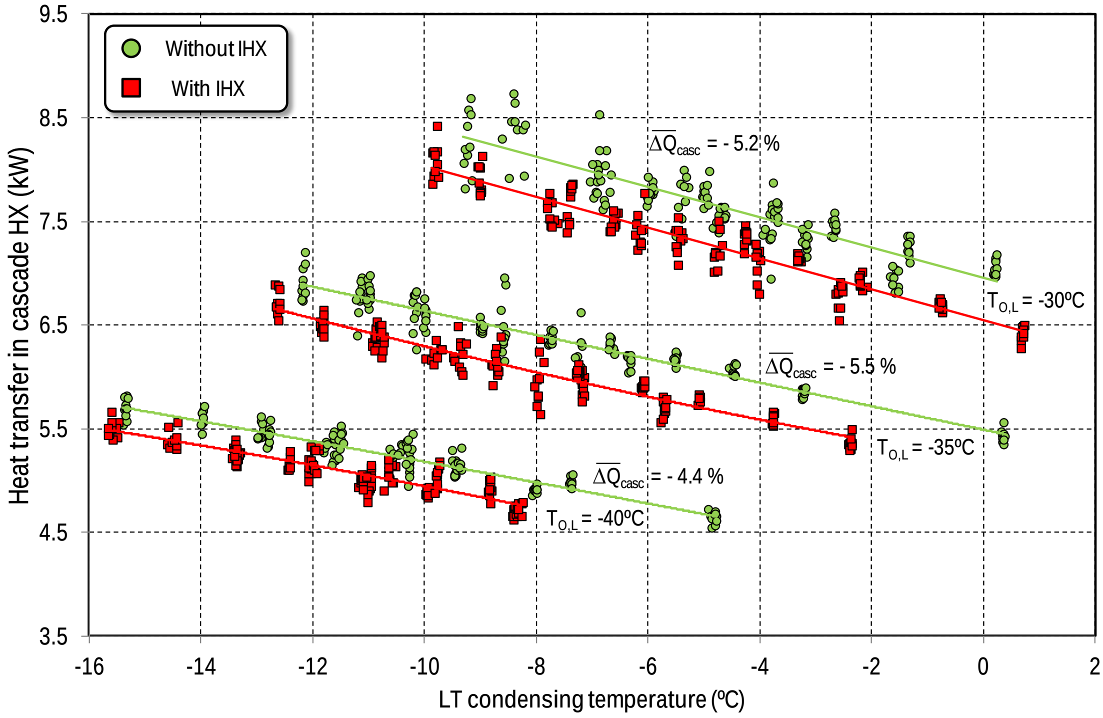

Figure 5. Heat transfer in the cascade heat exchanger with and without IHX

4.2. Modifications in the cascade heat exchanger and the high temperature cycle

The cascade heat exchanger is the thermal coupling between the LT cycle and the HT cycle, it acts as condenser of the LT cycle and evaporator of the HT cycle. Heat transfer in the cascade HT is evaluated using Eq. (4) for LT cycle condensation and with Eq. (5) for HT cycle evaporation, where the enthalpy values of the refrigerants at the inlet and exit of both cascade heat exchangers are averaged values. The average discrepancy between both heat transfer rates is of 2.7 % working without IHX and 3.3 % working with IHX, as presented in [13].

$$\dot{Q}_{K,L} = \dot{m}_{r,L} \cdot \left(\frac{\left(h_{C1,i,L} - h_{C1,o,L}\right) + \left(h_{C2,i,L} - h_{C2,o,L}\right)}{2} \right) = \dot{m}_{r,L} \cdot q_{K,L} \tag{4}$$

$$\dot{Q}_{O,H} = \dot{m}_{r,H} \cdot \left(\frac{\left(h_{C1,O,i,H} - h_{C1,O,o,H}\right) + \left(h_{C2,O,i,H} - h_{C2,O,o,H}\right)}{2} \right) = \dot{m}_{r,H} \cdot q_{O,H} \tag{5}$$

Neglecting heat losses to the environment in the cascade heat exchangers, the heat balance in the cascade heat exchanger relates the refrigerant mass flow rate of the HT cycle needed to absorb the heat rejection of the LT cycle in the cascade heat exchanger, Eq. (6). Since the heat rejection of the LT cycle decreases with the use of the IHX (Figure 5), the refrigerant mass flow rate of the HT is also reduced. This effect can be observed in Figure 6 for the operation of the plant at three evaporating levels (-30, -35 and -40 ºC) and a HT condensing temperature of 50 ºC. For these cases, the reduction of the HT mass flow rate ranges from 3.4 to 4.9 %.

$$\dot{m}_{r,H} = \frac{\dot{Q}_{K,L}}{q_{O,H}} \tag{6}$$

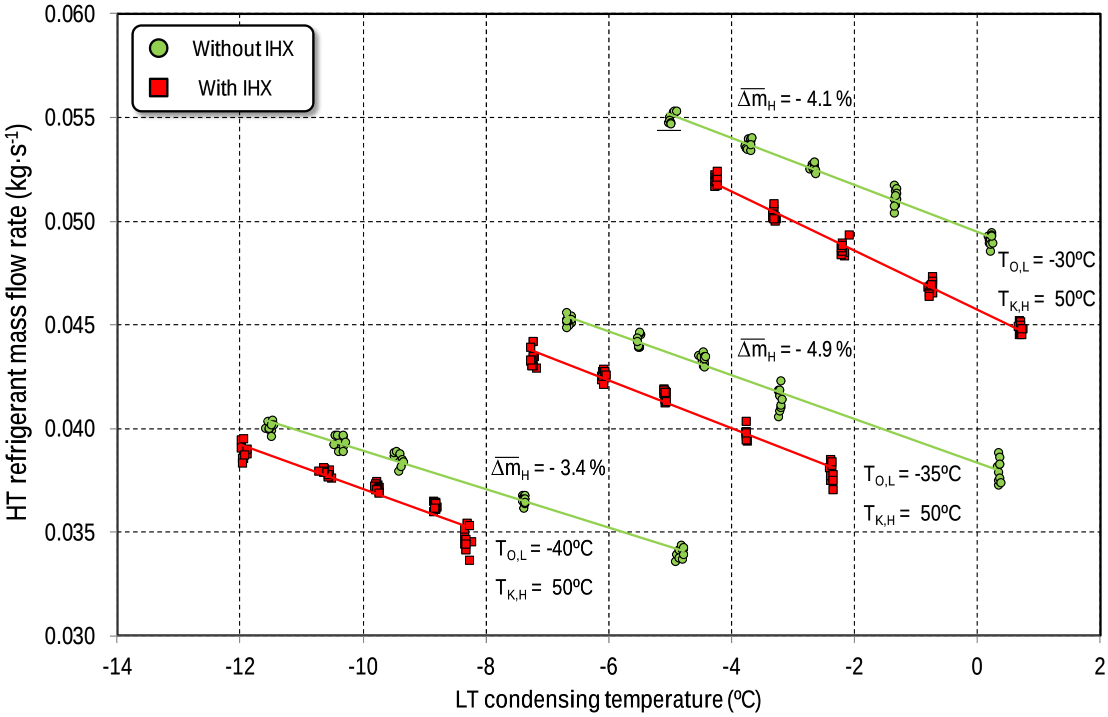

Figure 6. HT refrigerant mass flow rate with and without IHX ($T_{K,H}$ = 50.0 ºC)

As it can be seen in Figure 2, the use of the IHX scrolls down the intermediate level of the cascade system, reducing the HT evaporating temperature and the LT condensing level. It has been verified that the reduction of the intermediate level is independent of the use of the IHX, as observed in Figure 7, where the HT evaporating temperature is contrasted with the LT condensing level. For each evaporating level the phase change temperatures in the cascade heat exchanger are independent of the use of the IHX. The temperature difference of the data to the continuous line ($T_{O,H}$ = $T_{K,L}$) corresponds to the temperature difference between condensation and evaporation in the cascade heat exchanger, Eq. (7).

$$\Delta T_{casc} = T_{K,L} - T_{O,H} \tag{7}$$

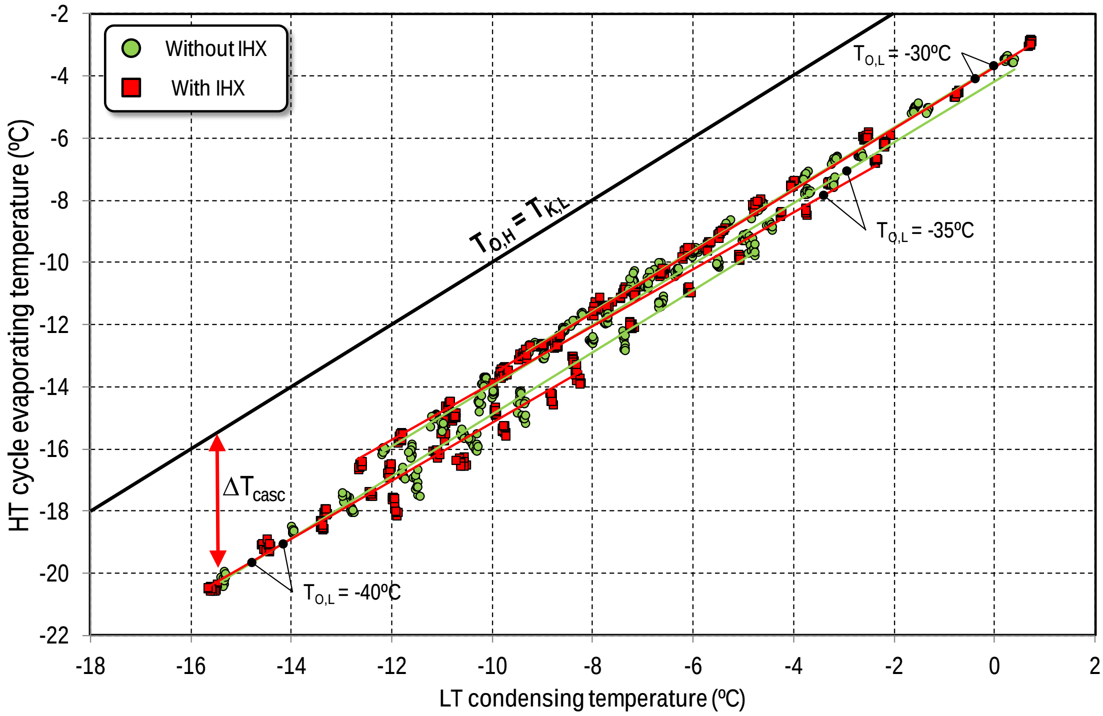

Figure 7. Temperature difference in cascade heat exchanger with and without IHX

Obviously, if there is a reduction of the heat to be transferred in the cascade heat exchanger (Figure 5), the heat transfer surface is the same and the temperature difference in the cascade heat exchanger remains constant (Figure 7), there will be a modification of the heat transfer characteristics in the cascade heat exchanger. That can be approximately analysed by evaluating the overall heat transfer coefficient in the cascade heat exchanger. For this comparison, that is presented in Figure 8 for three evaporating levels, it has been evaluated as done by Dopazo et al. [17] by referring it to the temperature difference in the cascade heat exchanger, Eq. (7).

$$\overline{U} = \frac{\dot{Q}_{O,H}}{S_{casc} \cdot \Delta T_{casc}} \tag{8}$$

As it can be observed in Figure 8, the use of the IHX, causes reductions of the overall heat transfer coefficient in the cascade heat exchanger, being the measured reductions ranging from 7.0 to 8.6 %. Although it has not been extensively analysed, this reduction could be caused by the reductions of both refrigerant mass flow rates (m_L, m_H) when the IHX is used. Accordingly, although the heat transfer in the cascade heat exchanger is reduced due to the use of the IHX, this reduction is not translated directly to the improvement of the cycle, since there is a degradation of the heat transfer characteristics in the cascade heat exchanger that does not allow lowering the temperature difference in the cascade heat exchanger.

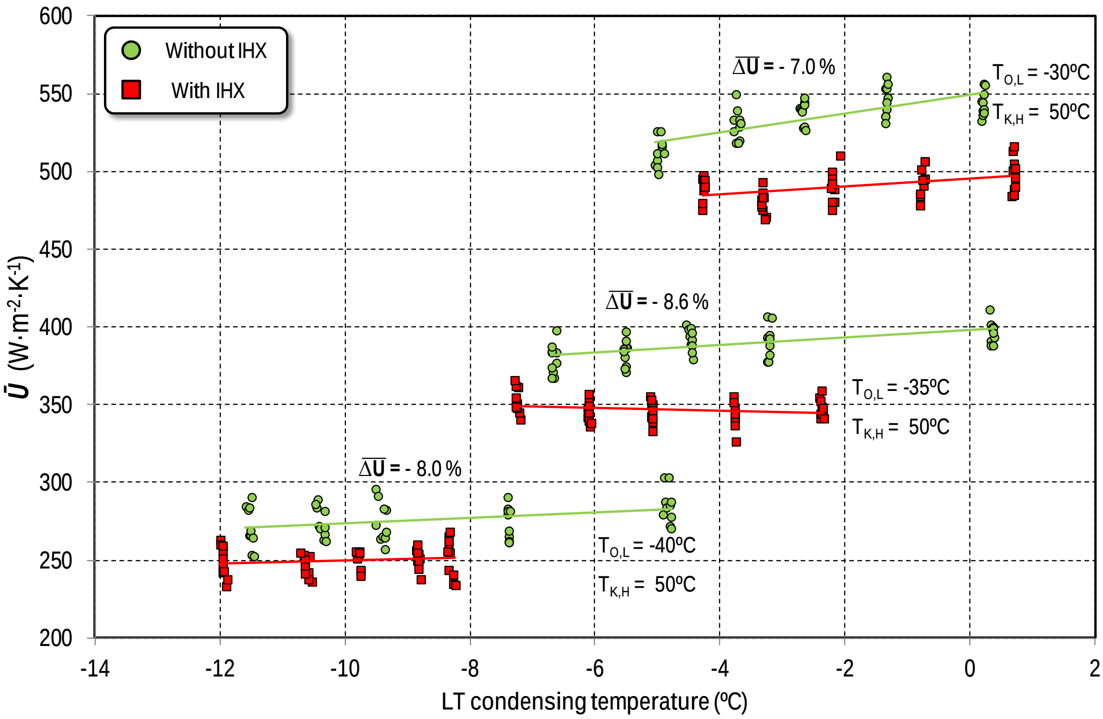

Figure 8. Overall heat transfer coefficient of cascade heat exchanger with and without IHX ($T_{K,H}$ = 50.0 ºC)

4.4. Modifications of the cascade cycle

All the modifications introduced by the use of the IHX can be summarized attending to the coefficient of performance of the cascade system, Eq. (9). It is the quotient of the cooling capacity provided by the LT cycle and the sum of the electrical power consumptions of the LT and HT compressors and the electrical fan of the gas-cooler (75 W).

$$COP = \frac{\dot{Q}_{O,L}}{P_{c,L} + P_{c,H} + P_{c,gc}} \tag{9}$$

As detailed previously, there is a slight decrement of the capacity of the cycle due to the use of the IHX (Figure 4), mainly driven by the reduction of refrigerant mass flow rate in the low temperature cycle. This reduction of the mass flow rate has also effect on the power consumption of the LT compressor, as presented in Figure 9. For the operation at a HT condensing temperature of 50 ºC the reduction of this consumption is inside the range 0.92 to 1.60 %. The reduction of mass flow rate is not translated directly to the power consumption, since the increase of the CO_2 specific suction volume increases the isentropic specific compression work. The most important reduction is observed at the HT compressor, which data for the operation at a HT condensing temperature are presented in Figure 10. In this case, since the heat to be transferred in the cascade condenser is reduced, it has a direct impact in the HT refrigerant mass flow rate (Figure 6) and thus in the power consumption of the compressor. In this case, the average measured reductions are between 3.05 to 4.97 %.

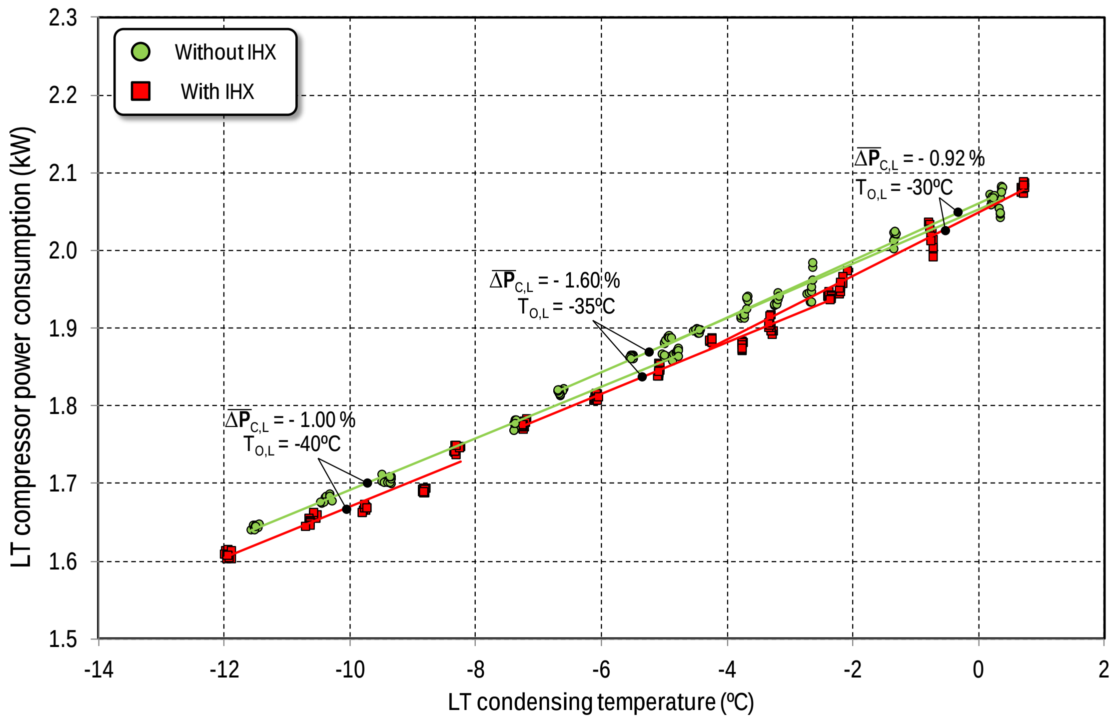

Figure 9. Low temperature compressor power consumption with and without IHX ($T_{K,H}$ = 50.0 ºC)

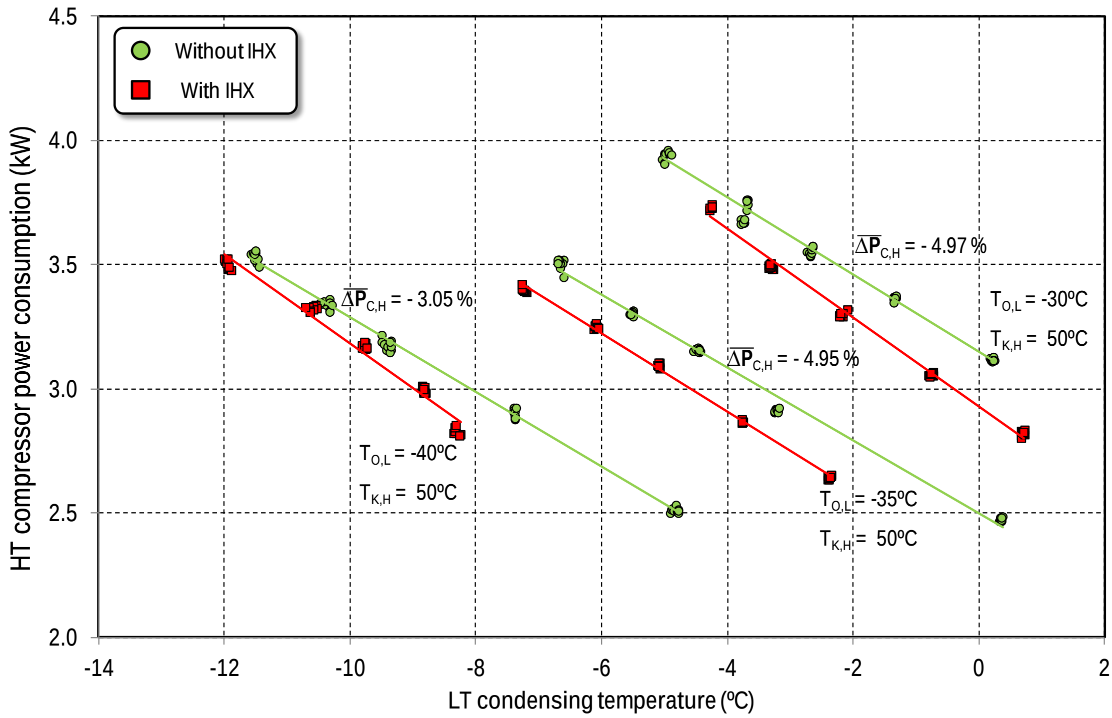

Figure 10. High temperature compressor power consumption with and without IHX ($T_{K,H}$ = 50.0 ºC)

Finally, the calculated COP from measured data with and without IHX are presented in Figure 11 for the operation at three LT evaporating levels (-40, -35 and -30 ºC) and three HT condensing levels (30, 40 and 50 ºC) for a wide range of intermediate temperatures. The variation of the intermediate level is achieved with variation of the speed of the HT compressor. As it can be observed, for all tested conditions, the introduction of the IHX in the LT temperature cycle brings improvements on the overall energy performance of the cascade system. The conclusion that can be extracted is that the reduction of the capacity due its use (Figure 4) is lower than the benefit obtained due to the reduction of the heat to be transferred in the cascade heat exchanger (Figure 5). Accordingly, its use is also beneficial from the point of view of energy efficiency.

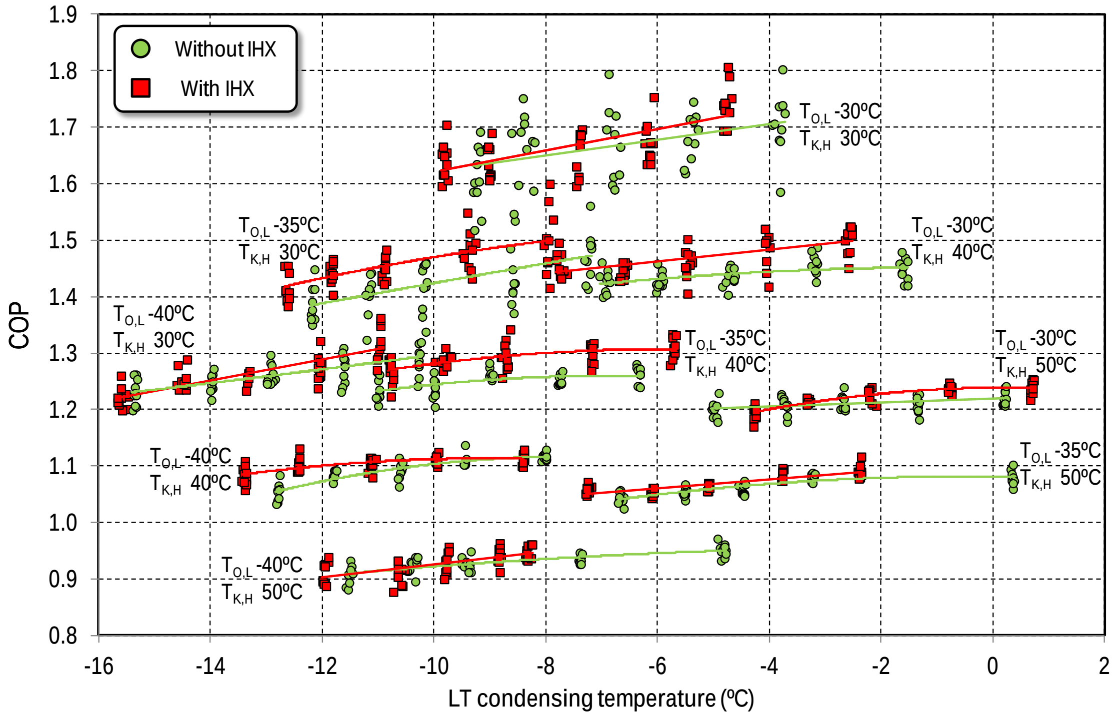

Figure 11. COP with and without IHX

4.5. Comparison at the best conditions

Finally, the main energy parameters of the cascade system using or not using the IHX are summarized: cooling capacity in Figure 12 and COP Figure 13. The data here considered corresponds to the condition at the maximum measured COP (Table 1). It needs to be mentioned that some of the values are not at the optimum (maximum COP) condition, since we could not reach them because of the security restrictions of the plant. Regarding COP, the maximum measured increment has been of 3.7 % at $T_{O,L}$ = -35 ºC and $T_{K,H}$ = 40 ºC, but for the operation at $T_{O,L}$ = -40 ºC and $T_{K,H}$ = 50 ºC a decrement of 0.5% has been registered. Nonetheless, in general the IHX does not reduce the overall performance of the cascade system. For cooling capacity, the maximum measured reduction has been of 4.1% at $T_{O,L}$ = -30 ºC and $T_{K,H}$ = 50 ºC.

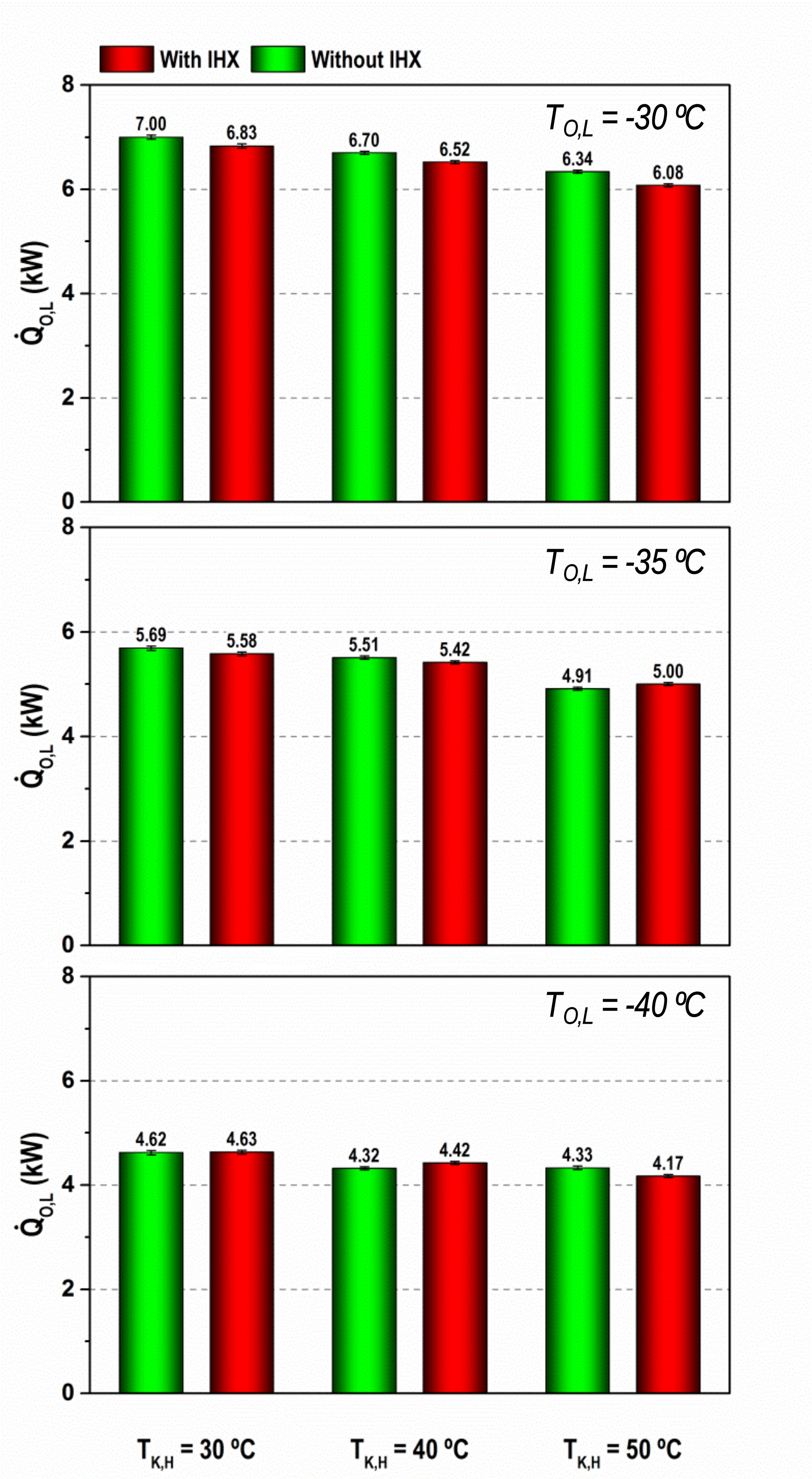

Figure 12. Cooling capacity with and without IHX at the maximum COP condition

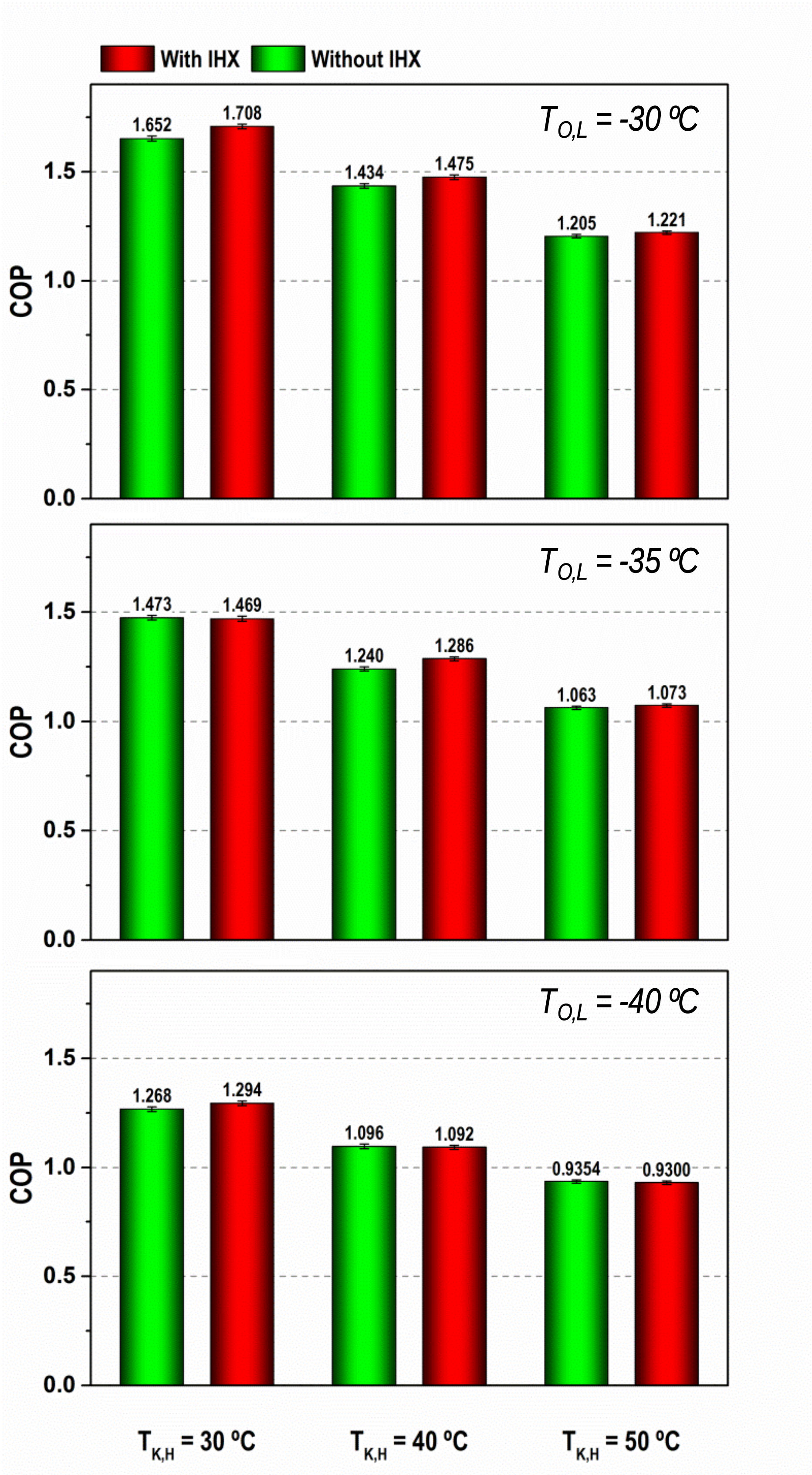

Figure 13. COP with and without IHX at the maximum COP condition

5. Conclusions

This work presents the experimental evaluation of the influence of the use of an internal heat exchanger in a CO_2 subcritical cycle in a whole HFC134a/CO_2 cascade refrigeration system.

Previous theoretical studies predicted degradation of the main energy parameters for CO_2 in subcritical operation and recommended not to use this element. This study verified this affirmation inside a wide range of operating conditions, measuring reductions of capacity between 2.4 to 1.1 %. However, the previous studies do not consider the operation of the subcritical cycle inside a cascade system.

This work extends the analysis to a whole HFC134a/CO_2 cascade system. It has been measured that the use of the IHX reduces the heat to be transferred in the cascade heat exchanger (in average between 4.4 to 5.2 %), reducing thus the thermal load of the high temperature cycle. This heat load reduction allows reducing the high temperature refrigerant mass flow rate between 3.4 to 4.9 %. In addition, it has been observed that the temperature difference in the cascade heat exchanger remains independent of the use of the internal heat exchanger, but there is a degradation of the heat transfer characteristics in the cascade heat exchanger due to the reduction of the refrigerant flows.

For all the operation range, an improvement of the COP of the whole cascade system has been measured for all tested conditions inside a wide range of intermediate temperature levels. For the best performing intermediate level, the use of the internal heat exchanger in the CO_2 cycle has offered a maximum increase of COP of 3.7 % at -35ºC and 40 ºC of evaporating and condensing temperatures. However, at -40 and 50ºC a decrement of 0.5% in COP has been measured.

According to the evaluation, we can affirm that the internal heat exchanger in the CO_2 subcritical cycle in a whole cascade system is recommended if it incorporates a gas-cooler at the exit of the low temperature compressor. Since with a small reduction in capacity and a slight increment in COP, the plant benefits for the increment of the specific suction volume of the low temperature compressor, increasing thus the expected useful life of the plant.

6. Acknowledgements

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$T_{O,L}$	$T_{K,H}$	$T_{K,L}$	$T_{O,H}$	ΔT_{casc}	N_H	ε_{IHX}	ε_{gc}	Q_{IHX}	$Q_{K,L}$	$Q_{O,L}$	$P_{C,L}$	$P_{C,H}$	COP
(ºC)	(ºC)	(ºC)	(ºC)	(ºC)	(rpm)	(%)	(%)	(W)	(W)	(W)	(W)	(W)	(-)
Operation with IHX													
-30.0	30.0	-9.81 to -4.76	-13.47 to -8.12	3.26 to 3.82	806 to 1209	84.4 to 94.0	97.2 to 99.8	146 to 265	7016 to 8419	6650 to 7710	1680 to 1890	2030 to 2860	1.61 to 1.71
-30.0	40.1	-7.73 to -2.57	-11.41 to -5.95	3.28 to 3.77	806 to 1209	83.1 to 86.5	96.1 to 98.9	189 to 287	6541 to 7779	6518 to 7125	1722 to 1956	2393 to 3171	1.42 to 1.47
-30.0	50.0	-4.26 to 0.70	-8.40 to -2.91	3.52 to 4.23	806 to 1209	75.7 to 78.5	96.4 to 98.5	216 to 305	6281 to 7459	5943 to 6801	1883 to 2089	2806 to 3744	1.18 to 1.22
-35.0	30.1	-12.63 to -7.96	-16.51 to -11.43	3.21 to 3.97	806 to 1209	89.3 to 98.8	98.1 to 99.5	149 to 239	5640 to 6889	5289 to 6256	1593 to 1763	1954 to 2724	1.39 to 1.47
-34.9	40.0	-10.77 to -5.72	-14.92 to -9.42	3.61 to 4.28	806 to 1209	86.9 to 91.0	97.6 to 99.9	184 to 260	5564 to 6498	5285 to 6011	1654 to 1826	2300 to 3019	1.25 to 1.29
-35.0	50.0	-7.24 to -2.38	-10.01 to -6.70	4.26 to 4.89	806 to 1209	82.6 to 83.7	97.3 to 99.2	204 to 272	5294 to 6118	4937 to 5546	1771 to 1947	2637 to 3425	1.04 to 1.07
-39.9	30.0	-15.59 to -10.97	-20.47 to -15.43	4.10 to 5.04	806 to 1209	92.9 to 99.8	99.2 to 99.9	144 to 201	4788 to 5665	4437 to 5091	1503 to 1652	1842 to 2567	1.20 to 1.29
-40.0	40.0	-13.39 to -8.10	-18.45 to -13.11	4.58 to 5.22	806 to 1209	89.4 to 90.7	98.8 to 99.7	150 to 204	4623 to 4903	4254 to 4903	1562 to 1727	2151 to 2860	1.06 to 1.09
-40.0	50.0	-11.95 to -8.32	-17.79 to -13.61	4.93 to 6.25	1007 to 1310	81.4 to 83.9	97.3 to 98.6	138 to 182	4569 to 5335	4266 tp 4772	1604 to 1749	2815 to 3527	0.89 to 0.93
Operation without IHX													
-30.0	30.2	-9.23 to 3.82	-12.71 to -7.25	3.28 to 3.71	806 to 1209	-	94.7 to 95.6	-	6944 to 8733	6357 to 7818	1706 to 1953	2043 to 2912	1.58 to 1.67
-30.0	40.1	-6.93 to -1.59	-10.68 to -5.00	3.31 to 3.90	806 to 1209	-	97.2 to 99.3	-	6816 to 8054	6302 to 7396	1799 to 1995	2430 to 3286	1.41 to 1.43
-30.0	50.1	-4.98 to 0.21	-9.17 to -3.43	3.57 to 4.28	907 to 1310	-	95.5 to 98.9	-	6971 to 7987	6255 to 7175	1862 to 2073	3112 to 3962	1.18 to 1.20
-34.9	30.0	-12.17 to -7.23	-16.02 to -10.57	3.08 to 4.03	806 to 1209	-	97.3 to 99.9	-	5996 to 7202	5471 to 6410	1631 to 1804	1981 to 2746	1.38 to 1.47
-35.0	40.1	-11.00 to -6.34	-15.29 to -10.28	3.86 to 4.42	907 to 1310	-	98.6 to 99.9	-	6043 to 6978	5427 to 6218	1661 to 1843	2525 to 3271	1.22 to 1.24
-35.0	50.0	-6.66 to 0.35	-11.29 to -3.50	3.75 to 4.75	705 to 1209	-	97.3 to 99.4	-	5349 to 6391	4832 to 5622	1814 to 2084	2469 to 3521	1.03 to 1.06
-40.0	30.1	-15.35 to -10.27	-20.15 to -14.43	4.00 to 5.02	806 to 1209	-	99.1 to 99.9	-	4950 to 5823	4432 to 5173	1540 to 1695	1877 to 2589	1.21 to 1.27
-40.0	40.1	-12.79 to -8.02	-17.96 to -12.49	4.38 to 5.26	806 to 1209	-	99.1 to 99.9	-	4865 to 5589	4374 to 4917	1620 to 1764	2185 to 2920	1.04 to 1.10
-40.0	50.0	-11.52 to -4.83	-17.09 to -9.52	4.37 to 6.06	806 to 1410	-	97.6 to 99.7	-	4544 to 5534	4081 to 4839	1641 to 1875	2503 to 3558	0.89 to 0.94

Table 1. Reference parameters of the experimental evaluation of the cascade with and without IHX