# Experimental evaluation of a $CO_2$ transcritical refrigeration plant with dedicated mechanical subcooling

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## ABSTRACT

$CO_2$ transcritical refrigeration cycles require optimization to reach the performance of conventional solutions at high ambient temperatures. Theoretical studies demonstrated that the combination of a transcritical cycle with a mechanical subcooling cycle improves its performance; however, any experimentation with $CO_2$ has been found. This work presents the energy improvements of the use of a mechanical subcooling cycle in combination with a $CO_2$ transcritical refrigeration plant, experimentally. It is tested the combination of a R1234yf single-stage refrigeration cycle with a semihermetic compressor for the mechanical subcooling cycle, with a single-stage $CO_2$ transcritical refrigeration plant with a semihermetic compressor. The combination is evaluated at two evaporating levels of the $CO_2$ cycle (0 and -10 ºC) and three heat rejection temperatures (24, 30 and 40 ºC). The optimum operating conditions and capacity and COP improvements are analysed with maximum increments on capacity of 55.7 % and 30.3 % on COP.

## KEYWORDS

$CO_2$; transcritical; subcooling; energy efficiency

NOMENCLATURE

| | |
|---|---|
| $COP$ | coefficient of performance |
| $MS$ | dedicated mechanical subcooling cycle |
| $t$ | compression ratio |
| $T$ | temperature, ºC |
| $TRANS$ | referring to the $CO_2$ transcritical cycle |
| $\dot{m}$ | mass flow rate, $kg \cdot s^{-1}$ |
| P | pressure, bar |
| $P_C$ | power consumption, kW |
| $\dot{Q}$ | heat transfer rate, kW |
| $q_o$ | specific cooling capacity, $kJ \cdot kg^{-1}$ |
| $h$ | specific enthalpy, $kJ \cdot kg^{-1}$ |
| $x_v$ | vapour title |

GREEK SYMBOLS

| | |
|---|---|
| $\eta_i$ | Isentropic efficiency |
| $\eta_v$ | Volumetric efficiency |
| $\Delta$ | Increment |

SUBSCRIPTS

| | |
|---|---|
| $CO_2$ | Referring to $CO_2$ cycle |
| $env$ | environment |
| gc | gas-cooler |
| in | inlet |
| $K$ | Condenser of MS cycle |
| $MS$ | referring to the dedicated mechanical subcooling cycle |
| $out$ | outlet |
| $O$ | evaporator |
| $r$ | refrigerant |
| $w$ | water |
| $sf$ | secondary fluid of the evaporator |
| $sub$ | referring to the subcooler |

# 1. Introduction

After the approval of the bans to the use of fluorinated fluids in the Refrigeration sector by the F-Gas in Europe (European Commission, 2014) the interest for $CO_2$ as refrigerant has been taken a step forward, especially in commercial refrigeration where its use is not questioned any more. $CO_2$ in centralized commercial refrigeration systems is mainly put into practice with cascades or pure transcritical systems. According to Shecco Guide (2014), there were 2885 supermarkets operating with pure transcritical systems and 1638 with HFC/$CO_2$ cascades in Europe in 2013. However, when referring to warm countries, such as Spain or Italy, only 21 supermarkets operate with transcritical systems and 231 with cascades. The preferred solution for warm countries is the cascade, since when the ambient temperature is high the performance of pure transcritical systems is far away from that offered by the cascades (Llopis et al., 2015b).

Different researchers have worked to improve the efficiency of $CO_2$ transcritical systems trying to reach the performance of other systems. Aprea & Maiorino (2008) and Sánchez et al. (2014a) studied the improvements by using the internal heat exchangers (IHX) in single-stage plants and Cavallini et al. (2007) in two-stage systems. The experimental measurements demonstrated that the IHX can improve the COP up to a maximum of 10 %. Others analyzed the effect of extracting vapour from the intermediate vessel to be injected in different points of the cycle, measuring maximum increments of 7 % in single-stage plants (Cabello et al., 2012) and 16.5 % in double-stage cycles (Cho et al., 2009). The use of expanders (Li et al., 2004), ejectors (Lee et al., 2014) and regulation strategies (Peñarrocha et al., 2014) are also considered. And now the improvements of the $CO_2$ transcritical plants are looked for its combination with other systems, such as desiccant wheels (Aprea et al., 2015) or absorption plants (Arora et al., 2011).

In 1994, Zubair reintroduced the use of dedicated mechanical subcooling systems. Recently, this option has been studied for supermarket refrigeration systems in warm countries by Hafner et al. (2014) and Gullo el. At. (2016). Llopis et al. (2015a) analysed theoretically the use of a dedicated mechanical subcooling system for $CO_2$ transcritical systems, where the possibilities of increasing the energy performance of a transcritical $CO_2$ cooling system by subcooling the $CO_2$ at the exit of the gas-cooler were studied. The $CO_2$ subcooling, with degrees from 2.5ºC to 10 ºC, was done thanks to a dedicated refrigeration system, which rejected heat to the same hot sink than the transcritical. The MS cycle evaporated at a temperature established by the $CO_2$ conditions at the exit of the gas-cooler, but there it was limited to 10ºC. The pressure drops and heat transfer in pipes were neglected. Evaporating temperature was fixed and a useful overheating of 10 ºC was considered. The gas temperature at the exit of the gas-cooler was calculated considering an approach temperature of 5ºC (Kim et al., 2004), as shown in equation (1), and of 10ºC at the condenser of the MS cycle, equation (2).

$$T_{gc,out} = T_{env} + 5 \quad (1)$$

$$T_{K,MS} = T_{env} + 10 \quad (2)$$

That work considered simplified models for the compressors, using the same internal and volumetric efficiency curve:

$$\eta_i = \eta_v = 0.95 - 0.1 \cdot t \quad (3)$$

Llopis et al. (2015a) concluded that the MS cycle improves the overall energy efficiency if $COP_{MS}$ > $COP_{TRANS}$, i.e., when the COP of the MS cycle is higher than the COP of the transcritical cycle working together. The advantages that it introduces are a reduction of the optimum working pressure, an increase of the specific cooling capacity and the $CO_2$ refrigerant mass flow rate, reduction of the $CO_2$ compressor power consumption and important increments of the overall COP and cooling capacity.

In fact, the theoretical results predicted maximum increments in COP of 9.5 %, 13.5 % and 13.1 % and in cooling capacity of 20.7 %, 19.7 % and 12.7 % at evaporations levels of 5 ºC, -5 ºC and -30 ºC, respectively. However, the improvements that the MS cycle can introduce are higher than those mentioned previously, due to the conservative conditions established in the theoretical study and limited subcooling degrees, which are lower than those analysed experimentally.

This work has been developed in order to evaluate experimentally the impact of a dedicated mechanical subcooling system on a $CO_2$ transcritical refrigeration plant. The main objective is to quantify the energy improvements that can be achieved with this cycle modification. The evaluation presented in this paper corresponds to the operation of the plant at two evaporating levels (0 and -10ºC) tested at three different heat rejection temperatures (24, 30.2 and 40 ºC) covering a wide range of gas-cooler operating pressures. The study has been done without IHX because the mechanical subcooling is supposed to introduce the same benefits without the penalisation the IHX causes at the compressor suction conditions, thus, it avoids the increase of compressor suction volume, reduction of mass flow rate, increase of compression work and increase in the discharge temperature. The evaluation of the improvements introduced by the mechanical subcooling has been carried out by comparing to pure transcritical system working at the same conditions. In the analysis the evaporating level at the evaporator has been maintained artificially with an external system, since as detailed below, there is a big increment of the cooling capacity that difficult interpretation of results if it is analysed using external conditions. The main energy parameters, capacity and COP, are analysed and discussed.

## 2. Experimental plant and measurement system

The results analysed in this work are based on the experimental results obtained with a $CO_2$ single-stage transcritical refrigeration system coupled thermally through a subcooler with a single-stage refrigeration cycle, which main components are detailed in Figure 1.

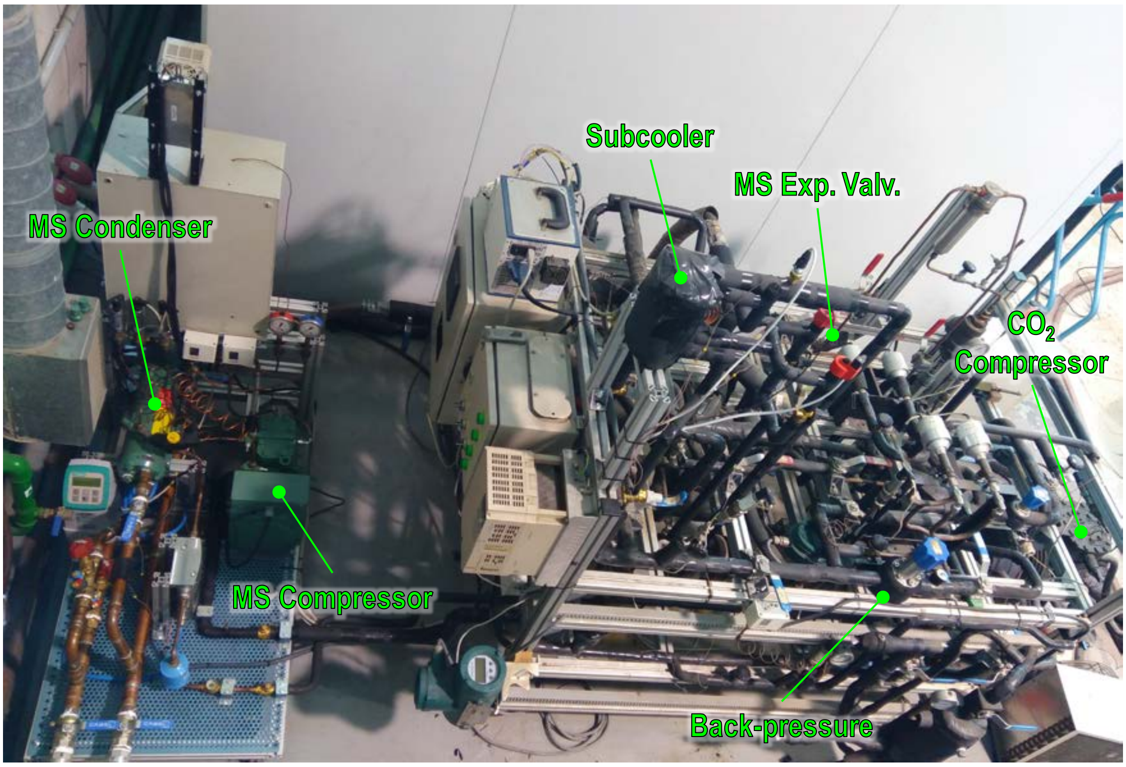


Figure 1. View of the $CO_2$ transcritical plant (right) and MS cycle (left)

The $CO_2$ transcritical refrigeration plant, previously presented by Cabello et al. (2008), uses a semihermetic compressor with a displacement of 3.48 $m^3 \cdot h^{-1}$ at 1450 rpm and a nominal power of 4 kW. The expansion is carried out by a double-stage system, composed of a pressostatic valve (back-pressure) controlling the gas-cooler pressure, a liquid receiver between stages and an electronic expansion valve, working as thermostatic, to control the evaporating process. Evaporator and gas-cooler are concentric counter current heat exchangers with exchange surface of 0.6 $m^2$ and 0.42 $m^2$, respectively.

The subcooler is situated directly downstream of the gas-cooler. It is a brazed plate heat exchanger with an exchange surface of 0.576 $m^2$. It works as evaporator of the mechanical subcooling system and couples both cycles thermally. The mechanical subcooling cycle, working with R1234yf, is driven by a semihermetic compressor with displacement of 4.06 $m^3 \cdot h^{-1}$ at 1450 rpm and nominal power of 0.7 kW, working always at nominal speed. A shell-and-tube heat exchanger is used as condenser and liquid recipient. The expansion valve is electronic working as thermostatic.

Heat dissipation in gas-cooler and condenser of the MS cycle is done with a loop working with water, providing them with water at the same temperature, thus both cycles have the same hot sink. The evaporator is supplied with another loop, working with a tyfoxit-water mixture (84 % by volume) that enables a constant temperature to be maintained in the evaporator.

The thermodynamic properties of the working fluids are obtained thanks to the measurement system presented in Figure 2. It is composed by 21 T-type thermocouples measuring fluids' temperatures at the entrance and at the exit of each element; and 14 pressure gauges, 4 placed in the mechanical subcooling cycle and 10 in the transcritical cycle. $CO_2$ and R1234yf flows are measured by two Coriolis mass flow meters, as well as dissipation water flow of the gas-cooler, which is measured using another Coriolis. The flow of the other secondary fluids is measured by two magnetic volumetric flow meters. Compressors' power consumptions are measured by two digital wattmeters.

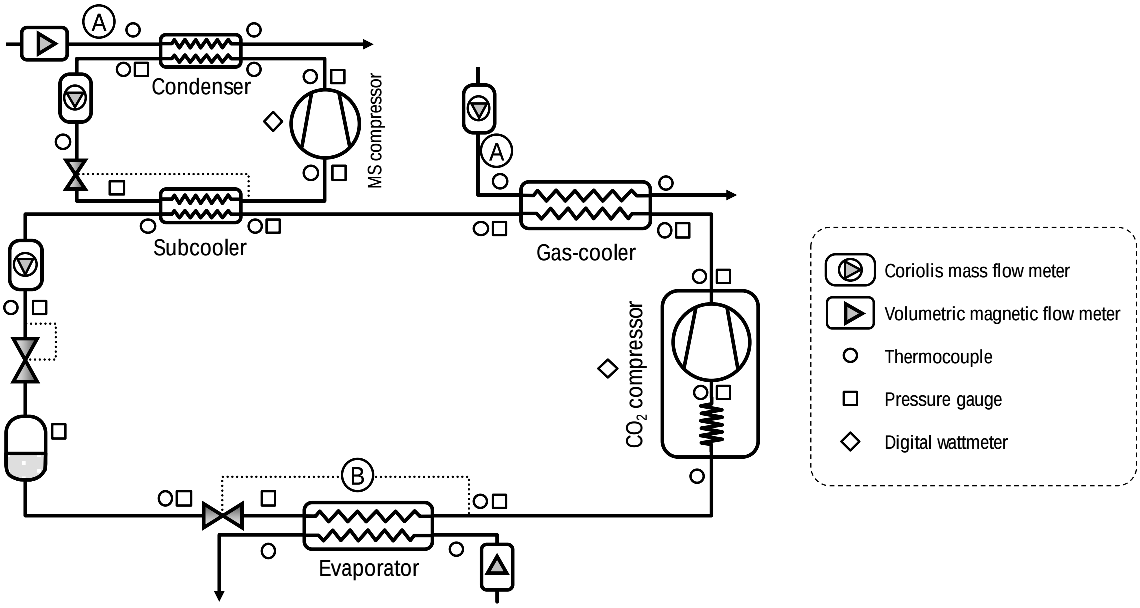


Figure 2. Schematic diagram of the test plant.

Table 1 presents the accuracies of the measurement devices. These accuracies have been used to calculate the main energy parameters measurement uncertainty, they have been evaluated using Moffat's method (1985).

Table 1. Accuracies and calibration range of the measurement devices

## 3. Experimental procedure and data validation

To quantify the effect of the mechanical subcooling system, the plant has been tested at several working conditions with and without subcooling in order to compare the behaviour of the plant in both configurations. The evaluated test conditions are:

- Heat rejection level: the plant has been studied at three different heat rejection temperatures: 24.0, 30.2 and 40.0 ºC, with maximum deviation of ± 0.35 ºC. These temperatures correspond to the inlet temperature of the water to the gas-cooler and to the condenser of the MS cycle (points A in Figure 2). The heat dissipation sink is the same for the gas-cooler and the MS condenser, because the inlet temperature of both heat exchangers is the same. The dissipation water flows are constant for all the studied conditions, being 0.32 $kg \cdot s^{-1}$ at the gas-cooler and 0.21 $kg \cdot s^{-1}$ at the condenser.
- Evaporation level: The plant has been tested at two $CO_2$ evaporating temperatures: 0.0 ± 0.3 ºC and -10.0 ± 0.3 ºC (point B in Figure 2). The evaporation level has been maintained artificially using an external secondary fluid system. It has not been tested using external loads, as done by Sánchez et al. (2014b), because there are large increments of the cooling capacity that will difficult to interpret the results.
- Gas-cooler pressure: All working conditions have been tested at a wide range of gas-cooler pressures by manual adjustment with the back-pressure valve.
- Compressors: Both compressors have worked at their nominal speed of 1450 rpm. Although lower velocity is possible in the MS compressor, it has been observed that the improvements are higher when higher the capacity of the MS cycle is.
- Electronic expansion valves: All the valves have been set to obtain a degree of superheat in the evaporator and subcooler of 5 ºC.

Each measurement point corresponds to a steady-state condition, obtained as average value of the operation of the plant during 15 minutes of stable operation with a sampling rate of 5 seconds.

Figure 3 shows heat transfer validation at the evaporator, the gas-cooler and the condenser of the MS cycle with and without subcooling. The average difference of the heat transfer rates between the refrigerant and the secondary fluid are of 4.3 % at the $CO_2$ evaporator, 3.3 % at the gas-cooler and 1.9% at the condenser of the MS cycle. Points A, B and C in Figure 3 have discrepancies greater than 10 % because those points are placed near the critical point, where the isobaric heat capacity of carbon dioxide varies significantly and small changes in temperature result in high measurement uncertainties (Torrella et al., 2011).

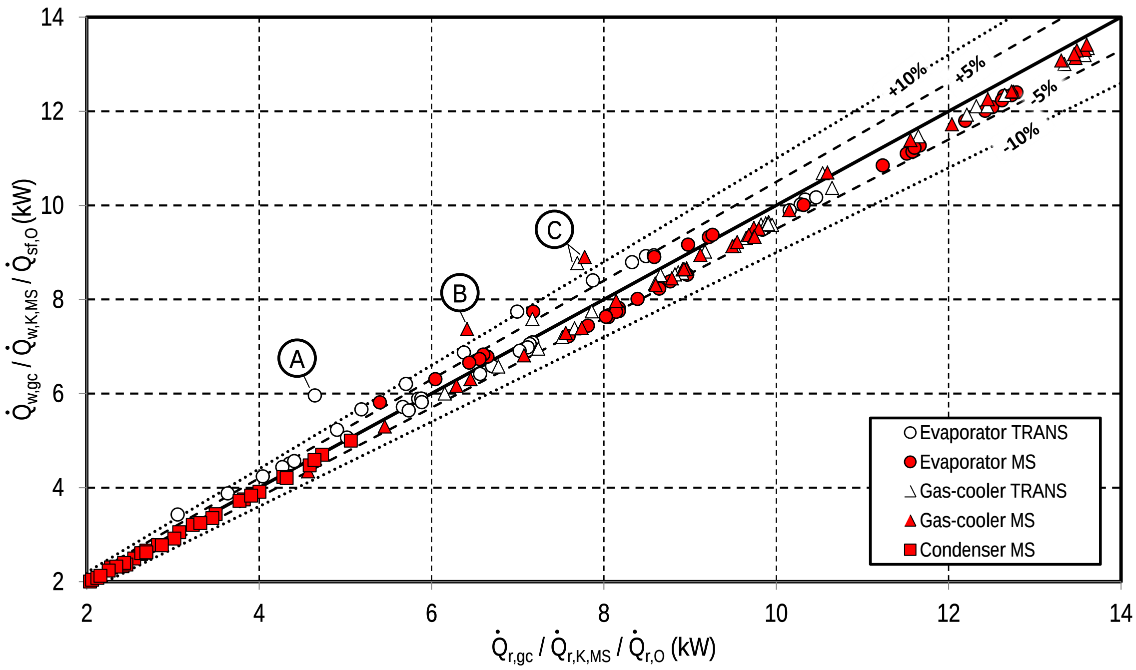


Figure 3. Heat transfer validation

## 4. Discussion of experimental results

### 4.1. Modification of the optimum working conditions

Experimental results indicate that the optimum operating conditions (gas-cooler pressure maximizing the overall COP) are different from those of the pure transcritical cycle, corroborating experimentally that studied previously by Llopis et al. (2015a).

To illustrate the reasoning, Figure 4 presents the pressure-enthalpy diagram of the transcritical cycle and the transcritical with the MS cycle at an evaporation temperature of 0.0 ºC and water inlet temperature of 30.2 ºC at the maximum COP condition. The main energy parameters for these conditions are detailed in Table 2. As it can be observed, the subcooling introduced by the MS cycle in the $CO_2$ at the exit of the gas-cooler causes: an increase of the specific cooling capacity, a reduction of the optimum high working pressure (reducing the compression ratio), an increase of the refrigerant mass flow rate (Table 2), and a reduction of the specific compression work. All the modifications are positive, however, to achieve them additional energy consumption in the MS compressor is needed.

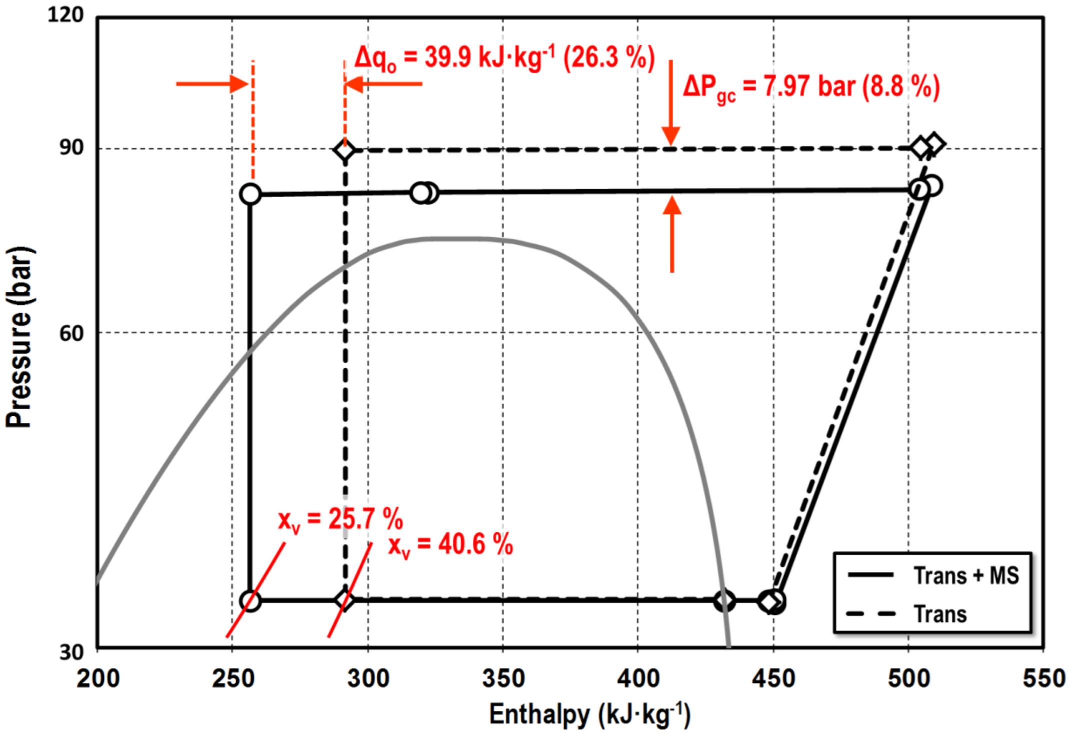


Figure 4. $CO_2$ pressure-enthalpy diagram with and without MS at $T_O$ = 0.0ºC, $T_{w,in}$ = 30.2ºC.

## 4.2. Cooling capacity

Cooling capacity provided by the cycle working with the MS cycle is obtained at the $CO_2$ evaporator. As presented in equation (4), it can be calculated as the product of the $CO_2$ mass flow rate and the specific cooling capacity at the evaporator. Considering the expansion processes isenthalpic, the specific cooling capacity can be split as enthalpy difference of $CO_2$ at the exit of the evaporator and the exit of the gas-cooler (equal to the specific cooling capacity that the pure transcritical cycle has) plus the enthalpy difference of $CO_2$ achieved in the subcooler. After manipulation of equation (4), considering that no energy losses exist at the subcooler, according to its energy balance (5), the cooling capacity can be expressed as presented by equation (6). For a given working gas-cooler pressure, the resulting capacity of the cycle, is obtained as the pure transcritical system capacity plus the cooling capacity of the MS cycle. Accordingly, from the point of view of capacity, the use of the dedicated mechanical subcooling system will be always positive.

$$\dot{Q}_o = \dot{m}_{CO2} \cdot q_{o,CO2} = \dot{m}_{CO2} \cdot \left(h_{O,out} - h_{gc,out} + \Delta h_{sub}\right) \quad (4)$$

$$\dot{m}_{CO2} \cdot \Delta h_{sub} = \dot{m}_{MS} \cdot q_{o,MS} \quad (5)$$

$$\dot{Q}_o = \dot{Q}_{o,CO2} + \dot{Q}_{o,MS} \quad (6)$$

The experimental measurements of cooling capacity for an evaporating level of 0.0 ºC are detailed in Figure 5 for the three dissipation temperatures with and without MS cycle. As it can be observed, high increments of capacity are achieved, especially when the transcritical plant operates at high pressures below the optimum condition of the pure transcritical system. For these cases, the increments on capacity reach up to 120 % at 30.2 ºC. That indicates that the MS cycle reduces the quick decrease of capacity when the plant operates at gas-cooler pressures below the optimum condition (Cabello et al., 2008). Although this is the maximum increment, the comparison must be done between the right operating points of the cycle, those at the maximum COP. These conditions, summarized in Table 2, result in increments on capacity of 23.1 % at 24.0 ºC, 34.0 % at 30.2 ºC and of 39.4 % at 40.0 ºC. Figure 6 shows experimental cooling capacity for an evaporating level of -10.0 ºC, at three dissipation temperatures. The mechanical subcooling, at the best COP conditions (Table 2), enhances the capacity in 24.2 % at 24.0 ºC, 41.1 % at 30.2 ºC and 55.7 % at 40.0 ºC of dissipation. It needs to be highlighted that the increments on capacity are higher at low evaporating levels and more prominent at high heat rejection temperatures.

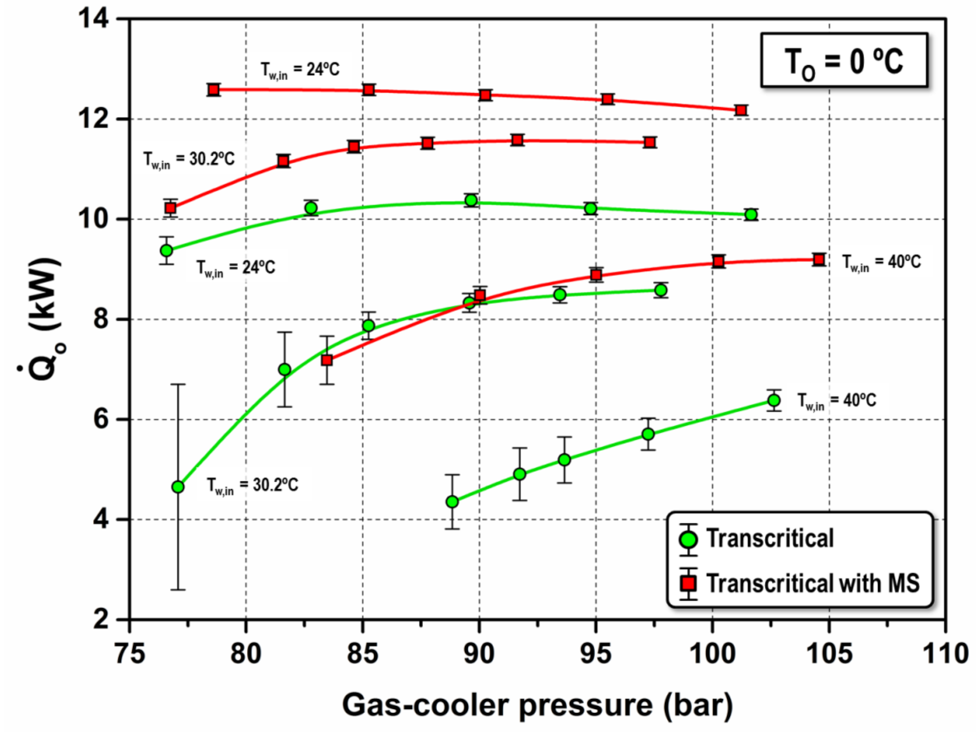


Figure 5. Cooling capacity with and without MS at $T_o$ = 0.0 ºC.

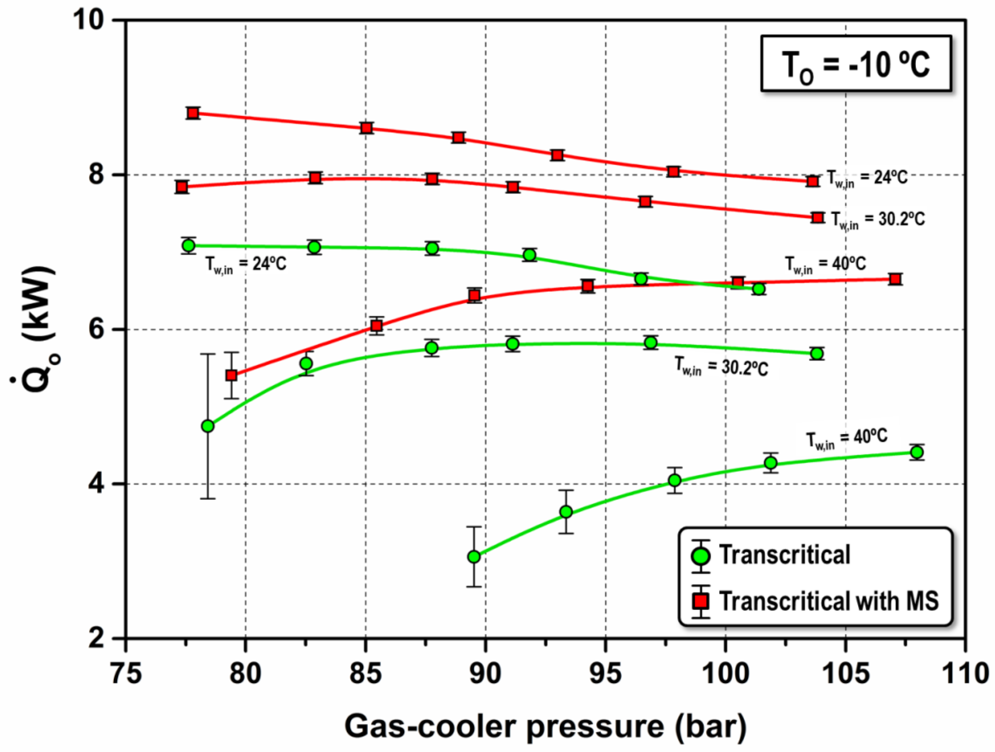


Figure 6. Cooling capacity with and without MS at $T_o$ = -10.0 ºC.

### 4.3. Coefficient of performance (COP)

Llopis et al. (2015a), from a theoretical approach, said that the COP of the cycle combination is higher than the COP of the transcritical cycle if $COP_{MS} > COP_{TRANS}$. For the same fluid and same compressor efficiencies, this condition is always satisfied if both cycles perform heat rejection to the same hot heat sink and the evaporating temperature of the MS cycle is higher than the evaporating temperature of the $CO_2$ cycle. For the data evaluated this condition occurs, accordingly the COP will be enhanced with the MS cycle.

The COP of the global cycle, Eq. (7), considers the cooling capacity obtained at the $CO_2$ evaporator, equation (4), and the power consumptions of the $CO_2$ compressor and the MS compressor.

$$COP = \frac{\dot{Q}_o}{P_{C,CO2} + P_{C,MS}} \tag{7}$$

Figure 7 presents the measured COP for an evaporating level of 0.0 ºC for three dissipation temperatures, operating with and without the MS cycle. High increments of COP have been measured, especially when the transcritical cycle is operated at pressures below the optimum. Nonetheless, these increments are not representative, since they are placed far away from the optimum condition. However, it needs to be highlighted that the MS cycle attenuates the quick COP drop when working below the optimum condition. The comparison must be done at the optimum working points, where the COP is the highest. For these conditions the use of the mechanical subcooling system increases the global COP by 10.9 % at 24 ºC, 22.1 % at 30.2 ºC and 26.1 % at 40 ºC. Results regarding the evaporating level of -10.0 ºC are plotted on Figure 8, where the COP increments introduced by the MS cycle can be observed. At -10.0ºC the increments at the optimum conditions are of 6.9 % at 24 ºC, 24.1 % at 30.2 ºC and 30.3 % at 40 ºC. For 40ºC, the maximum measured COP does not correspond to the optimum condition, since it is in pressures above 107 bar and it could not be reached.

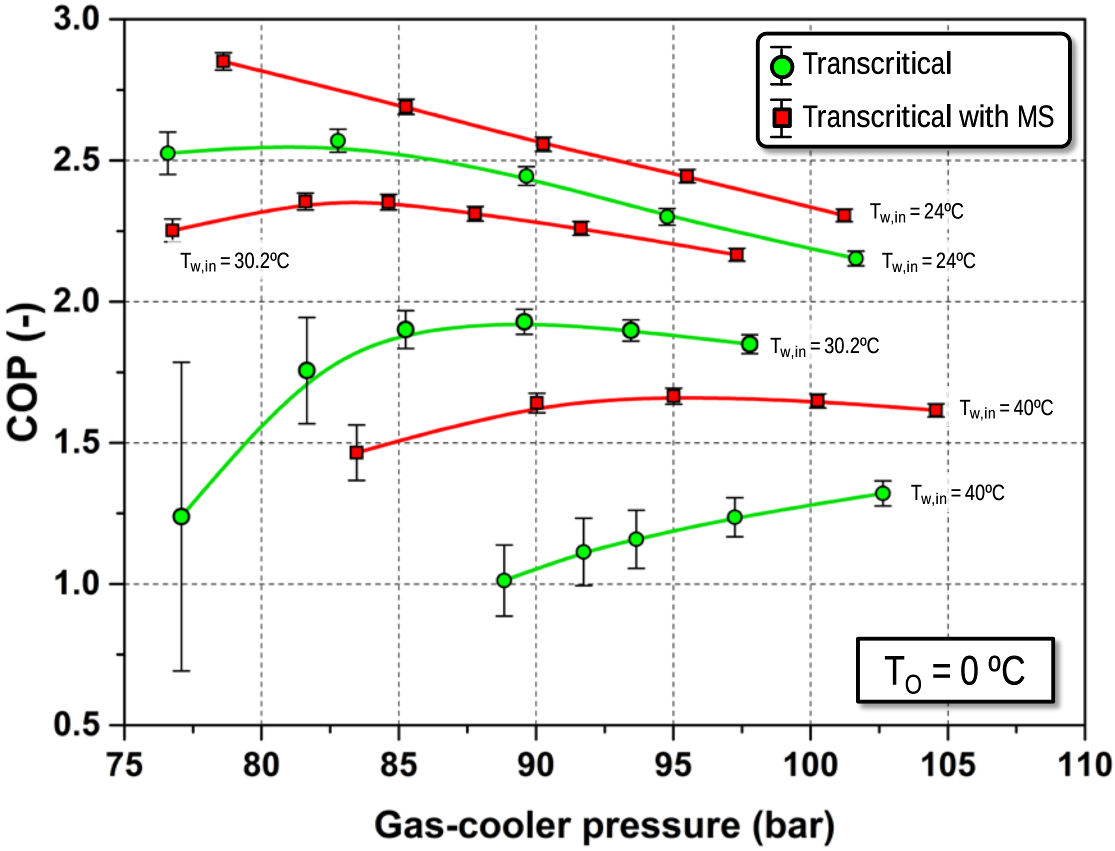


Figure 7. COP with and without MS at $T_O$ = 0.0 ºC.

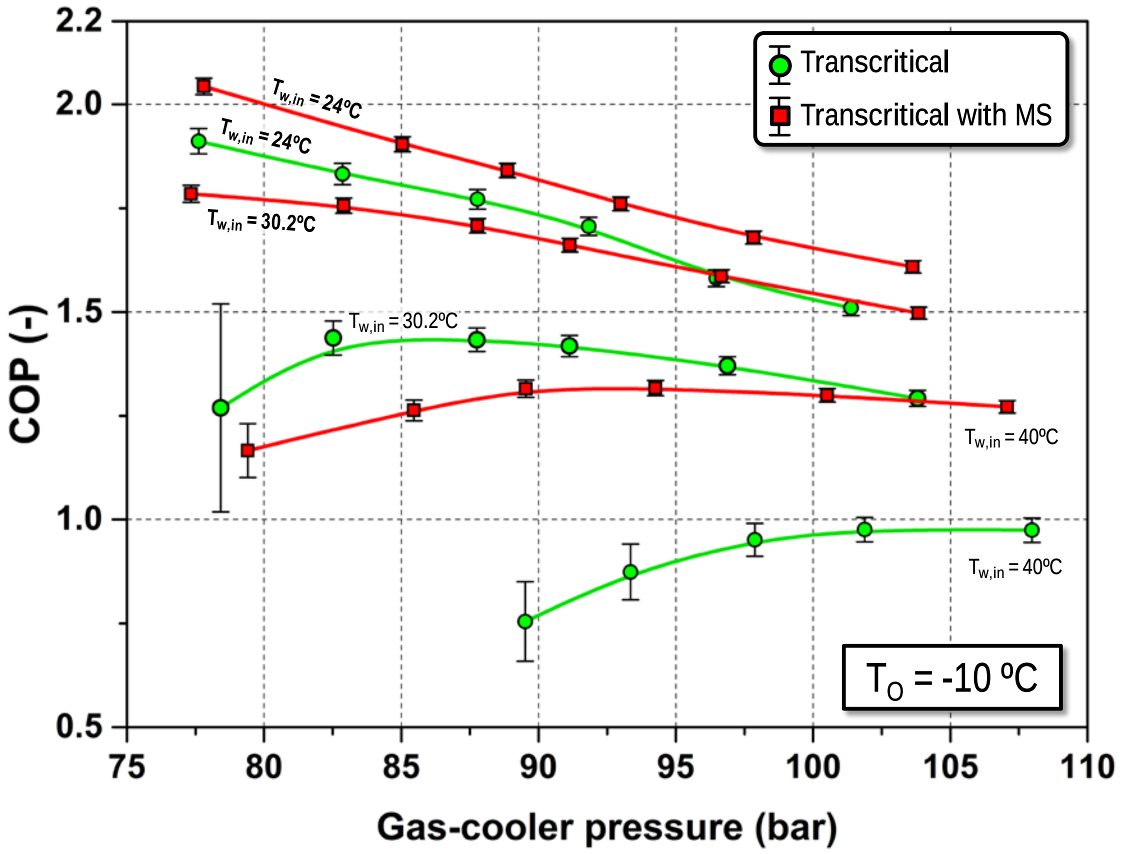


Figure 8. COP with and without MS at $T_O$ = -10.0 ºC.

It can be concluded that the MS cycle is beneficial for the COP and for the cooling capacity, being the improvement greater for the highest dissipation temperatures evaluated. So, it can be affirmed, as previously presented in a theoretical way (Llopis et al., 2015a), that the use of the MS cycle is more convenient for hot climates, although it is always favourable. It can also be affirmed that the use of the MS cycle in a $CO_2$ transcritical plant probably brings one of the highest COP increments measured up to now, and it must be seriously considered as a way to improve their energy efficiency.

## 5. Conclusions

This communication presents the experimental evaluation of the energy improvements that can be achieved in $CO_2$ transcritical refrigeration plants by using a dedicated mechanical subcooling cycle to subcool $CO_2$ at the exit of the gas-cooler. The evaluation, made using an experimental $CO_2$ plant and a dedicated mechanical subcooling cycle using R1234yf as refrigerant, has covered the evaporating levels of 0.0 ºC and -10.0 ºC and three heat rejection temperatures of (24.0, 30.2 and 40.0 ºC) at the nominal rotation speed of the compressors. All the experimental data has been validated comparing the heat transfer rates in the main heat exchangers.

The experimental evaluation has verified the modifications of the optimum operating conditions of the transcritical plant previously predicted theoretically, that are the increments of cooling capacity and COP, as well as reductions of the optimum gas-cooler pressure.

The use of the mechanical subcooling cycle softens the drop in capacity and COP of the transcritical cycle when working at pressures below the optimum gas-cooler pressure. For all the conditions, its use is recommended, since capacity and COP are enhanced all over the operating range. However, the best improvements in terms of COP have been obtained at high evaporating levels with high heat rejection temperatures. Accordingly, its use would be more convenient for hot climates.

The increments on capacity at the maximum measured COP conditions ranged from 23.1 to 39.4 % at an evaporating level of 0.0 ºC and from 24.2 to 55.7 % at -10.0 ºC. This increment in capacity would also allow reducing the size of the $CO_2$ compressors for a given application, although it has not been analysed in this work.

The increments in COP, for the best-tested conditions, ranged from 10.9 to 26.1 % at an evaporation level of 0.0ºC and from 6.9 to 30.3% at -10.0 ºC. The increments are higher for high heat rejection temperatures, however, the difference among the evaporating levels could not be evaluated since the optimum working conditions were out of the operation range of the plant.

Finally, it needs to be mentioned that this modification of the $CO_2$ transcritical refrigeration plants introduces one the highest improvements measured up to now. Accordingly, it must be seriously considered to help lowering the $CO_2$ equator to hottest countries. Furthermore, further research is needed to determine the optimum sizing of the compressor's combination, its effect in booster systems and its optimum working conditions in each case.

## 6. Acknowledgements

The authors gratefully acknowledge the Ministerio de Economía y Competitividad (project ENE2014-53760-R.7) and Jaume I University (project P11B2015-66) for financing this research work.

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Table 1. Accuracies and calibration range of the measurement devices

| Sensor | Measured variable | Measurement device | Calibration range | Calibrated accuracy |
|---|---|---|---|---|
| 21 | Temperature (ºC) | T-type thermocouple | -40.0 to 145.0 | ±0.5 |
| 6 | $CO_2$ pressure (bar) | Pressure gauge | 0.0 to 160.0 | ±1.2 |
| 4 | $CO_2$ pressure (bar) | Pressure gauge | 0.0 to 80.0 | ±0.7 |
| 2 | MS pressure (bar) | Pressure gauge | 0.0 to 16.0 | ±0.096 |
| 2 | MS pressure (bar) | Pressure gauge | 0.0 to 40.0 | ±0.24 |
| 1 | $CO_2$ refrigerant mass flow rate ($kg{\cdot}s^{-1}$) | Coriolis mass flow meter | 0.00 to 1.38 | ±0.1% of reading |
| 1 | MS refrigerant mass flow rate ($kg{\cdot}s^{-1}$) | Coriolis mass flow meter | 0.0 to 0.05 | ±0.1% of reading |
| 1 | Water mass flow rate in gas-cooler ($kg{\cdot}s^{-1}$) | Coriolis mass flow meter | 0.0 to 0.7 | ±0.1% of reading |
| 2 | Secondary fluid volume flow rates ($m^3{\cdot}s^{-1}$) | Magnetic flow meter | 0.0 to 4.0 | ±0.25% of reading |
| 2 | Power consumption (kW) | Digital wattmeter | 0.0 to 6.0 | ±0.5% of reading |

Table 2. Main energy parameters at the best-measured conditions with and without MS

| | | $P_{gc}$ (bar) | $T_{gc,o}$ (ºC) | $T_{sub,o}$ (ºC) | $\dot{Q}_o$ (kW) | $q_o$ (kJ·kg$^{-1}$) | $\dot{m}_{CO2}$ (kg·s$^{-1}$) | $x_v$ (%) | $P_{c,CO2}$ (kW) | $\dot{Q}_{o,MS}$ (kW) | $P_{c,MS}$ (kW) | $COP_{MS}$ (-) | $P_c$ (kW) | COP (-) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $T_O$ = 0.0 ºC | Water dissipation inlet temperature = 24.0 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 82.8 | 28.36 | - | 10.22 | 176.1 | 0.059 | 30.0 | 3.98 | - | - | - | 3.98 | 2.57 |
| | Trans + MS | 78.6 | 29.71 | 16.43 | 12.58 | 211.6 | 0.060 | 15.7 | 3.78 | 2.61 | 0.64 | 4.10 | 4.41 | 2.85 |
| | Variation (bar / %) | -4.2 | - | - | 23.1 | 20.2 | 1.8 | -14.3 | -5.0 | - | - | - | 11.0 | 10.9 |
| | Water dissipation inlet temperature = 30.2 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 89.6 | 33.51 | - | 8.33 | 145.3 | 0.057 | 38.7 | 4.32 | - | - | - | 4.32 | 1.93 |
| | Trans + MS | 81.6 | 34.78 | 23.23 | 11.16 | 193.1 | 0.058 | 24.2 | 4.03 | 3.14 | 0.71 | 4.45 | 4.74 | 2.35 |
| | Variation (bar / %) | -8.0 | - | - | 34.0 | 32.8 | 1.6 | -14.5 | -6.6 | - | - | - | 9.8 | 22.1 |
| | Water dissipation inlet temperature = 40.0 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 102.6 | 41.69 | - | 6.38 | 117.5 | 0.054 | 50.8 | 4.83 | - | - | - | 4.83 | 1.32 |
| | Trans + MS | 95.0 | 41.70 | 32.37 | 8.89 | 162.8 | 0.055 | 31.8 | 4.48 | 3.46 | 0.85 | 4.05 | 5.34 | 1.67 |
| | Variation (bar / %) | -7.6 | - | - | 39.4 | 38.5 | 1.6 | -18.9 | -7.2 | - | - | - | 10.5 | 26.1 |
| $T_O$ = -10.0 ºC | Water dissipation inlet temperature = 24.0 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 77.6 | 27.00 | - | 7.08 | 177.7 | 0.040 | 34.9 | 3.71 | - | - | - | 3.71 | 1.91 |
| | Trans + MS | 77.8 | 26.89 | - | 8.80 | 238.0 | 0.039 | 13.1 | 3.71 | 1.72 | 0.60 | 2.88 | 4.31 | 2.04 |
| | Variation (bar / %) | 0.2 | - | - | 24.2 | 33.9 | -2.3 | -21.8 | 0.1 | - | - | - | 16.2 | 6.9 |
| | Water dissipation inlet temperature = 30.2 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 82.5 | 32.79 | - | 5.56 | 146.5 | 0.039 | 44.1 | 3.87 | - | - | - | 3.87 | 1.44 |
| | Trans + MS | 77.3 | 33.42 | 20.87 | 7.84 | 205.6 | 0.039 | 26.6 | 3.69 | 2.84 | 0.71 | 4.00 | 4.40 | 1.78 |
| | Variation (bar / %) | -5.2 | - | - | 41.1 | 40.3 | 1.2 | -17.5 | -4.7 | | | | 13.6 | 24.1 |
| | Water dissipation inlet temperature = 40.0 ºC | | | | | | | | | | | | | |
| | Transcritical cycle | 101.9 | 41.09 | - | 4.27 | 123.2 | 0.035 | 53 | 4.38 | - | - | - | 4.38 | 0.98 |
| | Trans + MS | 107.1 | 40.48 | 22.27 | 6.65 | 198.8 | 0.035 | 25 | 4.50 | 2.00 | 0.73 | 2.72 | 5.23 | 1.27 |
| | Variation (bar / %) | 5.2 | - | - | 55.7 | 61.4 | -0.5 | -27.9 | 2.6 | - | - | - | 19.4 | 30.3 |