Current limits of CO2 compressors working in integrated mechanical subcooling cycles

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Abstract. CO2 systems used in refrigeration are becoming more complex with the aim of improving their energy performance. The integrated mechanical subcooling system is being studied to enhance the COP and the cooling capacity of the plants as one of the possible solutions for hot climates in the last years.

Last studies determined their optimum working conditions where the COP is maximum. However, the compressor of the integrated mechanical subcooling cycle works out of the range of operation in some cases. This work presents the main technical limitations of this compressor in an experimental plant.

The main objective is to determine what these limits are and to highlight the need to develop new compressors for these applications. Improving the operation of these compressors will further increase the energy performance of the overall system.

1. Introduction
Carbon dioxide refrigeration systems have been the target of many of the scientific investigations in recent years, with the aim of improving classical CO2 systems to make them more competent, especially in hot climates. All of these efforts have been driven by the F-Gas Regulation [1] which limits the use of high GWP refrigerants in many of the refrigeration applications. It is necessary to look for refrigerants with low GWP and for certain applications, CO2 is the only candidate with low GWP that guarantees safety and is neither flammable nor toxic (A1 ASHRAE classification). Although CO2 systems are widely used in northern Europe, in areas where the average annual temperature is higher, these systems suffer a significant decrease in performance.

Subcooling methods define a clear line of research that is gaining much weight at the moment [2]. Llopis et al. [3] compiled in a general way the effects of subcooling on CO2 cycles and reviewed all the existing methods. Initially, with the most basic subcooling methods, such as the internal heat exchanger [4] and later with more complex systems such as dedicated mechanical subcooling DMS [5, 6], which uses a refrigerant different from CO2 in the auxiliary cycle.
The Integrated Mechanical Subcooling (IMS) cycle, presented in this document, only uses CO\(_2\) as refrigerant. The purpose of this system is to subcool the CO\(_2\) at the gas-cooler outlet thanks to a CO\(_2\) flow extracted from the main cycle and that is expanded, passing through the subcooler and being re-compressed until the inlet of the gas-cooler. The interest of this cycle is that it is simpler than the dedicated one; it has fewer components and works only with CO\(_2\). Like DMS, this subcooling cycle increases the plant's cooling capacity and COP with respect to the base cycle [7].

The IMS system was first proposed by Shapiro's patent [8]. Cecchinato et al. [9] evaluated the system from a theoretical point of view and obtained an increase of 17.3% in energy efficiency in relation to a basic single-stage CO\(_2\) cycle for an evaporation level of -10°C and an outlet temperature of gas-cooler at 30°C. Later, Catalán-Gil et al., [7] analyzed the thermodynamic models of the IMS and the DMS for supermarket applications, achieving annual reductions in energy consumption between 2.9% and 3.4% for warm areas and between 1.3% and 2.4% for temperate regions. Nebot-Andrés et al. [10] study the IMS system from a theoretical approach, optimizing the gas-cooler pressure and degree of subcooling, reaching increases of 15.9% for −10°C of evaporation temperature and 35°C of ambient temperature compared to the CO\(_2\) cycle with intermediate exchanger [11].

Integrated mechanical subcooling represents an important interest for the improvement of CO\(_2\) cycles, since it has great potential for improvement. This cycle was experimentally tested by Nebot-Andrés et al. [12] and its optimal conditions have been determined.

The experimental results showed that the pressure conditions and degree of subcooling for which the maximum COP values were achieved correspond to operating points that are outside the application limits of the IMS compressor. Consequently, this work has been developed to highlight the operating limits of these systems and emphasize the need to develop equipment that adapts to these operating conditions.

2. Experimental plant description
This section presents the experimental installation used to evaluate the optimal conditions of the CO\(_2\) transcritical cycle with the integrated mechanical subcooling system. The most important details of the main components of the cycle are provided and the measurement system used in the plant is described.

2.1. Experimental plant
The scheme of the experimental plant tested in this work is shown in Figure 1 and its real plant in Figure 2. The plant is a CO\(_2\) single-stage transcritical refrigeration system with an integrated mechanical subcooling system extracting gas at the exit of the subcooler. The main single-stage refrigeration cycle uses a semi-hermetic compressor with a displacement of 3.48 m\(^3\)·h\(^{-1}\) at 1450 rpm and a nominal power of 4 kW. The expansion is carried out by a double-stage system, composed of an electronic expansion valve (back-pressure) controlling the gas-cooler pressure, a liquid receiver between stages and an electronic expansion valve, working as thermo-static, to control the evaporating process. Evaporator and gas-cooler are brazed plate counter current heat exchangers with exchange surface area of 4.794 m\(^2\) and 1.224 m\(^2\), respectively. The subcooler is situated directly downstream of the gas-cooler. It is a brazed plate heat exchanger with an exchange surface area of 0.850 m\(^2\). It works as evaporator of the mechanical subcooling system and subcools the CO\(_2\) at the exit of the gas-cooler. The mechanical subcooling cycle is driven by a variable speed semi-hermetic compressor with displacement of 1.12 m\(^3\)·h\(^{-1}\) at 1450 rpm. The expansion valve of the IMS cycle is electronic, working as thermostatic.
Heat dissipation in gas-cooler is done with a water loop, simulating the heat rejection level. The evaporator is supplied with another loop, working with a propylene glycol–water mixture (60% by volume) that enables a constant entering temperature in the evaporator. Both the mass flow and the inlet temperature are controlled in these loops.

2.2. Measurement system
The thermodynamic properties of the working fluids are obtained thanks to the measurement system presented in Figure 1. All fluid temperatures are measured by 18 T-type thermocouples. The thermocouples placed at the evaporator and at the exit of gas-cooler and subcooler are immersion thermocouples. 11 pressure gauges are installed along all the circuit. CO2 mass flow rates are measured by two Coriolis mass flow meters, as well as dissipation water flow of the gas-cooler, which is measured using another one. The flow of the other secondary fluids is measured by a magnetic volumetric flow
Compressors’ power consumptions are measured by two digital wattmeters. The accuracies of the measurement devices are presented in Table 1.

| Measured variable                        | Measurement device            | Range            | Calibrated accuracy  |
|------------------------------------------|-------------------------------|------------------|----------------------|
| Temperature (ºC)                         | T-type thermocouple           | -40.0 to 145.0   | ±0.5K                |
| CO₂ pressure (bar)                       | Pressure gauge                | 0.0 to 160.0     | ±0.6% of span        |
| CO₂ pressure (bar)                       | Pressure gauge                | 0.0 to 100.0     | ±0.6% of span        |
| CO₂ pressure (bar)                       | Pressure gauge                | 0.0 to 60.0      | ±0.6% of span        |
| CO₂ main mass flow rate (kg·s⁻¹)         | Coriolis mass flow meter      | 0.00 to 1.38     | ±0.1% of reading     |
| CO₂ IMS mass flow rate (kg·s⁻¹)          | Coriolis mass flow meter      | 0.00 to 0.083    | ±0.1% of reading     |
| Water mass flow rate (kg·s⁻¹)            | Coriolis mass flow meter      | 0.00 to 13.88    | ±0.1% of reading     |
| Glycol volume flow rate (m³·h⁻¹)         | Magnetic flow meter           | 0.0 to 4.0       | ±0.25% of reading    |
| Power consumption (kW)                   | Digital wattmeter             | 0.0 to 6.0       | ±0.5% of reading     |

Table 1. Accuracies of the measurement devices.

### 2.3. Experimental test procedure

This section describes the strategy carried out to do the experimental tests in order to determine the optimal cycle conditions. Three heat rejection levels and three different evaporation temperatures have been tested.

- All tests are carried out under transcritical conditions.
- Hot sink temperatures: 25.0, 30.4 and 35.1 ºC (maximum deviation of ± 0.20 ºC). The inlet temperature of the water in the gas-cooler remains fixed at the stipulated values and the flow rate remains fixed at 1.167 m³·h⁻¹.
- Cold sink temperatures: The propylene glycol-water inlet temperature in the evaporator and the flow rate are maintained. The temperatures tested are -1.3, 3.8 and 10.1 ºC (± 0.23 ºC). The flow is fixed at 0.7 m³·h⁻¹.
- The pressure in the gas-cooler is regulated with an electronic Back-Pressure thanks to a PID controller. In each test the pressure is varied in order to identify the optimum and reach the maximum COP conditions.
- The main compressor always runs at nominal speed (1450 rpm). The IMS compressor speed varies to modify the subcooling degree.
- The electronic expansion valves were configured to obtain a useful degree of superheating of 10K in the evaporator and 5K in subcooler.

All tests are carried out under stationary conditions of at least 10 minutes, taking data every 5 seconds, obtaining the test point as the average value of the entire test. Thermodynamic properties are determined based on data measured using Refprop v.9.1.[13].

### 3. Optimization of the plant and main energy results

This section details how the existence of an optimum COP has been determined experimentally, and how the plant has been evaluated to identify the optimal working conditions.

#### 3.1. Optimization process

To obtain the maximum COP, the pressure and the subcooling values are modified following a method similar to a Simplex algorithm. When having three initial points, gas-cooler pressure or subcooling degree were increased or decreased following the trend of the previous points, to get closer and closer to the maximum COP point, until the increases achieved between the new value and the previous one were less than 1%.
Figure 3. COP and cooling capacity as a function of gas-cooler pressure and subcooling for \( t_{\text{w,in}}=35.1^\circ\text{C} \) and \( t_{\text{g,in}}=10.0^\circ\text{C} \).

Figure 3 shows the COP and the cooling capacity measured for water inlet conditions of 35.1°C and glycol inlet of 10.0°C, for different pressure levels and degrees of subcooling. The existence of a maximum COP is clearly observed. Regarding the capacity of the plant, it is observed that it is greater the greater the subcooling degree is. It means that the capacity of the plant can be adjusted by modifying the degree of subcooling.

3.2. COP and cooling capacity

The evolution of the COP for all the evaluated conditions is shown in Figure 4 (left) and the cooling capacity that the plant is providing under maximum COP conditions is represented in Figure 4 (right).

Regarding the COP, a clear trend in its evolution can be seen, marked by the inlet temperatures of water and glycol. It can be perceived that the COP is lower when the glycol inlet temperature is lower, that is, when the evaporation level is lower. It can also be observed that, as the inlet water temperature increases, the COP decreases. The measured values range from 1.40 to 1.87 for \( t_{g,\text{in}}=-1.3^\circ\text{C} \), from 1.56 to 2.13 for \( t_{g,\text{in}}=3.8^\circ\text{C} \) and from 1.81 to 2.48 for \( t_{g,\text{in}}=10.0^\circ\text{C} \).

For the cooling capacity, a linear trend can be observed depending on the hot sink temperature, reducing capacity as the inlet water temperature increases. Likewise, it is clearly seen how the capacity
is greater when the evaporation level is greater. The measured values range from 6.5kW to 7.7kW for $t_{g,in}=-1.3^\circ C$, from 7.3kW to 8.9kW for $t_{g,in}=3.8^\circ C$ and from 8.6kW to 10.3kW for $t_{g,in}=10.0^\circ C$.

4. Operation of the auxiliary compressor at optimum conditions

As mentioned in the previous sections, this system bases its operation on the use of an auxiliary compressor that recompresses to gas-cooler pressure a part of the CO$_2$ mass flow that has been evaporated in the subcooler. The operation of this compressor is very particular since it must adapt its rotational speed to obtain a specific subcooling degree for which the plant's COP is maximum.

Table 2 presents the main operating parameters of the IMS compressor for the optimal points that have been obtained experimentally and presented in section 3. The presented parameters are the rotation frequency, the compression ratio, the global and volumetric performances and the suction temperature. As can be seen, the compression ratios of all the tests are very low and specifically less than 1.5 for some test cases (test points 2, 6, 7, 8 and 9). These compression ratios are outside the operating range of the compressor, which must work with a minimum compression ratio of 1.5. Regarding the frequency, we observe that all the optimal points are achieved with very low frequencies, in some cases lower than 30Hz (test points 1, 2, 4, 5, 7 and 9). For this type of plant, a smaller compressor should therefore be implemented, but the one used is the smallest existing of this type of compressors. Regarding the volumetric efficiency, the compressor operates with values between 40 and 50% while the global efficiency’s values are better (except point 1) due to the low compression ratio. Finally, another operating parameter that needs to be highlighted is the temperature obtained in the suction of the compressor.

| Test point | $t_{g,in}$ (°C) | $t_{w,in}$ (°C) | $\tau$ | $\eta_v$ | $f$ (Hz) | $t_{suc,ims}$ (°C) | $\eta_g$ |
|------------|----------------|----------------|--------|---------|--------|-------------------|--------|
| 1          | -1.3           | 25             | 1.8    | 0.42    | 29     | 8.4               | 0.52   |
| 2          | -1.3           | 30             | 1.4    | 0.42    | 27     | 21.5              | 0.75   |
| 3          | -1.3           | 35             | 1.7    | 0.47    | 32     | 20.0              | 0.76   |
| 4          | 3.8            | 25             | 1.7    | 0.42    | 28     | 13.9              | 0.72   |
| 5          | 3.8            | 30             | 1.5    | 0.44    | 29     | 20.3              | 0.78   |
| 6          | 3.8            | 35             | 1.4    | 0.54    | 33     | 26.1              | 0.79   |
| 7          | 10.0           | 25             | 1.4    | 0.40    | 25     | 20.6              | 0.70   |
| 8          | 10.0           | 30             | 1.3    | 0.51    | 30     | 28.0              | 0.86   |
| 9          | 10.0           | 35             | 1.2    | 0.42    | 25     | 33.9              | 0.91   |

Table 2. Main operation parameters of the IMS compressor.

In Figure 5 the application limits of this particular compressor in terms of gas-cooler pressure and evaporation temperature can be seen. The experimental data measured in the plant for the IMS compressor are shown in red. As can be clearly seen, although the pressure levels are correct, the evaporation temperature of the IMS cycle is too high, falling outside the compressor operation boundary.
COP values presented in section 3.2 are quite lower than those obtained in previous theoretical studies [10]. The experimental COP are in average 21.5% lower than the values obtained theoretically. A large part of this reduction may be due to the work of the compressor.

5. Conclusions
This work presents the experimental test of a CO$_2$ transcritical refrigeration plant with integrated mechanical subcooling working at optimum conditions. The heat rejection levels of 25.0°C, 30.4°C and 35.1°C and the evaporator inlet temperatures of -1.3°C, 3.8°C and 10°C are evaluated under steady-state conditions. The main compressor operates at nominal speed, while the speed of the auxiliary compressor is varied to obtain the optimum degree of subcooling. It is a system with a very high potential for improvement compared to classic CO$_2$ refrigeration systems.

Experimental tests have demonstrated the existence of a maximum COP, obtained under optimal conditions of pressure and subcooling degree. One of the optimized parameters is controlled by modifying the auxiliary compressor’s speed. However, results showed that to reach the optimum COP conditions, the auxiliary compressor must work out of the operability limits, working with high suction temperatures and compression ratios too short.

In conclusion, it would be interesting to develop CO$_2$ compressors for this application, making the overall system performance even better.

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