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Additional Information

Experimental evaluation of the energy efficiency of a CO_2 refrigerating plant working in transcritical conditions

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Abstract

This work presents the experimental evaluation of the energy efficiency and optimal gas-cooler pressures of a single-stage refrigerating plant working with carbon dioxide as refrigerant in transcritical conditions. The performance of the plant was tested at three different evaporating temperatures (-0.9, -10.1 and -18.1 ºC), for three gas-cooler refrigerant outlet temperatures (31.2, 33.6 and 40.0 ºC.) at each evaporating temperature and in a wide range of gas-cooler pressures (74.4 to 104.7 bar).

The experimental tests enabled us to calculate accurately the optimal gas-cooler pressures and compare them with the most commonly used relations to define this value in single-stage refrigerating cycles operating with carbon dioxide in transcritical conditions. Furthermore, an analysis of the reduction in energy efficiency produced in the plant if the optimum pressure is not well defined is also presented.

Keywords: carbon dioxide; CO_2; refrigerating plant; optimum pressure; gas-cooler; transcritical cycle

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Nomenclature

COP	coefficient of performance
$\dot{m}$	mass flow rate ($kg \cdot s^{-1}$)
P	pressure (bar)
P_C	compressor power consumption (kW)
$\dot{Q}_O$	cooling capacity (kW)
T	temperature (ºC)
$\dot{V}$	Volumetric flow rate ($m^3 \cdot h^{-1}$)

Subscripts

BP	pressostatic expansion valve
dis	compressor discharge
env	environment
g	water / ethylene-glycol mixture
GC	gas-cooler
hp	high pressure
i	element inlet
IHX	internal heat exchanger
inMot	inlet port to compressor
lp	low pressure
LR	liquid receiver
O	evaporator
o	element outlet
opt	optimum
ref	refrigerant
suc	compressor suction
w	water

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1. Introduction

The greenhouse effect associated to artificial refrigerants [1, 2] is forcing the scientific community to devise and develop solutions related to the air-conditioning and refrigeration fields in order to avoid this problem. Many efforts are focused on using natural refrigerants, such as hydrocarbons (butane, propane and isobutane) [3] and carbon dioxide [4, 5], because of their low or minimal contribution to direct greenhouse effect [6, 7].

Among these groups of natural refrigerants, with a minimal impact on the environment, carbon dioxide seems to be the trend for the future owing to its safety characteristics, but a lot of effort needs to be made in order to overcome the technical problems and in order to achieve a level of energy efficiency comparable with that achieved using HFCs. Within carbon dioxide technical advantages, it could be highlighted its high heat transfer coefficients in the supercritical region [8] and its high pressure levels combined with low specific volumes, which allow the size of the components to be reduced. On the side of disadvantages, carbon dioxide needs special equipment due to the high working pressures, and a special regulation it is required because temperature and pressure are decoupled in the supercritical working region and a maximum efficiency is presented [8, 9].

Experimental results from prototypes in the automobile industry [10, 11] have shown that it is possible to obtain a similar level of energy efficiency with respect to R134a, but for the time being, this comparison could not be extended to other applications. Some other researchers are experimenting in other sectors, examples being Girotto *et al.*

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[12] in commercial refrigeration with two-stage compression systems, Cavallini *et al.* [13] and others [14] in air conditioning, Nekså [15, 16] in heat pumps where the results are quite good, and some others that are developing new cycle configurations [17, 18] and new elements [19, 20]. But what is common to all transcritical carbon dioxide applications is the existence of an optimal pressure in the gas-cooler at which the maximum efficiency of the refrigerating cycle is achieved. Liao *et al.* [21], Sarkar *et al.* [9], Kauf [22] and Chen *et al.* [23] have worked, from a theoretical point of view, on the definition of this optimal pressure and its dependence on the refrigerating cycle variables, which are mainly the refrigerant outlet temperature in the gas-cooler (related to environment temperature in the case of Chen and Kauf) and the evaporating temperature.

In this work, we present the experimental evaluation of the energy efficiency of a carbon dioxide refrigerating facility working in transcritical conditions for several evaporating pressures at various refrigerant outlet temperatures in the gas-cooler, where the cycle efficiency was evaluated for a wide range of gas-cooler pressures. The experimental results were used to compare the optimal pressure allocation with the expressions proposed by the authors [9, 21–23]; the deviations of these expressions from the experimental values and their implications in the COP of the plant are also discussed in this work.

2. Experimental plant description

The main equipment that makes up the experimental plant developed to carry out the evaluation of carbon dioxide as refrigerant working in a transcritical cycle (Figure 1) is: a 4kW semi-hermetic single-stage vapour compressor, a double stage expansion system

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with an liquid receiver between stages, concentric counter current gas-cooler and evaporator and internal heat exchanger, as shown in Figure 2, although in this paper only the results obtained while operating without the internal heat exchanger are presented. The double stage expansion system consists of a pressostatic expansion valve (back-pressure) that is employed for the first expansion and an electronic expansion valve for the second. The first expansion stage, performed by the pressostatic expansion valve, allows the gas-cooler outlet pressure to be controlled, and the second enables us to control the evaporating process by means of the electronic expansion valve whose sensors are placed at the inlet and outlet of the evaporator. The liquid receiver is placed after the first expansion stage in order to regulate the mass of refrigerant in the plant.

Figure 1. Experimental carbon dioxide refrigerating plant

Figure 2. Schematic plant diagram

Plant regulation is performed by two auxiliary systems. The first one, devoted to controlling the heat rejected in the gas-cooler, consists of a loop working with water that allows the required refrigerant outlet temperature to be obtained at the gas-cooler. The second one, which is used to supply the evaporator with the refrigerant load, consists of a loop working with an ethylene-glycol mixture (50/50% by volume) that enables a constant pressure to be maintained in the evaporator. Further information about the auxiliary systems can be found in previous works presented by the authors [24].

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The thermodynamic properties of the working fluids are obtained at the entrance and exit of each element by measuring them with T-type thermocouples (fixed over the pipe surface for all the elements except the compressor discharge temperature, which is registered with a T-type immersion thermocouple placed inside the discharge chamber) and piezoelectric pressure transducers. Refrigerant mass flow rate is measured with a Coriolis-effect mass flow meter, secondary fluid volume rates with magnetic flow meters, compressor power consumption with a digital wattmeter and compressor speed with an analogical signal obtained from the inverter drive of the compressor, which was calibrated by means of a frequency analyser system with an accelerometer placed on the compressor head. All the sensors were calibrated, and the calibration range and accuracies are those shown in Table 1. All the signals were collected by a data acquisition system and processed on-line using a LABVIEW-based application developed by the authors [25] that uses the REFPROP dynamic routines [26] to obtain the thermodynamic properties of carbon dioxide and water, and interpolated polynomials from the ASHRAE Handbook [27] for the properties of the ethylene-glycol mixture.

Table 1. Accuracies of the measurement devices

3. Test procedure and data validation

Operation and efficiency of transcritical carbon dioxide refrigerating systems are highly influenced by pressure and refrigerant temperature at the gas-cooler outlet [8, 19], these variables being independent in the supercritical region and, therefore, the test campaign was designed in accordance with the independence of these variables. The experimental tests were performed for fixed gas-cooler refrigerant outlet temperatures operating at fixed evaporating temperatures while varying the gas-cooler pressure. The campaign

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consisted of 92 steady-state tests, each lasting 20 minutes, which are grouped into 9 sets depending on the gas-cooler outlet temperature and evaporating temperature, as shown in Table 2. The tests were performed at two compressor speeds (1450 and 1120 rpm).

Table 2. Experimental data test range

Thermodynamic properties of the refrigerant in the plant were evaluated using temperature and pressure measurements except for the point corresponding to the refrigerant outlet of the gas-cooler, whose enthalpy was estimated using a steady-state energy balance with the secondary fluid in the gas-cooler, as well as the gas-cooler outlet temperature used to perform the tests (presented in Table 2). This methodology was applied because in the critical point region a small error in the measurement of the temperature or pressure implies a large error in the enthalpy calculation. The evaporator inlet enthalpy of the refrigerant was evaluated considering the two expansion processes isenthalpic, and data validation was performed by comparing the secondary fluid heat transfer in the evaporator to the heat transfer in the refrigerant using the above mentioned enthalpy. The results of the validation for all the tests are presented in Figure 3, showing an agreement within ± 5%.

Figure 3. Energy balance validation at the evaporator

4. Experimental data analysis and discussion

In this section, the experimental results concerning energetic efficiency and optimal pressure of the carbon dioxide refrigerating plant working in a single-stage configuration are presented and analysed.

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Figure 4 presents the experimental measurements of cooling capacity, compressor power consumption and refrigerant mass flow rate evolutions for a gas-cooler pressure variation test at an evaporating temperature of -10.0ºC and a gas-cooler refrigerant outlet temperature of 40.2ºC. As can be observed in Figure 4, both the compressor power consumption and the refrigerant mass flow rate variations respond to a linear function of the gas-cooler pressure, while the cooling capacity presents a high decrease at low gas-cooler pressures. The division of these trends results in a maximum COP for a given gas-cooler pressure, which is discussed next.

Figure 4. Compressor power consumption, refrigerant mass flow rate and cooling capacity at several gas-cooler pressures. T_O -10.0ºC. $T_{GC,o}$ 40.2ºC

Figures 5, 6 and 7 present the experimental values of the COP reached by the plant operating at evaporating temperatures of -18.1, -10.1 and -0.9 ºC, respectively, at several gas-cooler outlet temperatures for a wide range of gas-cooler pressures (Table 2).

Figure 5. COP at an evaporating temperature of -18.1ºC for different gas-cooler outlet temperatures

Figure 6. COP at an evaporating temperature of -10.1ºC for different gas-cooler outlet temperatures

Figure 7. COP at an evaporating temperature of -0.9ºC for different gas-cooler outlet temperatures

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From the experimental measurements presented in Figures 5 to 7 it was observed that a maximum efficiency is given for a certain gas-cooler pressure, as presented by several authors [8, 21]. This optimal gas-cooler pressure strongly depends on the refrigerant outlet temperature, the optimal pressure value being higher when the refrigerant outlet temperature is also higher. In addition, this optimum pressure value depends on the evaporating temperature, since the lower evaporating temperature is the higher the optimal gas-cooler pressure will be. Furthermore, a high COP decrease was measured when the gas-cooler pressure was below the optimal value – a decrease that gets higher as the gas-cooler refrigerant outlet temperature approaches the critical temperature of carbon dioxide (30.978ºC), as shown in Figure 8, and when the evaporating temperature is higher. Kauf [22] reported a small variation in the allocation of the optimal pressure while varying the compressor speed, but in this case this possible variation was neglected because the variation in the compressor speed between tests (≈ 330 rpm) was lower than that considered by Kauf (≈ 1000 rpm).

Several authors [9, 21-23] have obtained theoretical expressions to define the optimal gas-cooler pressure in a refrigerating cycle similar to the one analysed in this paper. All of them, except Kauf [22], consider a single-stage refrigerating cycle working with an internal heat exchanger in their mathematical reasoning, but the final expressions presented are simplified by considering null efficiency in this heat exchanger, so all the expressions can be compared to the values obtained experimentally.

$$P_{opt} = (2.78 - 0.0157 \cdot T_O) \cdot T_{GC,o} + (0.381 \cdot T_O - 9.34) \text{ (bar)} \qquad \text{Liao } et\ al. \text{ [21]} \qquad (1)$$

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$$P_{opt} = 4.9 + 2.256 \cdot T_{GC,o} - 0.17 \cdot T_O + 0.002 \cdot T_{GC,o}^{\ 2} \quad \text{(bar)}$$ Sarkar *et al.* [9] (2)

$$P_{opt} = 2.6 \cdot T_{env} \approx 2.6 \cdot T_{GC,o} + 7.54 \quad \text{(bar)}$$ Kauf [22] (3)

$$P_{opt} = 2.304 \cdot T_{env} + 19.29 \quad \text{(bar)}$$ Chen *et al.* [23] (4)

All the above expressions are based on several assumptions that differ from one another and also vary with the experimental behaviour of the facility. Thus, Liao and Sarkar considered an ideal compression process and neglected the influence of superheating at compressor suction in the optimal pressure; however, a real compression process based on experimental data on real compressors was considered in the expressions of Chen and Kauf, who also neglected the degree of superheating at compressor suction. Moreover, the range of validity of the expressions is different. Liao developed his expression for an evaporating temperature range from -10 to 20ºC and gas-cooler refrigerant outlet temperature from 30 to 60ºC; Sarkar considered an evaporating temperature from -10 to 10ºC and 30 to 50ºC for the gas-cooler refrigerant outlet temperature (this expression was obtained for a heat pump/refrigerating cycle combination but it is also considered), and both Chen and Kauf developed their expressions for a constant evaporating temperature of 5.3ºC and for an environment temperature from 35 to 50ºC. Nevertheless, only Kauf expressed the mathematical relation using the gas-cooler refrigerant outlet temperature as a variable by considering a gas-cooler approach temperature of 2.9ºC, while Chen expressed his relation by using an experimental fitted curve (5) obtained from experimental data [10]. The relation (5) that represents the approach between the gas-cooler refrigerant outlet temperature and the air temperature was obtained from a gas-cooler cooled with air, but it is also considered in this work.

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$$T_{GC,o} - T_{env} = -0.0015269 \cdot T_{env}^{\;2} - 0.028866 \cdot T_{env} + 7.7126 \tag{5}$$

Given that the assumptions taken in order to develop the mathematical expressions to define the optimal gas-cooler pressure differ from the experimental behaviour of the real refrigerating cycle, a comparison between the optimal gas-cooler pressures obtained with expressions (1) to (4) and the optimal gas-cooler pressures from the experimental tests is presented. In order to compare the optimal values a simplified model was adjusted from experimental data to model the compressor power consumption and cooling capacity, and with this model the optimal values in the experimental tests were obtained.

Table 3 presents the experimental optimal gas-cooler pressures and their deviations with respect to the optimal pressures obtained using the authors' expressions inside their validity range, except for Chen's and Kauf's, which are not within the validity range but are also considered in order to analyse the influence of the evaporating pressure on the optimum pressure of the gas-cooler. The comparison shows that the best expression that matches the experimental results is the Sarkar's, which has a maximum deviation below 1.5%. Liao's expression presents a deviation smaller than 5% but the one presented by Kauf does not represent the experimental behaviour of the facility, mainly because this expression neglects the influence of the evaporating pressure in the optimal pressure allocation and this variable exerts an important influence on the gas-cooler optimal pressure, as seen in Figures 5 to 7. But on the other hand, the expression developed by Chen represents the experimental behaviour of the facility accurately, although it does not consider the influence of the evaporating pressure. However, it does consider an

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isentropic efficiency obtained from experimental data [10] and a correlation for the gas-cooler approach.

From the comparison presented in Table 3, it can be concluded that the expressions that best fit the experimental behaviour of the plant are the ones that consider an ideal compression process (Liao and Sarkar), an assumption that it is not real but works well according to the results.

Table 3. Experimental gas-cooler optimal pressures and deviations from authors' expressions

The importance of the definition of the optimal gas-cooler pressure lies in the reduction in energy efficiency that occurs if this value is not well defined. This reduction in efficiency, according to the experimental plant, is represented in Figure 8 for operation at an evaporating temperature of -1ºC and with several gas-cooler refrigerant outlet temperatures. The lines in Figure 8, which show the reduction in COP when the gas-cooler pressure is not optimum also represent the optimal values obtained with the expressions presented above. Figure 8 shows that the reduction in COP due to a deviation with respect to the optimum pressure is higher when the gas-cooler refrigerant outlet temperature is close to the critical temperature, and this reduction is larger if the pressure is under the optimal value, as seen in Figures 5 to 7. Furthermore, if the authors' expressions are used to obtain the optimum pressure the reductions in COP in the real plant reach a maximum of 1.25% in the case of Liao's expression, 0.2% in the case of Sarkar's, 0.6% in the case of Chen's and 12% in the case of Kauf's .

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Figure 8. Reduction in COP for optimal pressure deviation in % at an evaporating temperature of -1ºC for several gas-cooler outlet temperatures

As can be seen in Figure 8, it is obvious that refrigerating plants that use carbon dioxide as refrigerant need a precise system to control their operation in optimal conditions since a small error in precision in the gas-cooler pressure causes a high reduction in the efficiency of the plant. If the precision error could not be avoided it should overestimate the optimal pressure, since the reduction in COP is then smaller than if the optimal pressure is underestimated.

6. Conclusions

In this work the energy efficiency and the optimal gas-cooler pressures of an experimental single-stage refrigerating plant operating with carbon dioxide as refrigerant in a transcritical cycle have been presented. The results show that the optimal gas-cooler pressures depend on the gas-cooler refrigerant outlet temperature as well as on the evaporating temperature.

The experimental optimal gas-cooler pressures were contrasted with the most commonly used relations to define this value, which are Liao's, Kauf's, Chen's and Sarkar's expressions. Results show that the one that best represents the experimental performance of the plant is Sarkar's, although this expression was developed considering an ideal compression process and a simultaneous heat pump/refrigerating cycle combination.

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Furthermore, it can be concluded that a precise system to control the gas-cooler pressure is needed in this type of refrigerating plants, since a small error in pressure causes a strong reduction in efficiency, and if the precision error could not be avoided it should overestimate the optimal pressure since the reduction in COP is then smaller than if the optimal pressure is underestimated.

7. Acknowledgments

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TABLES

Sensors	Measured Variable	Measurement device	Calibration Range	Calibrated accuracy
17	Temperature (ºC)	T-type thermocouple	-40.0 to 145.0	± 0.5
6	Pressure (bar)	Pressure gauge	0.0 to 160.0	± 1.2
4	Pressure (bar)	Pressure gauge	0.0 to 80.0	± 0.7
1	Refrigerant mass flow rate (kg/s)	Coriolis mass flow meter	0.00 to 1.38	± 0.1% of reading
2	Secondary fluid volume rates (m^3/h)	Magnetic flow meter	0 to 4	± 0.25% of reading
1	Power consumption (kW)	Digital wattmeter	0 to 6	± 0.5% of reading
1	Compressor speed (rpm)	Analogical signal from the inverter drive	0 to 1550	± 1.3% of reading

Table 1. Accuracies of the measurement devices

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Evaporation Temperature (°C)		Calculated $T_{GC,o}$ (°C)		Gas-cooler pressure range (bar)		Useful superheating (°C)	Unuseful superheating (°C)	Compressor Speed (rpm)
-18.2	± 0.5	31.1	± 0.6	100.3	76.2	7.6	5.0	1449.0
-18.1	± 0.6	33.4	± 0.7	98.6	81.8	7.2	5.0	1450.4
-18.1	± 0.7	37.9	± 0.5	100.0	89.4	7.1	5.1	1448.6
-10.3	± 0.7	31.5	± 0.7	100.8	75.8	5.4	4.5	1449.4
-10.1	± 0.6	34.3	± 0.6	101.9	82.1	5.4	4.6	1451.1
-10.0	± 0.5	40.2	± 0.7	103.6	88.9	5.6	3.4	1450.9
-1.1	± 0.5	31.3	± 0.8	97.4	74.4	5.9	2.8	1121.2
-0.9	± 0.4	33.4	± 0.6	98.1	77.5	4.4	3.8	1121.0
-0.8	± 0.8	40.4	± 0.5	104.7	89.9	5.1	3.2	1121.3

Table 2. Experimental data test range

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T_O (°C)	$T_{GC,o}$ (°C)	Experimental $P_{GC,opt}$ (bar)	Optimal pressure and deviation from experimental optimal values in (%)							
			Liao S. M. *et al.*[21]		Sarkar J. *et al.* [9]		Kauf F. [22]		Chen Y. *et al.* [23]	
			$P_{GC,opt}$ (bar)	Deviation (%)	$P_{GC,opt}$ (bar)	Deviation (%)	$P_{GC,opt}$ (bar)	Deviation (%)	$P_{GC,opt}$ (bar)	Deviation (%)
-18.2	31.1	82.5	*out of range*		*out of range*		89.0	8.5	77.6	5.4
-18.1	33.4	89.4	*out of range*		*out of range*		94.6	5.1	83.2	7.5
-18.1	37.9	95.9	*out of range*		*out of range*		106.3	10.7	95.0	1.0
-10.3	31.5	78.8	79.5	1.1	79.9	1.5	89.6	13.9	78.2	2.9
-10.1	34.3	85.7	87.6	2.0	86.4	0.6	96.7	12.7	85.3	3.9
-10.0	40.2	101.2	104.9	3.1	100.6	1.1	112.1	10.2	101.0	1.3
-1.1	31.3	76.9	77.8	1.2	77.7	1.1	89.0	15.7	77.6	0.7
-0.9	33.4	82.4	83.9	1.9	82.9	0.7	94.7	15.0	83.3	0.3
-0.8	40.4	98.5	103.1	4.6	99.5	0.9	112.6	14.2	101.4	3.3

Table 3. Experimental gas-cooler optimal pressures and deviations from authors' expressions

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FIGURE CAPTIONS

Figure 1. Experimental carbon dioxide refrigerating plant

Figure 2. Schematic plant diagram

Figure 3. Energy balance validation at the evaporator

Figure 4. Compressor power consumption, refrigerant mass flow rate and cooling capacity at several gas-cooler pressures. T_O -10.0ºC. $T_{GC,o}$ 40.2ºC

Figure 5. COP at an evaporating temperature of -18.1ºC for different gas-cooler outlet temperatures

Figure 6. COP at an evaporating temperature of -10.1ºC for different gas-cooler outlet temperatures

Figure 7. COP at an evaporating temperature of -0.9ºC for different gas-cooler outlet temperatures

Figure 8. Reduction in COP for optimal pressure deviation in % at an evaporating temperature of -1ºC for several gas-cooler outlet temperatures

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Figure 1

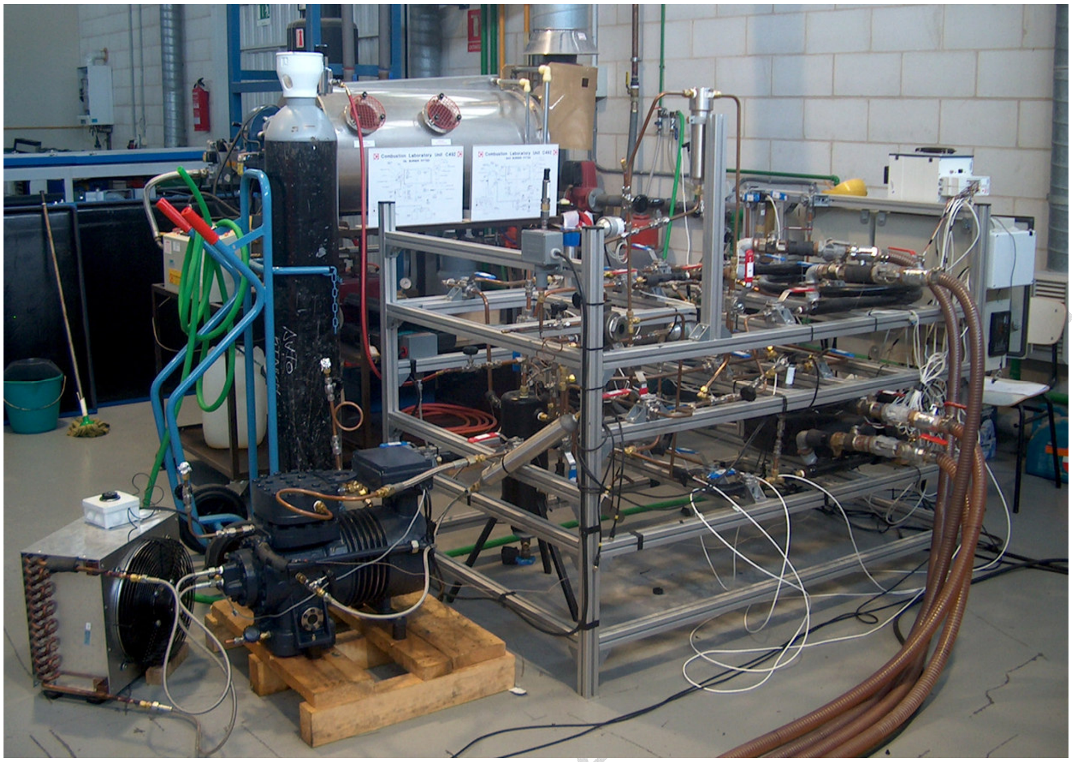

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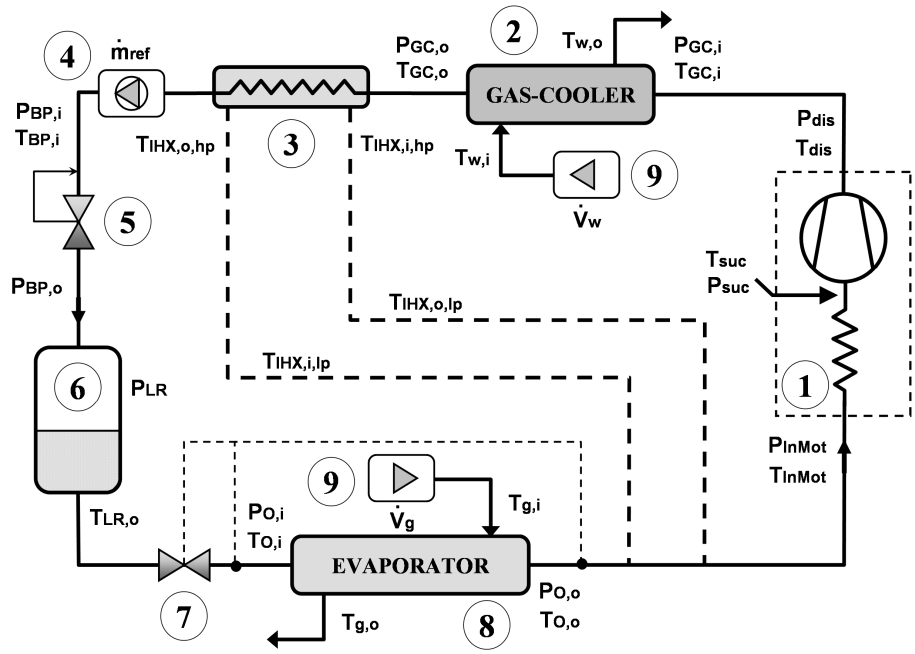
Figure 2



1 – Semi-hermetic compressor	4 – Coriolis mass flow meter	7 – Electronic expansion valve
2 – Gas-cooler	5 – Pressostatic expansion valve	8 – Evaporator
3 – Internal heat exchanger	6 – Liquid receiver	9 – Magnetic volumetric flow meter

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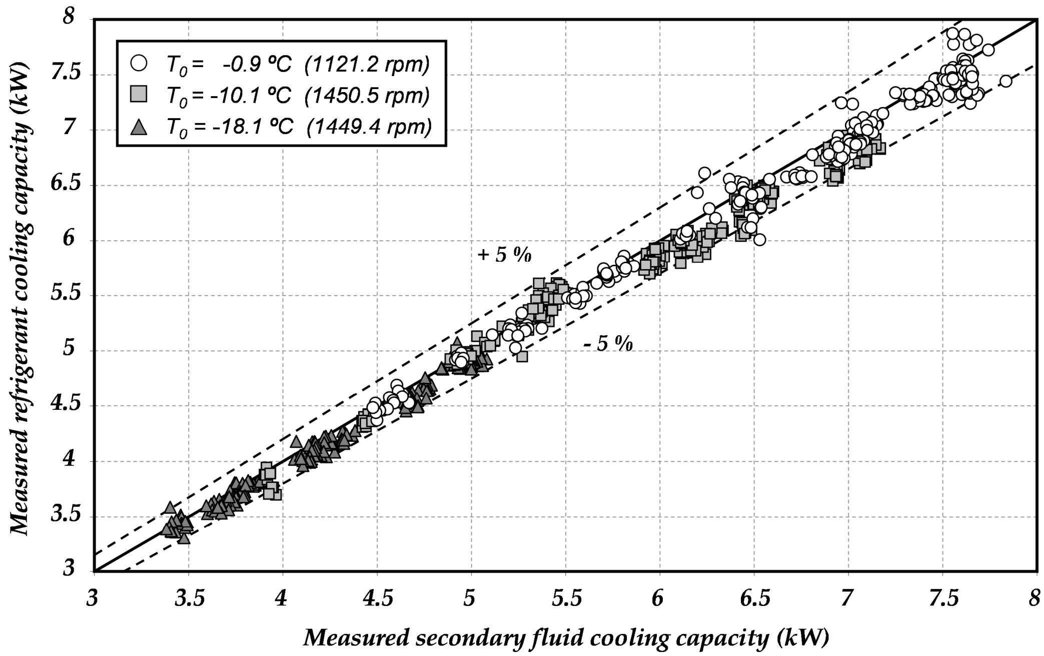
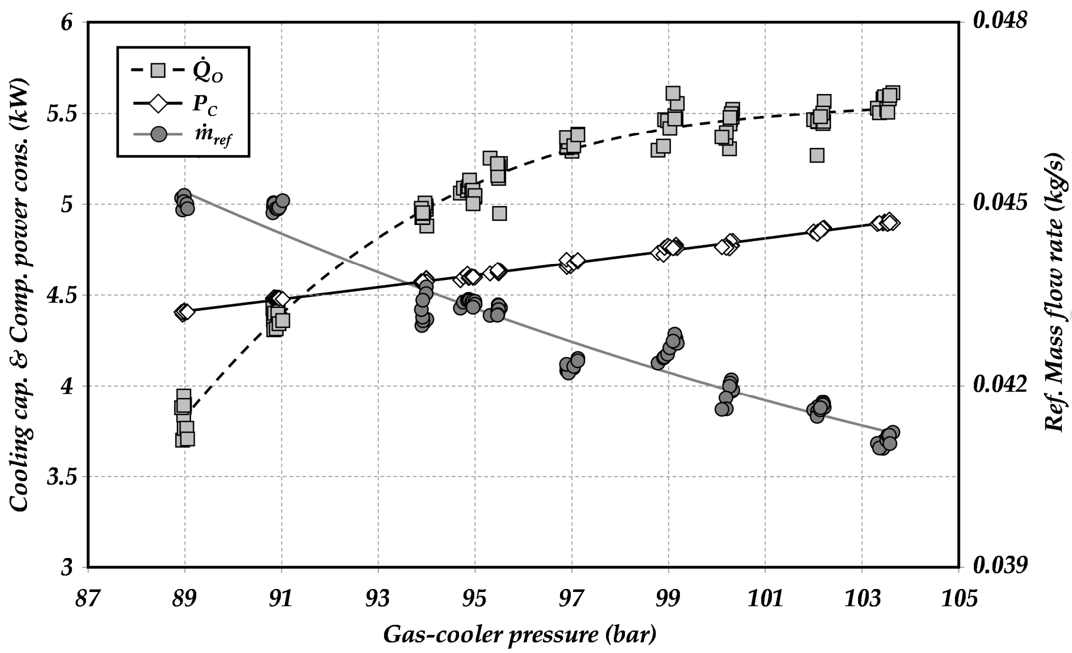
Figure 3



ACCEPTED MANUSCRIPT

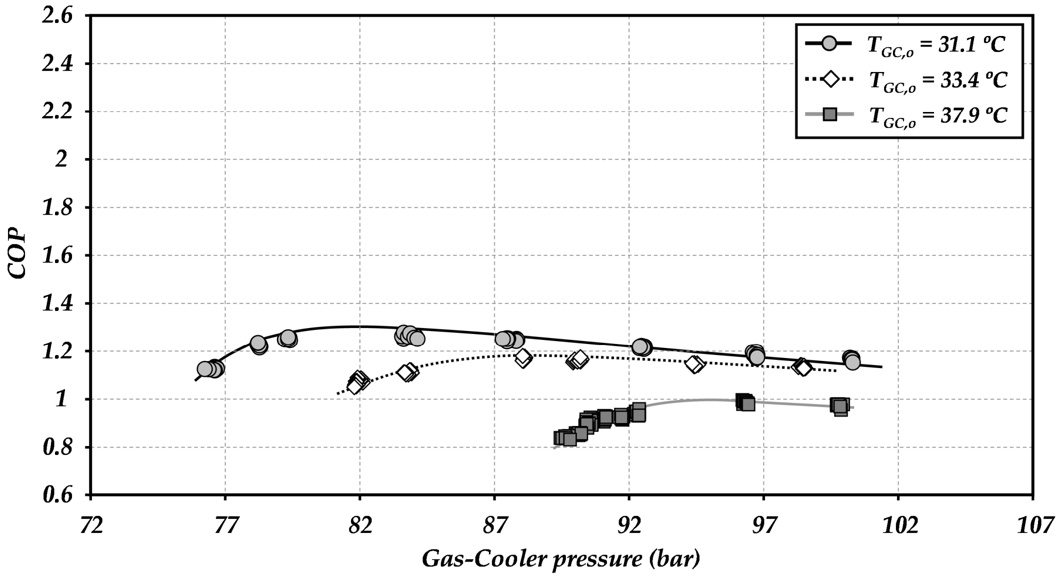
ACCEPTED MANUSCRIPT

Figure 4



ACCEPTED MANUSCRIPT

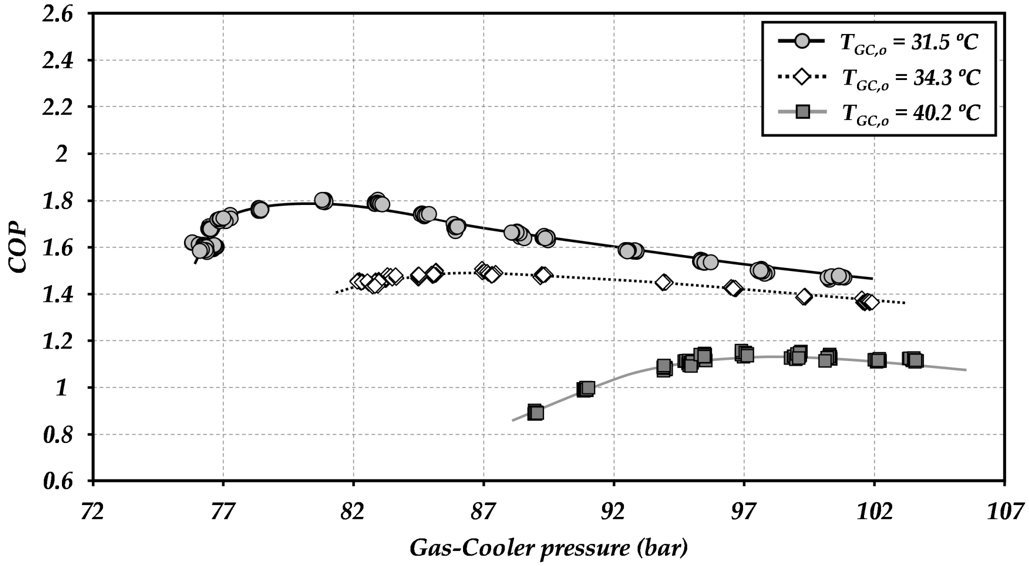
Figure 5



ACCEPTED MANUSCRIPT

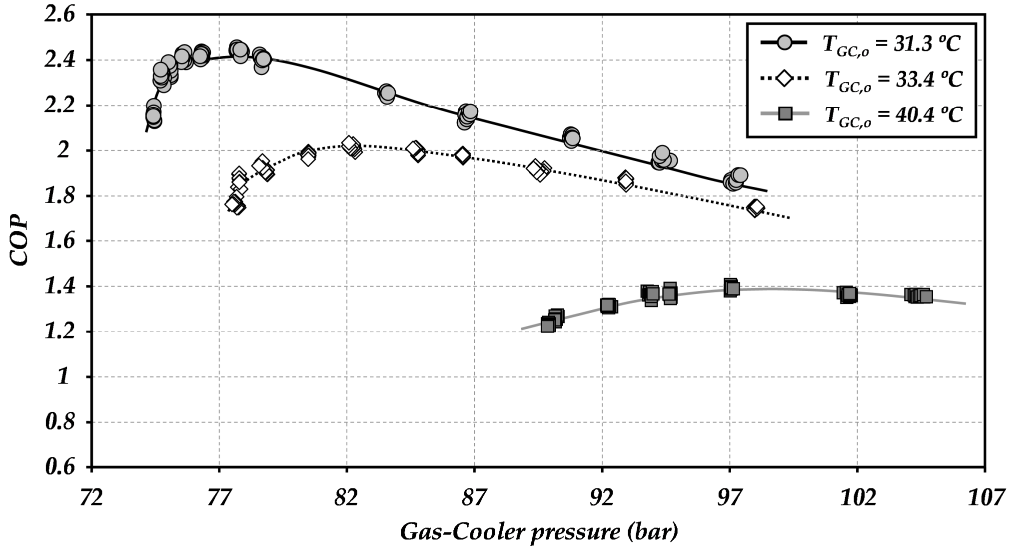
ACCEPTED MANUSCRIPT

Figure 6



ACCEPTED MANUSCRIPT

Figure 7



ACCEPTED MANUSCRIPT

ACCEPTED MANUSCRIPT

Figure 8