NUMERICAL ASSESSMENT OF THE USE OF A DEDICATED MECHANICAL SUBCOOLING SYSTEM DURING HOT WATER GENERATION IN A WATER TO WATER TRANSCRITICAL CO_2 HEAT PUMP

Fernando Illán Gómez[1], José Ramón García Cascales[1], Francisco Javier Sánchez Velasco[1] and Ramón A. Otón Martínez[2]

1: Departamento de Ingeniería Térmica y de Fluidos, ETSII, Universidad Politécnica de Cartagena. C/ Dr. Fleming s/n 30202, Cartagena, Murcia, España; 2: Departamento de Ingeniería y Técnicas Aplicadas, Centro Universitario de la Defensa en la Academia General del Aire. C/ López Peña s/n 30720, San Javier, Murcia, España

Fernando Illán Gómez
fernando.illan@upct.es

Abstract: *This paper presents a numerical study of the use of a dedicated mechanical subcooling (DMS) system using R1234yf, for hot water generation in a water-to-water CO2 heat pump. Compressor mass flow rates and power consumptions were modeled using the manufacturer's correlations, expansion valves were modeled as isenthalpic, and heat exchangers were modeled by deriving correlations for the evaporation/condensation pressure and heat transfer rate. In the condenser, IMST-ART was used to obtain condensation pressure and heat transfer rate. A cell-by- cell discretization model was used for the evaporator, which was a transcritical CO2, subcritical R1234yf heat exchanger. Three different systems were compared for the transcritical CO2 cycle: with internal heat exchanger (IHX), with DMS, and with IHX+DMS. Results showed that, for the conditions studied (hot water generation up to 60 ºC and evaporator water inlet temperature from 5- 25 ºC), the use of a DMS does not improve the performance of the system.*

Keywords: Water heater, transcritical cycle, heat pump, CO_2.

1. INTRODUCTION

Transcritical CO_2 heat pumps are currently improving their penetration in the market of the hot water generation especially in the building sector due to, among other factors, the properties of CO_2 as a refrigerant (A1 type, natural, cheap, and with a greenhouse warming potential (GWP) of 1) and, the high coefficient of performance (COP) obtained especially when the inlet water temperature is low [1,2]. However, as the inlet temperature of the water to be heated increases, the COP of the system decreases. This is due to the fact that these systems usually use a water-refrigerant heat exchanger working under counter-flow conditions as a gas cooler and, as the inlet water temperature increases, the enthalpy of the refrigerant (CO_2) at the outlet of the gas cooler also increases [3,4]. This limits the specific heat absorbed at the evaporator and, consequently, also the specific heat transferred at the gas cooler and the overall COP. In order to improve the performance of the system under these conditions, this work seeks to evaluate whether the use of a dedicated mechanical subcooling (DMS) system coupled to the transcritical CO_2 heat pump cycle can improve the overall performance of the coupled system and what are the operational parameters that optimize the system [5-7].

Thus, this paper presents a numerical study of the use of a DMS system using R1234yf as refrigerant, during hot water generation in a water-to-water transcritical CO_2 heat pump under different working conditions.

The paper is structured as follows: the following section describes the model developed in MATLAB. This section also describes the different operating conditions that have been introduced to the model in order to study the influence of the DMS system on the performance of the heat pump. The third section describes the results of the comparison of the three different systems that have been compared for the transcritical CO_2 cycle: with internal heat exchanger (IHX), with DMS, and with IHX and DMS. Finally, the last section summarizes the main conclusions drawn.

2. THE NUMERICAL MODEL

Figure 1 shows different configurations studied in this work for hot water generation. Figure 1a shows the base CO_2 cycle with IHX and without DMS, Figure 1b shows a CO_2 cycle with DMS and without IHX, Figure 1c shows a CO_2 cycle with DMS and IHX in which the DMS evaporator/subcooler is located before the IHX, whereas Figure 1d shows a similar CO_2 cycle with IHX and DMS in which the IHX is located before the DMS evaporator/subcooler.

A fundamental design condition when a DMS cycle is considered to be coupled to a base cycle (such as the case here with the transcritical CO_2 cycle), is that the COP of the auxiliary cycle (i.e., the DMS cycle) must be greater than the COP of the base cycle (i.e., CO_2 cycle). This and other conditions were evaluated for the coupled system under different working conditions in order to evaluate if the introduction of the DMS cycle improves the overall performance of the system. In this work, the global COP of the facility is defined as:

$$COP = \frac{q_{gc}+q_{cond}}{w_c+w_{c2}} \qquad (1)$$

where q_{gc} is the specific heat at the gas cooler of the main cycle, q_{cond} is the specific heat at the condenser of the DMS cycle, and w_c are the specific works at the compressors. It has to be taken into account that the refrigerant flow rate at the R1234yf cycle can vary depending on the working conditions, and it can be estimated as:

$$\dot{m}_{R1234yf} = \dot{m}_{CO2} \cdot \frac{\Delta h_{CO2}}{\Delta h_{R1234yf}} \qquad (1)$$

where Δh_{C02} and $\Delta h_{R1234yf}$ are the enthalpy change of CO_2 and R1234yf at the heat exchanger acting as CO_2 cycle subcooler and R1234yf cycle evaporator.

In this work a MATLAB model was developed for the main components (compressor, condenser, expansion valve, and evaporator) of the DMS system. The main characteristics of those components are summarized in Table 1

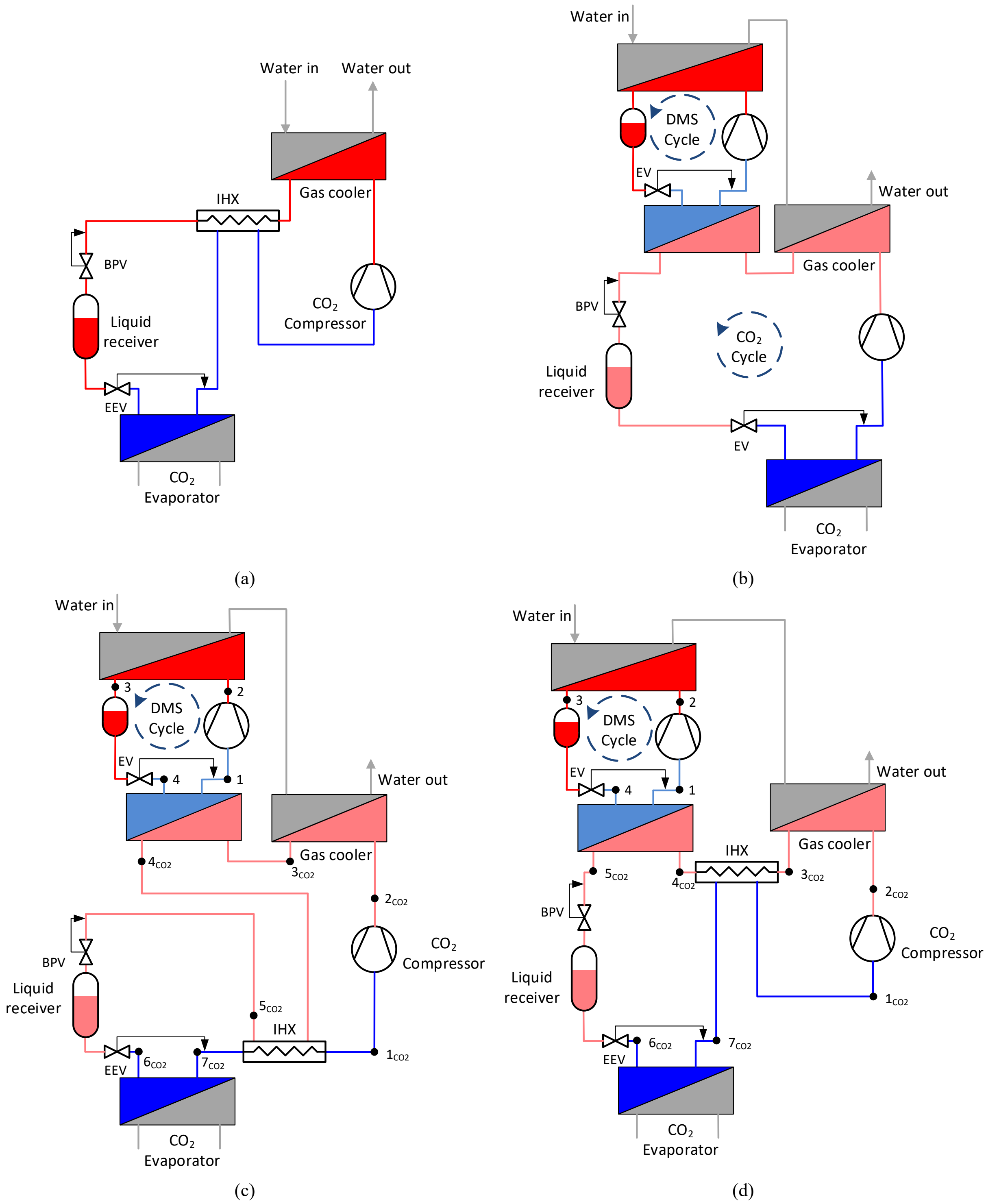

Figure 1. CO_2-DMS configurations analyzed for hot water generation. (a) CO_2 with IHX base cycle without DMS, (b) CO_2-DMS without IHX, (c) CO_2 cycle with DMS and IHX, (d) CO_2 cycle with IHX and DMS.

Table 1. Main components of the cycle and their characteristics.

	Equipment	Manufacturer	Model	Tech. info
CO_2 cycle	Compressor	Dorin	CD300H	$\dot{V}$ = 1.46 m³/h
	Evaporator	Swep	B8Tx26P	A=0.552 m²
	Gas cooler	Swep	B16x34P	A=1.31 m²
	IHX	Swep	B17x4P	A=0.082 m²
Both cycles	Subcooler/DMS evaporator	Swep	B18Hx20	A=0.738 m²
R1234yf cycle	Compressor	Copeland	YH04K1E	$\dot{V}$ = 5.76 m³/h
	Condenser	Swep	BX8THx16	A=0.322 m²

The compressor mass flow rate and power consumption are modeled using the correlations provided by the manufacturer, the expansion valve is modeled as isenthalpic, and the heat exchangers are modeled by deriving correlations for the evaporation/condensation pressure and heat transfer rate. In order to obtain those correlations, two different approaches have been used. Since the condenser is a conventional subcritical water/refrigerant plate heat exchanger, a well-known commercial software (IMST-ART) has been used to obtain condensation pressure and heat transfer rate. On the other hand, as the evaporator is a transcritical CO_2, subcritical R1234yf heat exchanger, nor IMST-ART or other commercial codes can be used, and a cell-by-cell discretization model developed in MATLAB has been used.

The procedure used to obtain the correlations of the evaporation pressure, condensation pressure and heat transfer rates were similar in both cases. First, it was defined a matrix of 16800 different input data that correspond to the combination of different realistic conditions of the CO_2 heat-pump experimental rig of the lab of the Research Group of Modeling of Thermal and Energy System at UPCT. To obtain these combinations it was considered different inlet temperatures of the water at the evaporator of the CO_2 cycle (between 10 and 30 ºC) and to the CO_2 gas-cooler (between 30 and 55 ºC) with different temperature lifts at those heat exchangers as well as at the condenser of the R1234yf cycle. Besides different superheating and subcooling degrees were considered to reach a total of 16800 different combinations that could be repeated experimentally at the experimental rig of the lab. As said before, these input data fed the MATLAB and IMST-ART models to obtain the correlations of the evaporation pressure, condensation pressure, and heat transfer rates.

Once all the components of the DMS system were modeled, they were joined in a model for the global CO_2- DMS cycle built up in MATLAB. In this case, a total of 5850 different input conditions were considered. They were obtained from combining different water flow rates (between 800 and 1600 kg/h) and inlet water temperatures at the evaporator (between 5 and 25 ºC) and outlet water temperature at the gas cooler (between 20 and 60 ºC), as well as different gas cooler pressure levels (between 70 and 120 bar). A superheat of 5 K was imposed in both evaporators and no subcooling was set at the R1234yf condenser.

3. RESULTS AND DISCUSSION

Figure 2 shows the evolution of the global COP of the system as a function of the gas cooler pressure for all the configurations studied, different heated water inlet/outlet temperatures, and a fixed water inlet temperature of 15 °C at the evaporator. Figure 2a shows the results for a heated water inlet/outlet temperature of 40/45 °C, which corresponds to intermediate temperature heating mode for a water-to-water heat pump according to EN- 14511-2 standard. Figure 2b shows the results for a heated water inlet/outlet temperature of 10/60 °C, which tries to simulate the first stage of the operation of a domestic hot water generation heat pump, whereas Figures 2c, and 2d show the results for the same outlet temperature and increasing inlet temperature, which tries to simulate later stages.

According to the results obtained, the base CO_2 transcritical cycle with IHX and without DMS, denoted as "Only IHX", performs always better, whereas the cycle with DMS and without IHX, denoted as "Only DMS", shows in most cases the worse efficiency. According to Figure 1, two different options have been studied for the CO_2-DMS cycle with IHX, which correspond to situate the IHX before the DMS evaporator (Figure 1c, denoted as "DMS-IHX" in Figure 2), or after the DMS evaporator (Figure 1d, denoted as "IHX-DMS" in Figure 2). Two different possibilities have been considered for this last two configurations, "(1GC-2COND)" in which the water flows first through the gas cooler and then through the DMS cycle condenser, and "(1COND-2GC)" in which the water flows first through the DMS cycle condenser and then through the gas cooler. The "IHX-DMS (1COND-2GC)" option seems to perform always better than all the other CO_2-DMS options. The only benefit obtained using a DMS cycle for the conditions studied is that the optimal COP is obtained at lower gas cooler pressure compared to the base transcritical CO_2 without DMS.

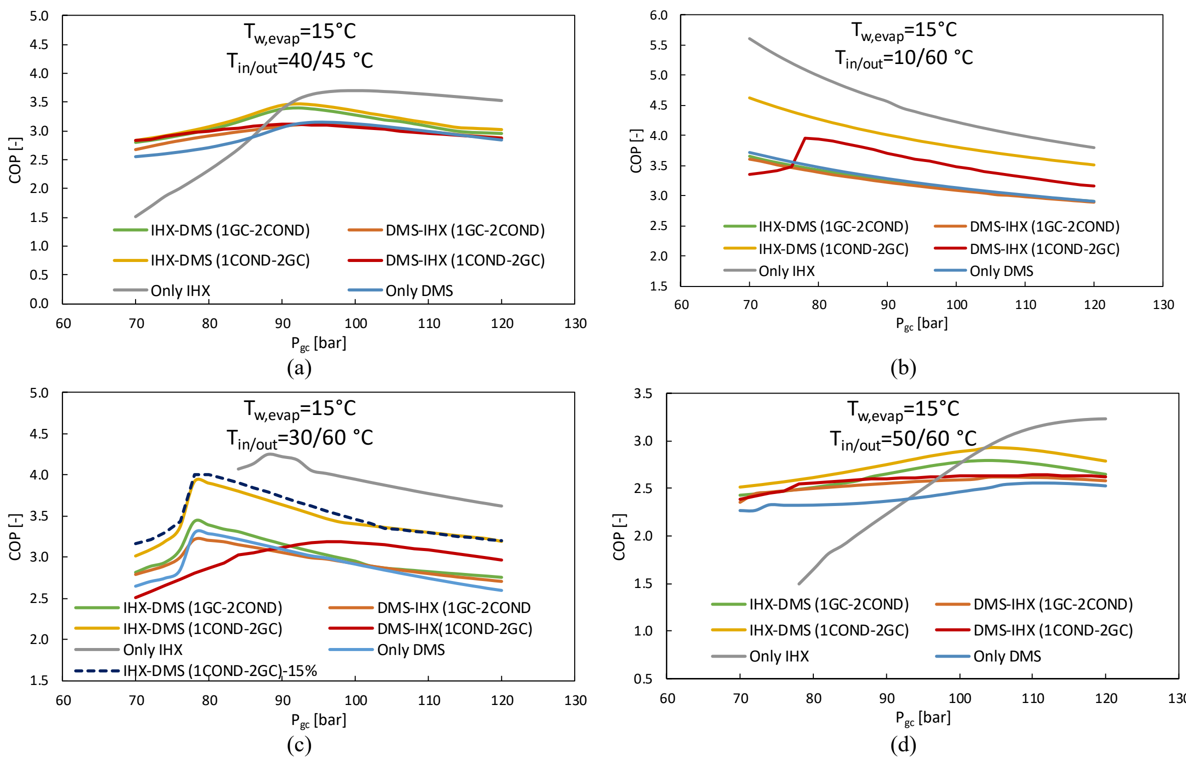

Figure 2. Global COP of the six configurations considered, different heated water inlet/outlet temperatures, and a fixed water inlet temperature of 15 °C at the evaporator.

Just as a first attempt to perform a very simplistic sensitivity analysis on the influence of the DMS components over the overall efficiency, Figure 2c includes extra results, denoted as "IHX-DMS (1COND-2GC)-15%", (black dashed line) obtained assuming that the capacity of all components of the DMS cycle is reduced in a 15 %. As a first approach, these results have been obtained using a multiplying factor of 0.85 for the compressor mass flow rate and power input, and the enthalpy difference in both, the DMS condenser and evaporator. According to the results obtained, a decrease in the DMS cycle capacity improves the overall efficiency of the system, although it seems that effect is not enough to make this configuration more efficient than the base CO_2 transcritical cycle with IHX and without DMS.

Regarding the influence of the inlet water temperature, Figure 3 shows the results obtained for the best (only IHX without DMS) and the worse (only DMS without IHX) options. As expected, as the evaporator water inlet temperature increases the global COP increases. Similar results have been obtained for other configurations and other heated water inlet/outlet temperatures.

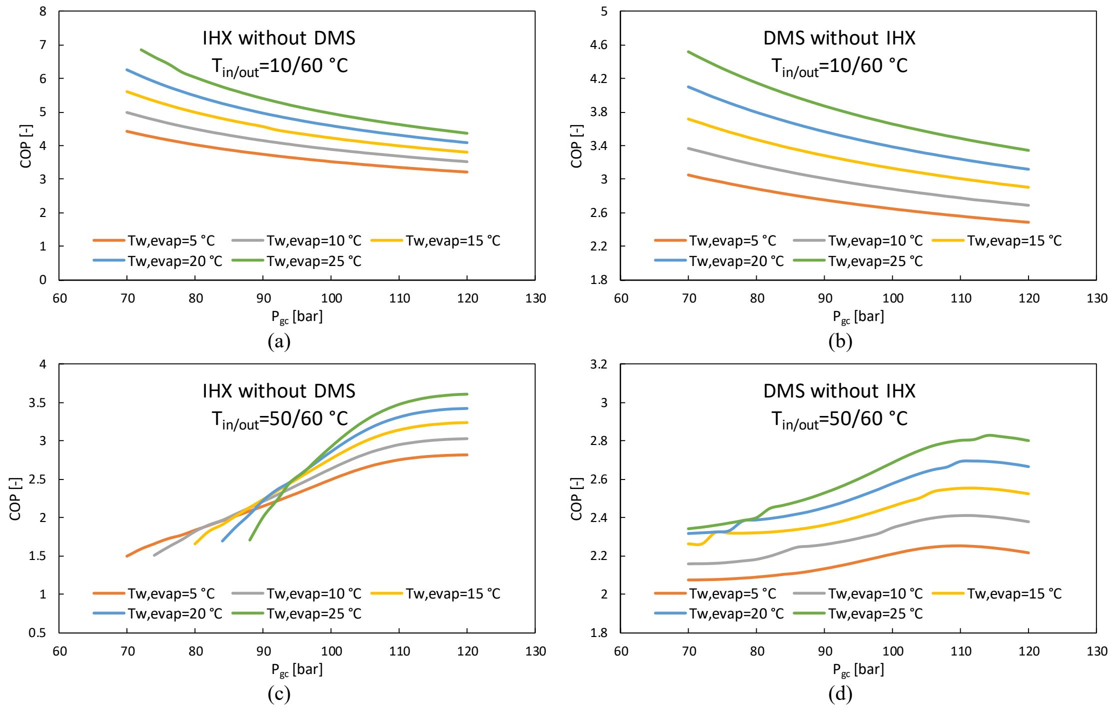

Figure 3. Influence of the evaporator water inlet temperature.

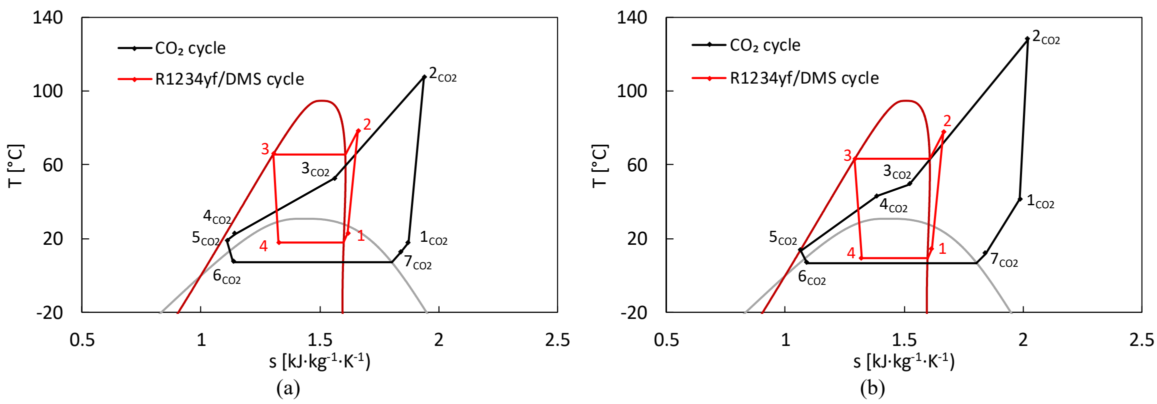

Figure 4. Temperature-entropy diagram for a CO_2-DMS cycle for an evaporator water inlet temperature of 15 °C, a heated water inlet/outlet temperature of 50/60 °C and two different configurations: (a) DMS evaporator/CO_2 subcooler located before the IHX; (b) DMS evaporator/CO_2 subcooler located after the IHX.

Finally, Figure 4 tries to explain the influence that the relative position of the IHX and the subcooler/DMS evaporator have on the performance of the system. Figure 4a shows the temperature-entropy diagram for the "DMS-IHX (1COND-2GC)" configuration and Figure 4b shows the same diagram for the "IHX-DMS (1COND-2GC)" configuration. The numeration of the points corresponds to Figure 1c and 1d. The figure shows that, when de DMS evaporator is located before the IHX, the temperature of the CO_2 entering the DMS evaporator is higher and thus the evaporation temperature increases (17.7 °C for the DMS-IHX configuration instead of 14.2 °C for the IHX-DMS configuration). Even though there is also an increase in the condensation temperature (65.5 °C instead of 63 °C) the COP of the DMS cycle increases (3.37 instead of 3). On the other hand, since the COP of the DMS cycle is higher for the DMS-IHX configuration, the water leaves the condenser and enters the gas cooler at a higher temperature and thus, the optimal CO_2 compressor discharge pressure increases (106 bar instead of 104 bar). Additionally, due to the subcooling effect that takes place in the DMS evaporator, the temperature of the CO_2

entering the high pressure side of the IHX is clearly lower and, therefore the temperature of the CO_2 leaving the low pressure side of the IHX (and entering the compressor) is clearly lower. For this reason the compressor discharge temperature decreases (107.5 °C instead of 127.8 °C) and, as a result, the CO_2 cycle COP decreases (2.01 instead of 2.86). Since the decrease of the CO_2 cycle COP has a stronger impact in the overall efficiency of the system, the global COP for the IHX-DMS configuration is lower than for the DMS-IHX configuration (2.63 instead of 2.92).

4. CONCLUSIONS

This work explores different configurations of CO_2 transcritical heat pump cycles coupled to IHX and DMS for hot water generation. A MATLAB model was developed and tested with different operating conditions obtained by the simulation of more than 5800 different inlet conditions. The main results are the following:

- For all conditions simulated the configuration with CO_2 cycle with IHX and without DMS shows the highest COP.
- Although the use of a DMS cycle allows to decrease the optimal pressure of the CO_2 cycle, the highest cost and complexity and the lower efficiency clearly advise against its use for the conditions studied.
- Further efforts are necessary to clarify the influence that the sizing of the DMS cycle components has on the overall efficiency of the system.

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