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Efficiency Enhancement by Subcooling the Carbon Dioxide Process with Water as Refrigerant

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ABSTRACT

As a consequence of the F-Gas regulation R404A is no longer an option for commercial refrigeration applications. Therefore, this paper focuses on natural refrigerants. There are a few options like carbon dioxide, which has an efficiency loss with increasing ambient temperatures. A promising option is subcooling of the carbon dioxide process with a chiller using water as the refrigerant. This will result in a new optimized high pressure of the carbon dioxide process depending on the ambient temperature. Finally the annual COP values of the standard transcritical and subcooled system will be discussed.

Keywords: Refrigeration, Water, Carbon Dioxide Cascade, Subcooling, Vapor Compression, Chiller, Energy Efficiency

1. INTRODUCTION

The F-Gas Regulation, which came into force on January 1st, 2015, envisages an EU-wide phase down of the CO₂ equivalent of hydrofluorocarbons (HFCs) by 79 % by 2030 compared to a reference value based on the annual average of the quantities of hydrofluorocarbons a producer or importer reported to have placed in the market between 2009 and 2012. From January 1st, 2020, the next step will be a ban on the placing on the market of refrigeration appliances for commercial use with HFCs with GWP > 2500 (European Commission, 2014). As a result, the refrigerant mixture R404A may no longer be used in newly installed systems from this point in time.

Investigations on the state of the art of CO₂ refrigeration systems have shown that this technology has an energetic advantage over direct expansion R404A systems despite the transcritical operation at high ambient temperatures (Gullo *et al.*, 2017).

As a possible alternative, in the small power range CO₂ (R744) systems are available in cascade connection. Here, the upper stage of the cascade has to absorb the complete condensing capacity of the CO₂ process. This means that with larger cooling capacities not only the CO₂ system but also the upper stage accordingly must be dimensioned correspondingly large. The advantage of this combination is that the CO₂ process can be operated subcritical regardless of the environment temperature. The effectiveness and the limits of use have been investigated by Bagarella *et al.* (2016).

Another alternative is the subcooling of a transcritical CO₂ process by means of mechanical subcooling. In this variant, the CO₂ process is followed directly after the gas cooler by a heat exchanger, which subsequently subcools the transcritical gas. The achievable increases in efficiency and performance were examined in detail by Llopis *et al.* (2015a), Dai *et al.* (2017) and Pottker and Hrnjak (2015) for different refrigerants, but without the refrigerant water.

In this paper, such a structure for the subcooling of transcritical CO₂ by means of mechanical subcooling with the refrigerant water (R718) is theoretically investigated. The advantages of R718, the increase in efficiency and the limits of application are presented.

2. THE INVESTIGATED SYSTEM AND USED REFRIGERANT

2.1 Water as refrigerant

R718 has a GWP and ODP of "0" each and is neither flammable nor toxic. When used in a centrifugal compressor refrigeration system, it occurs both in the liquid and in the gaseous state. The thermodynamic process takes place due to the vapor pressure curve of water in a rough vacuum, but then corresponds to the cycle of conventional refrigeration systems. Furthermore, the use of water in compression refrigeration systems with temperatures below 0 °C is usually avoided. In addition to the necessary operation in a rough vacuum, water has a low volumetric cooling capacity and requires higher pressure ratios for a given temperature spread than conventional refrigerants. These points require an implementation of the thermodynamic cycle with minimal losses (Hanslik and Suess, 2017), (Suess, 2016).

2.2 The investigated system

Figure 1 shows the schematic structure of the investigated system. It is a single-stage CO₂ cycle, the "Refrigeration Cycle" and a subcooling circuit "Mechanical Subcooling" downstream of the gas cooler. The aim is to further cool the transcritical CO₂ leaving the gas cooler by means of a compression refrigeration system. The refrigerant in this subcooling circuit is R718. The interface between the two circuits is a finned tube heat exchanger, which is traversed by CO₂ inside and is surrounded by circulating water in a vacuum atmosphere. The energy required for the evaporation of the water is taken from the CO₂ gas, thereby cooling it. The resulting water vapor is compressed by means of a centrifugal compressor and fed into the condenser. There, the steam releases its energy to another finned tube heat exchanger to an additional cooling circuit and condenses completely. The circuit is then closed by a self-regulating, pressure loss-free expansion device. The expansion device used in combination with the continuously variable centrifugal compressor allows a continuous adjustment of the delivered volume flow, and the pressure ratio between the pressure and suction side of the compressor from the ratio of "1".

The additional cooling circuit in the considered system consists of the heat exchanger, a circulation pump and a dry cooler. As a working medium, a glycol / water mixture is usually used. This extra circuit is needed because there are no commercially available air condensers for R718. The problem is the existing density ratio of > 10,000, at a temperature of 50 °C increasingly with decreasing water or steam temperature. Both systems, gas coolers and dry coolers, transfer their waste heat to the same heat sink, the environment.

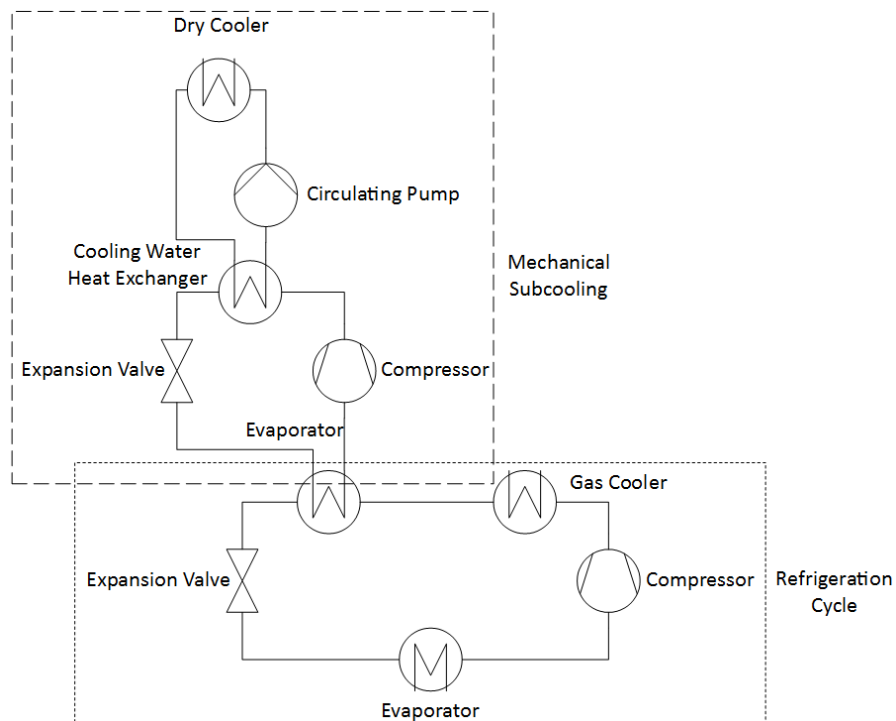


Figure 1: Schematic of the combined subcooling cycle

3. METHOD

For the evaluation of the system with and without mechanical subcooling, the assumptions given in Table 1 were used as the basis for the calculations. The physical properties of the refrigerants used for the respective cycles were generated with REFPROP (Lemmon *et al.*, 2013).

Table 1: Operating conditions

R744		
Superheating	10	K
Cooling capacity	150	kW
Evaporating temperature	-5 / -15	°C
$t_3 - t_{env}$	5	K
compressor efficiency (Llopis <i>et al.</i> , 2015b)	$0.95 - 0.1 * \pi$	-
R718		
$t_4 - t_{1*}$	5	K
compressor efficiency	0.7	-
$t_{3*} - t_{env}$	4	K
maximum compressor volume flow	1.2	$m^3 s^{-1}$

For the calculation of the individual COP values, the electrical power consumption of the circulation pump in the external cooling water circuit of the R718 chiller as well as the fans of the gas cooler and the dry cooler were neglected. Only the specific capacities were considered. Equation (1) shows the general calculation of COP, which is also used for the determination of pure transcritical operation. q_0 corresponds to the specific cooling capacity and w_c to the required specific compressor work of the refrigeration cycle.

$$COP = \frac{q_0}{w_c} \quad (1)$$

Equation (2) shows the calculation of the specific cooling capacity and Equation (3) the specific subcooling capacity of the CO₂ cycle. Equation (4) shows the specific cooling capacity of the R718 circuit. h_0 and h_5 correspond to the specific enthalpy at the outlet of the evaporator, or after the throttle, h_3 and h_4 of the specific enthalpy at the outlet of the gas cooler or after subcooling in the CO₂ cycle and h_{1*} and h_{4*} of the specific enthalpy in the evaporator or after relaxing in the R718 circle.

$$q_{0,R744} = h_0 - h_5 \quad (2)$$

$$q_{sub} = h_3 - h_4 \quad (3)$$

$$q_{0,R718} = h_{1*} - h_{4*} \quad (4)$$

The energy balance in the subcooler is shown in Equation (5) and Equation (6) shows the relation of the occurring mass flows.

$$\dot{m}_{R744} * q_{sub} = \dot{m}_{R718} * q_{0,R718} \quad (5)$$

$$\dot{m}_{R718} = \frac{\dot{m}_{R744} * q_{sub}}{q_{0,R718}} \quad (6)$$

The specific compressor work of the two single-stage systems is shown in Equation (7) for the R744 cycle and in Equation (8) for the R718 process. h_1 and h_{1*} represent the specific enthalpy at the compressor inlet, h_2 and h_{2*} the isentropic specific enthalpy at the compressor outlet. $\eta_{i,R744}$ and $\eta_{i,R718}$ are the isentropic efficiencies of the respective compressors.

$$w_{c,R744} = \frac{h_{2,s} - h_1}{\eta_{i,R744}} \quad (7)$$

$$w_{c,R718} = \frac{h_{2^*,s} - h_{1^*}}{\eta_{i,R718}} \quad (8)$$

Based on Equation (1), the COP of the entire system is calculated in subcooling mode according to Equation (9)

$$COP^* = \frac{\dot{m}_{R744} * q_{0,R744}}{\dot{m}_{R744} * w_{c,R744} + \dot{m}_{R718} * w_{c,R718}} = \frac{q_{0,R744}}{w_{c,R744} + \frac{q_{sub}}{q_{0,R718}} * w_{c,R718}} \quad (9)$$

The individual states of the respective circuits are shown in Figure 2.

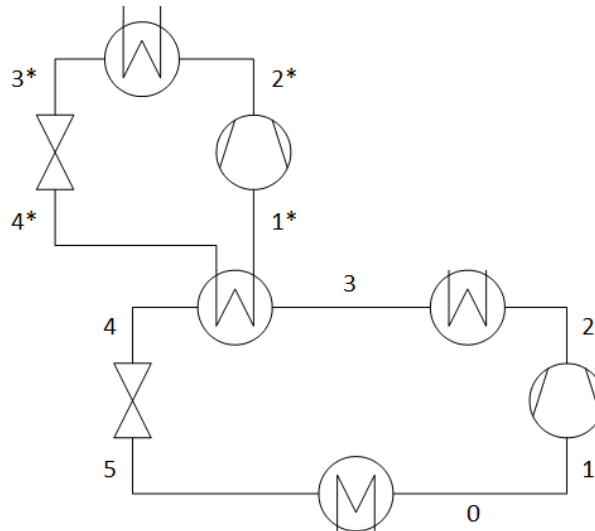


Figure 2: Schematic with state points

4. RESULTS

4.1 Optimum Operating conditions

Figure 3 shows the optimal high pressures of the transcritical CO₂ system, for a) for $t_0 = -5$ °C and for b) for $t_0 = -15$ °C, for different ambient temperatures. The respective dashed lines represent the interpolated connections between the individual maximum points. The individual marked values have been determined by means of a self-developed simulation. From each of the two diagrams, two curves at the ambient temperatures $t_{env} = 30$ °C and $t_{env} = 45$ °C are considered in more detail and the optimal pressures for operation with a subcooling of -2.5 K, -5 K and -7.5 K are shown. Diagram c) shows the values for $t_0 = -5$ °C and d) shows the values $t_0 = -15$ °C. In both diagrams, it can be seen that the optimum pressure drops as expected with increasing subcooling value. For c) and d), the optimum pressures at $t_{env} = 30$ °C and a subcooling of 7.5 K at 74 bar and at d) are only slightly higher when cooled by 5 K. This is followed by an increase in efficiency with subsequent increase in pressure, followed by a rise to a turning point. From this, the efficiency of the system continues to fall with further increases in process pressure. These inflection points are referred to in the diagrams as *optimized optimal pressure* and are preferable to the maximum efficiency points, since the efficiency values are only slightly lower and there are advantages for selecting the compressor for the subcooling stage. This can be explained by the p-h diagram shown in Figure 4.

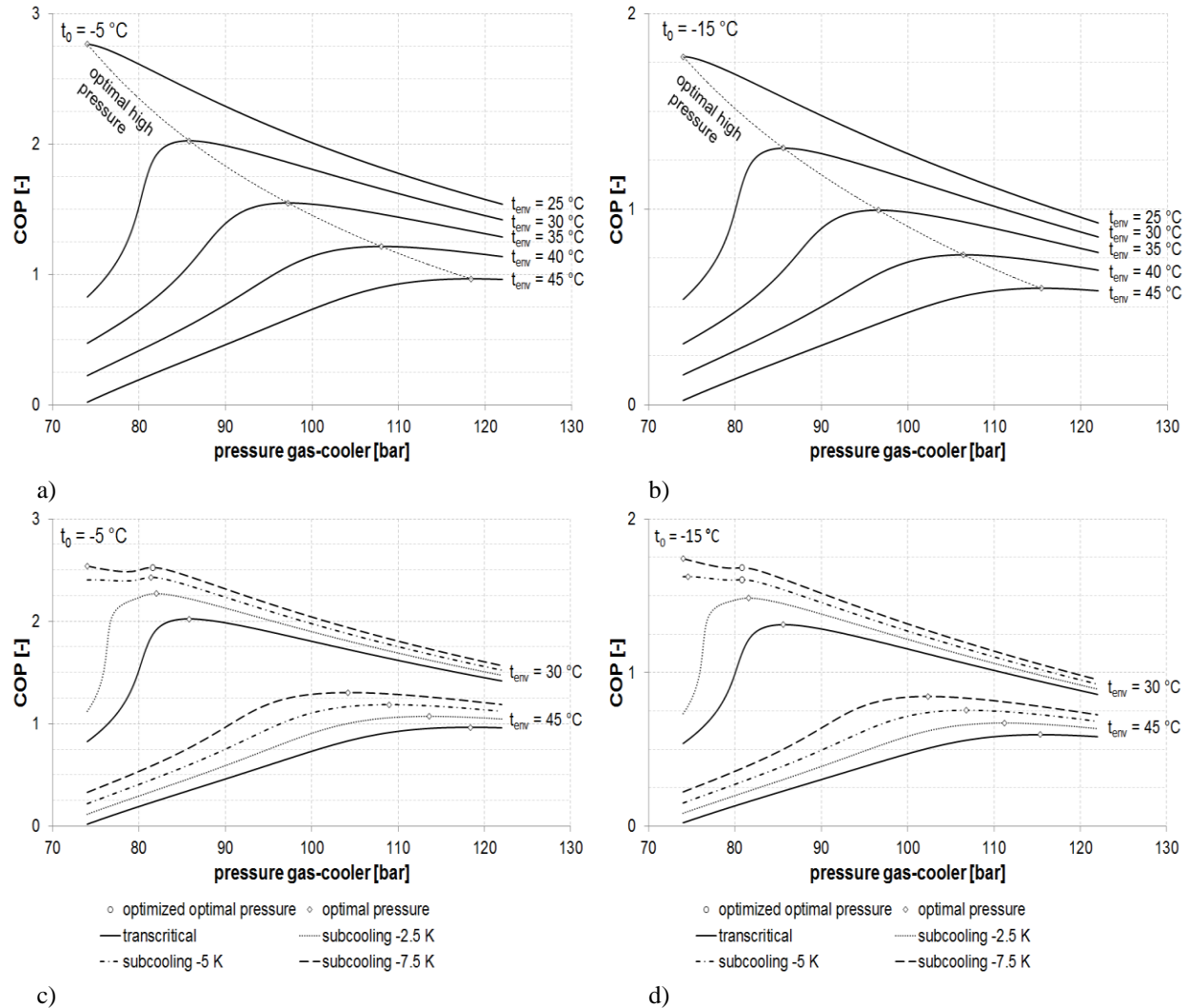


Figure 3: optimal high pressure for the transcritical CO₂ cycle with a) $t_0 = -5^\circ\text{C}$ and b) $t_0 = -15^\circ\text{C}$; optimal and optimized pressure with subcooling for c) $t_0 = -5^\circ\text{C}$ and d) $t_0 = -15^\circ\text{C}$

The three illustrated cycles each show the transcritical CO₂ cycle for the operating point $t_0 = -5^\circ\text{C}$ and $t_3 = 35^\circ\text{C}$. The solid line with the triangle symbols at the respective state points represents the pure transcritical cycle without mechanical subcooling at optimum high pressure. The dotted line with the circle symbols represents the transcritical cycle with a subcooling of 7.5 K at optimum high pressure (74 bars) and the dashed line with the rhombuses represents the transcritical cycle with subcooling at the optimized optimum pressure. The points 3 and 4 for the compared subcooling cycles are each on the same isotherms and represent at 3 the temperature at the gas cooler outlet and at 4 the temperature after the subcooling. Provided that the same cooling capacity is required for both systems, both systems need approximately the same mass flow of CO₂. If one compares the enthalpy difference q_{sub} with optimal and optimized optimal pressure, it clearly shows that the required subcooling performance at optimum pressure is more than a factor of 2 higher than at optimally optimized pressure. This would also result in a larger sizing of the R718 chiller.

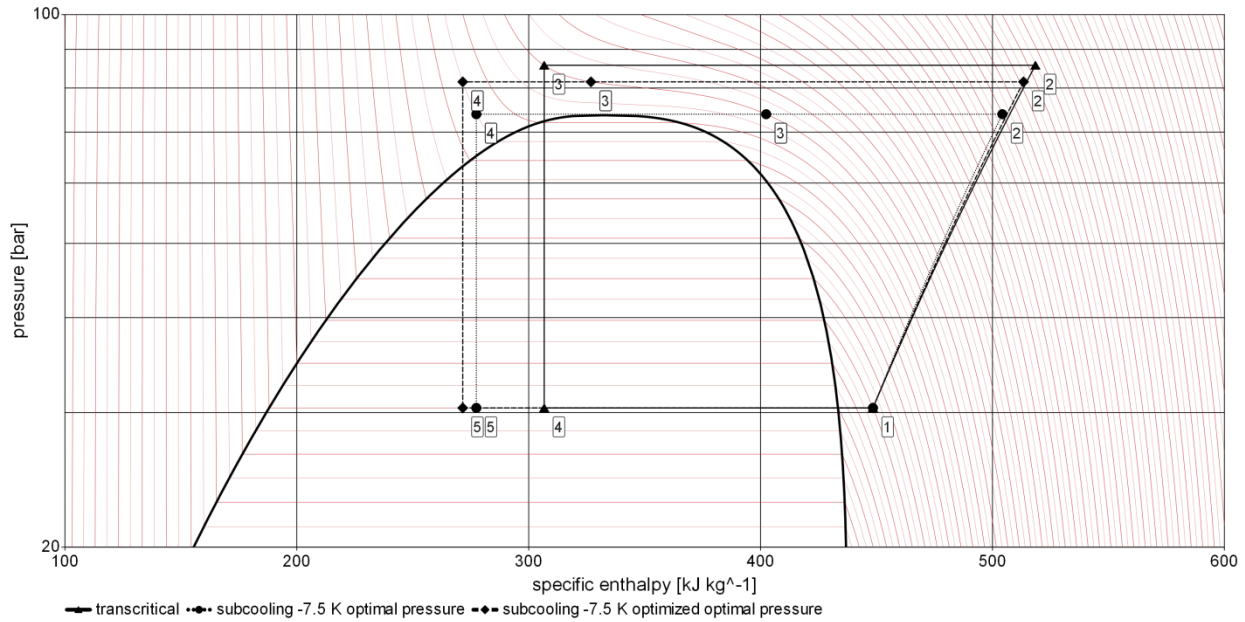


Figure 4: p-h diagram of the transcritical CO₂ cycle w and w/o mechanical subcooling ($t_0 = -5\text{ }^\circ\text{C}$, $t_3 = 35\text{ }^\circ\text{C}$)

4.2 Efficiency increase

Based on the optimum or optimized optimum operating pressures, Figure 5 shows the COP curves for pure transcritical operation and for transcritical operation with mechanical subcooling as a function of the environment temperature. Diagram a) refers to $t_0 = -5\text{ }^\circ\text{C}$ and diagram b) to $t_0 = -15\text{ }^\circ\text{C}$. Furthermore, with the respective secondary axis, the efficiency increase between the purely transcritical operation and the operation with a subcooling of 7.5 K is shown. When comparing the two curves, it is noticeable that there is a dependency on the evaporation temperature and the ambient temperature. With decreasing evaporation temperature, as well as with increasing ambient temperature, the percentage increase in efficiency increases. Furthermore, it can be seen that the increase from an ambient temperature of $t_{env} = 35\text{ }^\circ\text{C}$ is significantly lower and seems to approach asymptotically to a maximum limit.

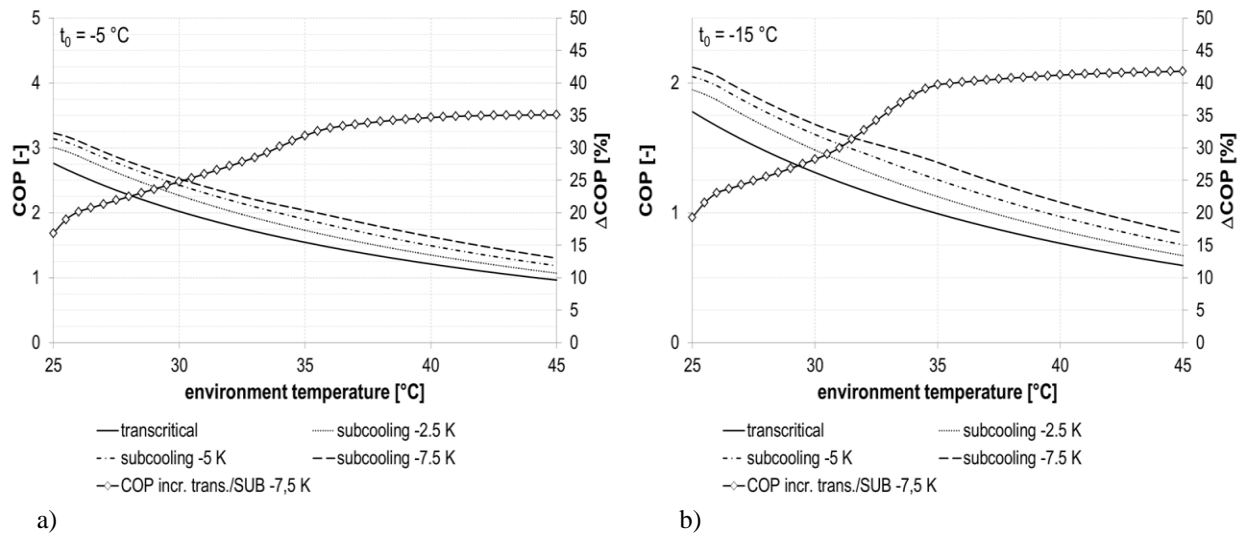


Figure 5: COP of the transcritical CO₂ cycle w and w/o subcooling a) $t_0 = -5\text{ }^\circ\text{C}$, b) $t_0 = -15\text{ }^\circ\text{C}$

4.3 Required subcooling capacity

In the following, the required subcooling capacities (SUB) are shown in Figure 6 with the solid lines and the maximum possible cooling capacity (Q_0) of the R718 circuit for the three indicated subcooling temperatures is shown by the dashed lines. Diagram a) refers to $t_0 = -5\text{ }^\circ\text{C}$ and diagram b) to $t_0 = -15\text{ }^\circ\text{C}$. The optimized optimum pressure was used as the basis for the calculation.

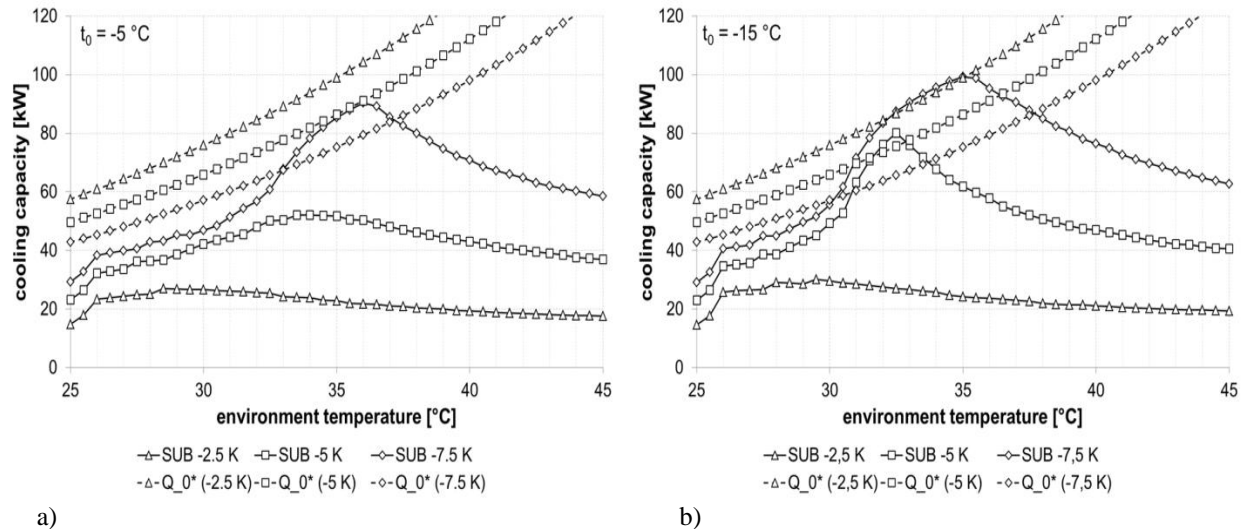


Figure 6: required subcooling capacity and possible cooling capacity

It is easy to see that for both evaporating temperatures, with a small exception at $t_0 = -15\text{ }^\circ\text{C}$, with the single-stage R718 system, with the maximum volumetric flow given in Table 1, a subcooling of -5 K over the entire temperature range of the environment can be realized. Over a wide range, a subcooling value of $> 7.5\text{ K}$ is possible with the above setting. Again, for a subcooling of 7.5 K , as in Figure 5, a turning point in the curve at $t_{env} = 35\text{ }^\circ\text{C}$ can be seen. In addition, significantly larger subcooling temperatures are possible. Another point is the increasing possible cooling capacity with higher environment temperatures. This is related to the increase in the density of water vapor as the evaporation temperature increases.

In order to be able to subcool at least 7.5 K over the whole range of the environmental temperature, there are two possibilities for optimization. On the one hand, one could use an R718 compressor with a larger maximum flow rate, on the other hand, one could increase the process pressure at the outlet of the R744 compressor in order to reduce the required subcooling performance. Both options require further investigation to determine which of the two is more efficient. Furthermore, the combination of the two systems can still be examined to see what absolute subcooling over the entire environment temperature range for the two evaporation temperatures can be achieved.

5. CONCLUSION AND FUTURE WORK

The simulation of a transcritical CO_2 process with subsequent mechanical subcooling with a refrigeration system with the refrigerant R718 has shown that efficiency increases of more than 35% compared to purely transcritical operation can occur. The main influencing factors regarding the efficiency are on the one hand the evaporation and ambient temperatures, on the other hand the process pressure on the pressure side of the compressor. It has been noticed that in the course of the COP curve above the ambient temperature there are, in addition to the optimal process pressure, also points which have a positive effect on the entire system with a slight loss of efficiency.

For both investigated evaporation temperatures in the CO_2 cycle, a subcooling of 5 K is possible with the considered system with a small exception over the entire ambient temperature range. Over much of the ambient temperatures, significantly greater temperature differences are possible. In order to allow a subcooling of 7.5 K over the entire temperature range, further investigations have to be made, which on the one hand consider a larger compressor and on the other hand a further optimized process pressure.

NOMENCLATURE

COP coefficient of performance (—)

COP^*	overall coefficient of performance	(–)
h	specific enthalpie	(kJ kg ⁻¹)
\dot{m}	mass flow	(kg s ⁻¹)
q_0	specific cooling capacity	(kJ kg ⁻¹)
q_{sub}	specific subcooling capacity	(kJ kg ⁻¹)
t_0	evaporating temperature	(°C)
w_c	specific compression work	(kJ kg ⁻¹)

Greek symbols

η	compressor efficiency	(–)
π	pressure ratio	(–)

Subscript

0...5	condition point of the CO ₂ cycle
1*...4*	condition point of the R718 cycle
env	environment
R718	related to the R718 cycle
R744	related to the R744 cycle

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