

CO₂ transcritical refrigeration cycles: potential for exploiting waste heat recovery with variable operating conditions

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Abstract. An analysis of CO₂ transcritical inverse cycles is presented, for investigating the exploitation potential of the heat rejected from the gas cooler, usually considered as a not recoverable waste heat. The cycle is analyzed in the cooling configuration during the summer season, when there are contemporarily a high temperature of the heat sink, normally over 30 °C, and a high temperature waste heat at the gas cooler, possibly exceeding 100 °C. The latter is usually rejected to the environment, but some useful applications could be analyzed, where a not negligible fraction is somehow recovered. Here a second cooling machine, thermally activated, which adds further cooling capacity to the traditional vapor compression one, is considered. A thermodynamic analysis was performed, with CO₂ evaporation temperature varying between -5 °C and 5 °C, compressor discharge pressure between 74 bar and 150 bar, and gas cooler outlet temperature variable between 30 °C and 45 °C, investigating the influence of the main thermodynamic parameters on the cycle performance by means of the software REFPROP. The depicted model allowed performing a wide sensitivity analysis of all the above mentioned thermodynamic variables, identifying the best operating conditions of the carbon dioxide transcritical inverse cycle, in terms of COP index and waste heat exploitation.

1. Introduction

The use of carbon dioxide as operating fluid in vapour compression refrigeration systems is nowadays receiving a new attention, due to its environmental impact but also due to unexpectedly good performance.

Since the second half of 19th century, such systems were used, at least until 30-40-ties of the past century, when carbon dioxide was slowly replaced by specifically designed synthetic refrigerants; an exhaustive chronicle can be found in many previous studies [1-4]

Over the last decades, the CFC firstly, then the HCFC and finally the HFC refrigerants, which were once expected to be acceptable fluids, were placed in the list of regulated substances due to their impact on climate change, so that a rising interest has been originated in environmentally safe natural working fluids, just like CO₂ is. Carbon dioxide is a very interesting alternative since it is non-flammable, non-toxic (even if it is classified as an asphyxiant gas), inexpensive and widely available, and does not affect the global environment. Carbon dioxide has GWP = 1, but the net global warming impact when used as a technical gas is zero, since the gas is a by-product from industrial production. In addition, CO₂ has excellent thermo-physical properties, leading to good heat transfer, efficient compression and compact system design due to high volumetric capacity.

In the early applications, one of the main problems of CO₂ as a refrigerant fluid was its low critical temperature, but today it can be effectively overcome by operating the system in the transcritical region, with a discharge pressure higher than the critical, which is approximately 73.8 bar with $T_{cr} = 30.98$ °C.



A transcritical cycle differs from a conventional phase-change vapour compression cycle as for the heat rejection phase in the high temperature cycle section, which is not operated through a condenser, being replaced by a gas cooler, where the CO₂ temperature decreases over a large temperature range. The use of a gas cooler may offer some advantages over the constant temperature vapour condensation phase [1], especially if a heat recovery has to be accounted for. Indeed, when compared with other refrigerants, the CO₂ heat rejection phase may begin at higher temperature levels, possibly exceeding 100 °C. This paper deals with the feasibility of using such waste heat as powering source for a thermally activated cooling machine, as an absorption system, which adds further cooling capacity to the basic cooling cycle.

2. Basic cycle description

A basic cycle configuration is assumed, shown in Figure 1, without any layout complexity, just to fix a reference line which can be rested on for the analysis of more complex cycle arrangements. The analysed cycle is a transcritical CO₂ inverse cycle, thus with a subcritical low pressure and a supercritical high pressure. The evaporating carbon dioxide removes heat from the low temperature heat source, then, at the point 1 (complete vaporization) it is compressed up to the high pressure level, point 2, where the highest pressure and temperature are reached within the cycle. After the compression phase, it enters the gas cooler or any other system of waste heat exploitation, in an ideally isobaric process. The temperature at the exit of gas cooler, point 3, may be as low as the external environment cold sink enables. Then, by entering the throttling valve, the carbon dioxide pressure is reduced down to the low pressure level, where the point 4 states the inlet into the two-phase region, the liquid and the gaseous phases of CO₂ coexisting. The two-phase mixture, point 5, enters the evaporator, being vaporized until the initial conditions.

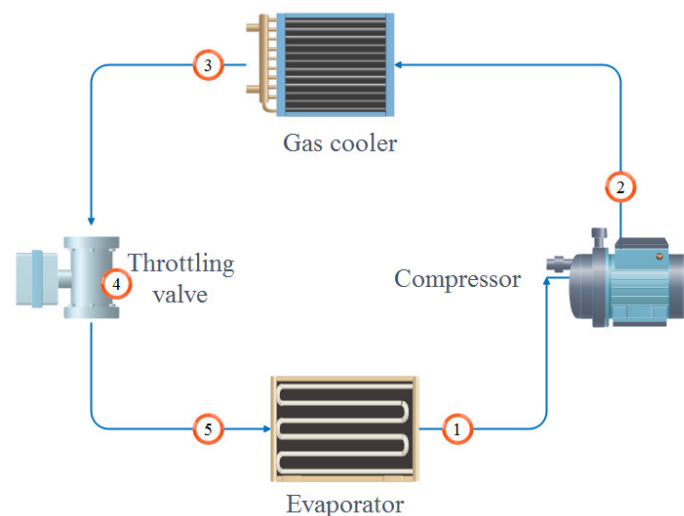


Figure 1. Basic layout of a transcritical CO₂ inverse cycle.

Starting from this basic layout configuration, some usual improvements of the cycle may concern the followings, although they will not be argument of the present study: a double stage compression phase, whose aim is to reduce the compressor power requirement, since the compression curve moves to near isothermal, reducing the power requirement too; expansion devices other than the throttle valve, as for example an ejector or a turbine, which would be able to recover some mechanical work from the expansion of CO₂; the internal heat regeneration, consisting of the heat transfer from the gas cooler outlet stream to the evaporated CO₂ before the latter enters the compressor, so that the quality at the evaporator inlet is lower and the cooling capacity is increased. The last issue would mean also a certain superheat degree at the compressor inlet, which would protect the compressor from unexpected and damaging drops last inside.

Figure 2 shows typical T-s and p-h diagrams for the examined cycle. Some assumptions are made for the analysis: each component is considered in a steady state condition, the kinetic and potential energy variations through it are neglected, as well as the heat and friction losses. As for the compressor isentropic efficiency, its value is set as constant and equal to 0.7, this being the most suitable from a technological point of view [5]. The throttling valve is considered as an isenthalpic device and the inlet conditions at the compressor are those of saturated vapour. Finally, the properties of CO₂ are obtained from REFPROP-NIST software, Ver. 9.0 [6]

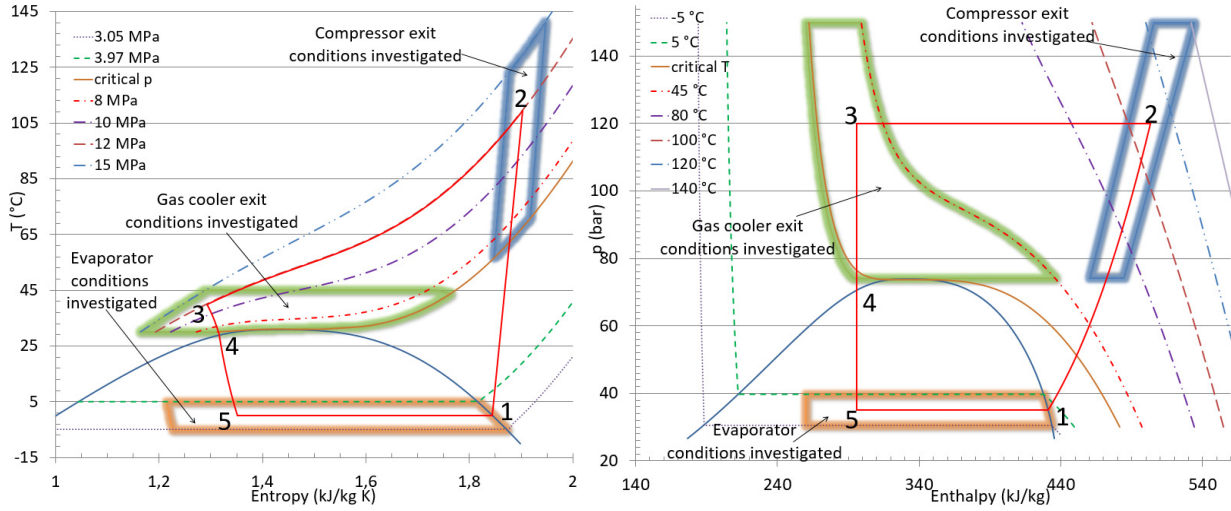


Figure 2. T-s and p-h reference diagrams for a transcritical CO₂ inverse cycle.

3. Cycle energy analysis

The Thermodynamics laws can be applied for each component of the simplified cycle, by means of the assumptions previously stated. Each component is considered as a control volume, to which the proper equations apply. This study would investigate, among the others, the effect of some thermodynamic parameters on the COP index, the coefficient of performance of the cycle. Its definition depends on the useful effect the cycle operates for; if a cooling machine is the aim, the COP index means the cooling capacity over the power requirement, i.e.

$$\text{COP} = \frac{\dot{q}_{\text{cool}}}{\dot{W}_{\text{net}}} \quad (1)$$

where \dot{q}_{cool} is the cooling capacity and \dot{W}_{net} is the mechanical power (usually provided as electric power) required to raise the pressure from the low level to the high value. In the simplest case, referring to the Figure 2

$$\dot{W}_{\text{net}} = \dot{W}_{\text{compr}} = \dot{m}(h_2 - h_1) \quad (2)$$

with \dot{m} the mass flow rate of CO₂ in the cycle loop and h the specific enthalpy of the fluid. The compressor isentropic efficiency $\eta_{\text{c, is}}$ has to be accounted for, which is defined as the ratio of the power required in an ideal reversible (adiabatic and isentropic) compression process to the actual power requirement of the component, as

$$\eta_{\text{c, is}} = \dot{W} / \dot{W}_{\text{c, is}} \quad (3)$$

where $\dot{W}_{\text{c, is}}$ is the power related to the ideal isentropic compression.

If an additional useful effect is considered, besides the cooling capacity, that is the heat recovery potential at the cycle high temperature level, a different index should be used, as some authors proposed [7-13], that is a system COP, named COP_s , defined as the sum of the cooling capacity and the heating capacity over the power requirement:

$$COP_s = \frac{\dot{q}_{cool} + \dot{q}_{heat}}{\dot{W}_{net}} \quad (4)$$

where \dot{q}_{heat} is the sensible heat available at the gas cooler. If all the available heat is recovered, then

$$\dot{q}_{heat} = \dot{m}(h_2 - h_3) \quad (5)$$

In the developed model, a set of equations is to be solved, which links all the thermodynamic states of the cycle. Once the high and the low pressures in the cycle are fixed, as well as the outlet temperature of the gas-cooler, the solving sequence is well defined, if the isentropic efficiency of the compressor is assumed. Table 1 depicts such solving sequence, starting from the compressor inlet conditions, point 1.

Table 1. Set of equations describing the thermodynamic states of the transcritical inverse CO₂ cycle.

State n.	Known variables	Link equation with the other states	Variables to be determined	Notes
1 – compressor inlet	p_1, T_1	$T_1 = T_{sat}(p_1)$	$h_1 = h_1(p_1)$ $s_1 = s_1(p_1)$	
2 _{is} – compressor outlet	$p_2, s_{2,is}$	$s_{2,is} = s_1$	$h_{2,is} = h_{2,is}(p_2, s_{2,is})$	Ideal state
2 – compressor outlet	$p_2, \eta_{is,c}$	$h_2 = h_1 + (h_{2,is} - h_1)/\eta_{is,c}$	$T_2 = T_2(p_2, h_2)$ $s_2 = s_2(p_2, h_2)$	Actual state
3 – gas cooler outlet	p_2, T_3	$p_3 = p_2$	$h_3 = h_3(p_3, T_3)$	
4 – two-phase region inlet	h_4	$h_4 = h_3$		
5 – evaporator inlet	p_1, T_1	$h_5 = h_4$	$h_{1,l} = h_{1,l}(p_1)$ $x_5 = (h_5 - h_{1,l})/(h_1 - h_{1,l})$	

A specific application is simulated, that is a cooling machine operating during the summer season, when there are contemporarily a high temperature level for the environmental heat sink, normally over 30 °C, and a high temperature waste heat at the gas cooler, which possibly exceeds 100 °C.

The influence of the following parameters is investigated: the temperature range for the evaporator is set between -5 °C and 5 °C, this being suited for the cooling effect in many applications for environmental air conditioning. As for the discharge pressure at the compressor outlet, its lower limit is set at 74 bar, just above the critical value, and the maximum considered is 150 bar, being currently a very high value from a technological point of view. Finally, the CO₂ temperature at the gas cooler outlet is assumed to vary from 30 °C up to 45 °C, depending of the external sink temperature and the heat exchanger effectiveness; the latter one was not analysed here more in depth. The analysis was performed with no superheat at the compressor inlet.

When dealing with the theoretical COP_s , ideally assuming that all the rejected heat is recovered, the model shows the existence of a maximum, depending on rejection pressure p_{gc} and the gas cooler outlet temperature $T_{out,gc}$; in this regard, an original view can be shown by plotting the COP_s index as iso-curves in a $(T_{out,gc}, p_{gc})$ graph for a fixed evaporator temperature. Figure 3 depicts $COP_s(T_{out,gc}, p_{gc})$ in the ranges investigated by the present study when the evaporator temperature is fixed at 5 °C and the compressor isentropic efficiency at 0.7. Such a diagram shows very concisely the influence of the two variables so that for each couple of values the COP_s is quickly determined. In addition, the iso- COP_s curves allow to identify where the COP_s may increase or decrease faster than elsewhere. In this regard, it can be noted that the highest COP_s values will be get with low pressures at the gas cooler, indeed a small increase of the gas cooler outlet temperature is more detrimental at lower gas cooler pressure than

at high values, since a sharply reduction does occur. In other terms, when the iso-COP_s curves are very close each other with respect to a direction, it means a high slope of the COP_s value in that direction. A dividing line of the region where the COP_s slope is very high can be posed as the dotted line in the diagram. It is calculated as the maximum gas cooler outlet temperature which allows a fixed COP_s to be obtained, corresponding also to the optimum gas cooler pressure for a prefixed gas cooler outlet temperature and to the maximum COP_s achievable.

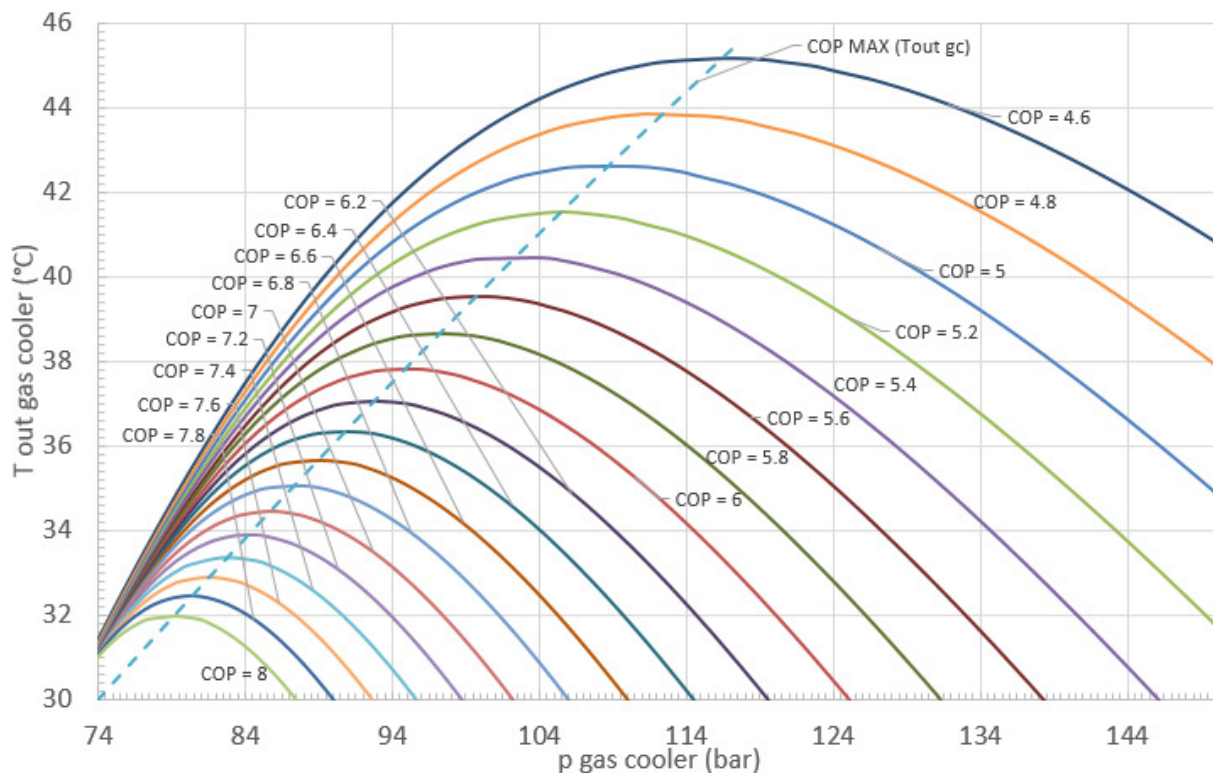


Figure 3. Iso-COP_s (system COP) diagram plot with evaporator temperature at 5 °C and $\eta_{is,c} = 0.7$.

Figure 4 shows the same quantity for an evaporator temperature of -5 °C. As it is fairly predictable, the COP_s is generally lower than that previously shown. For $p_{gc} > 85$ bar, the COP_s reduction is between 18% and 29%. Furthermore, for a fixed gas cooler outlet temperature the maximum COP_s is obtained at higher pressures when the evaporator temperature decreases from 5 °C to -5 °C.

Referring to the results shown in [7], they positively agree with the Figures 3 and 4, reminding the different assumptions as for the internal heat regeneration, supposed with a heat exchanger effectiveness of 60% in [7], while not considered at all in this study.

In addition to the system COP, another important parameter to be considered is the specific cooling and heating capacity, that is the sum of the enthalpy differences across the evaporator and the gas cooler. It is related to the size of the machines, since the cooling and heating capacity are given by the product of the mass flow rate and the corresponding unitary effects. In other terms, if a machine is considered with prefixed cooling and heating loads, a higher value of the total available enthalpy would mean a lower mass flow rate and consequently a lower size of the components. Figure 5 shows as the available total specific enthalpy depends on the gas cooler pressure and the gas cooler outlet temperature.

The total specific capacity increases for raising pressure and lowering gas cooler outlet temperature, however their influence is not linear. Indeed, in the left side of the diagram, a particular behavior of the curves may be noticed, which could be called a knee-curve, where a very steep increase of the available enthalpy occurs, more pronounced with pressure and temperature approaching the critical values. Such a trend, if a stable operation of the machine is the aim, would makes advisable to operate at pressure

higher than the knee, so that an unexpected reduction of the pressure or an increase of the temperature can be easily managed without excessively onerous mass flow rate adjustments.

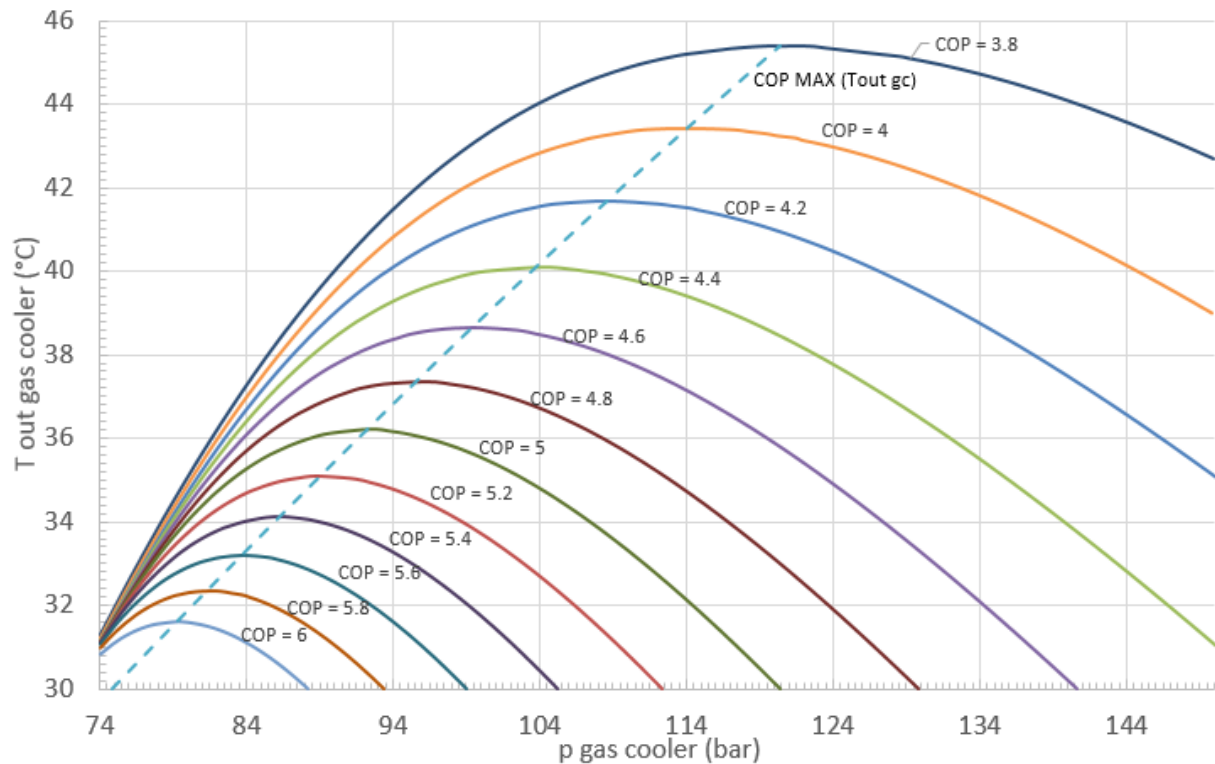


Figure 4. Iso-COP_s (system COP) diagram plot with evaporator temperature at -5 °C and $\eta_{is,c} = 0.7$.

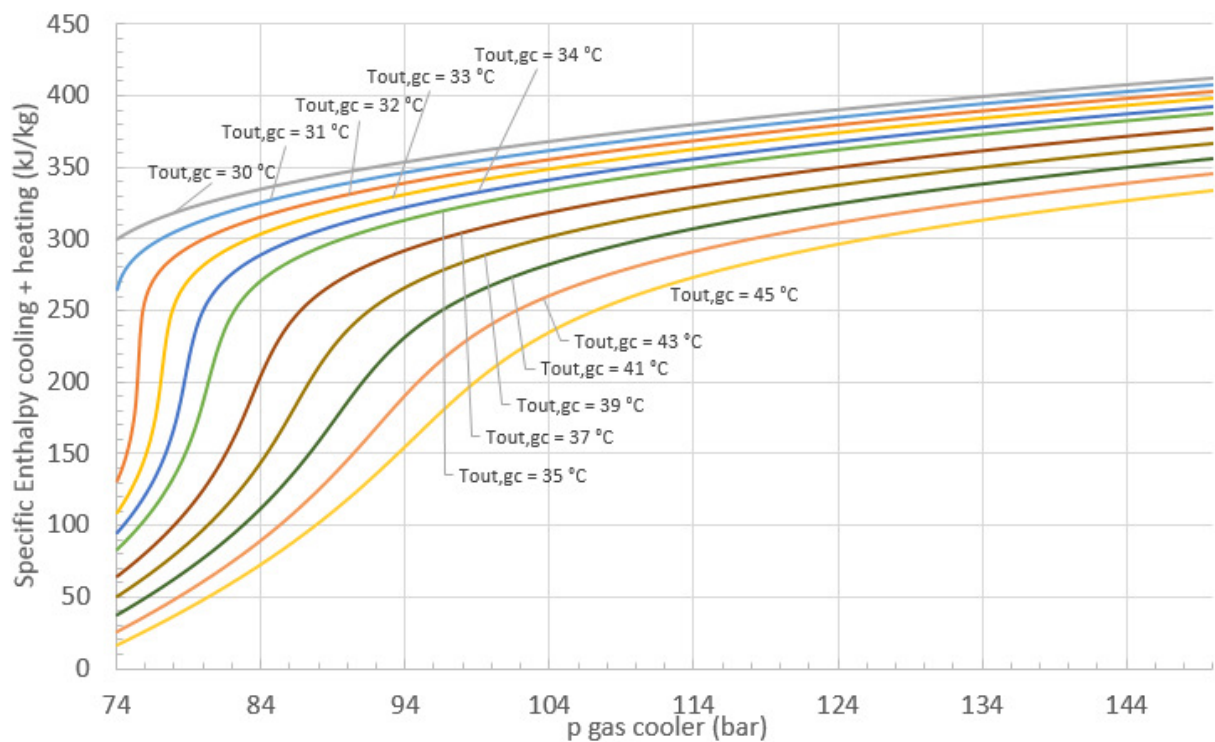


Figure 5. Available total specific enthalpy with $T_{eva} = 5$ °C and $\eta_{is,c} = 0.7$.

If the evaporation temperature is fixed at $-5\text{ }^{\circ}\text{C}$, a similar behavior will be noted, shown in Figure 6, with a generalized increase of the available total specific enthalpy, compared with the higher evaporation temperature level. In both cases, at temperatures equal or lower than the critical ($30\text{ }^{\circ}\text{C}$ and $31\text{ }^{\circ}\text{C}$ respectively, in the plot), the curves do not undergo such particular feature.

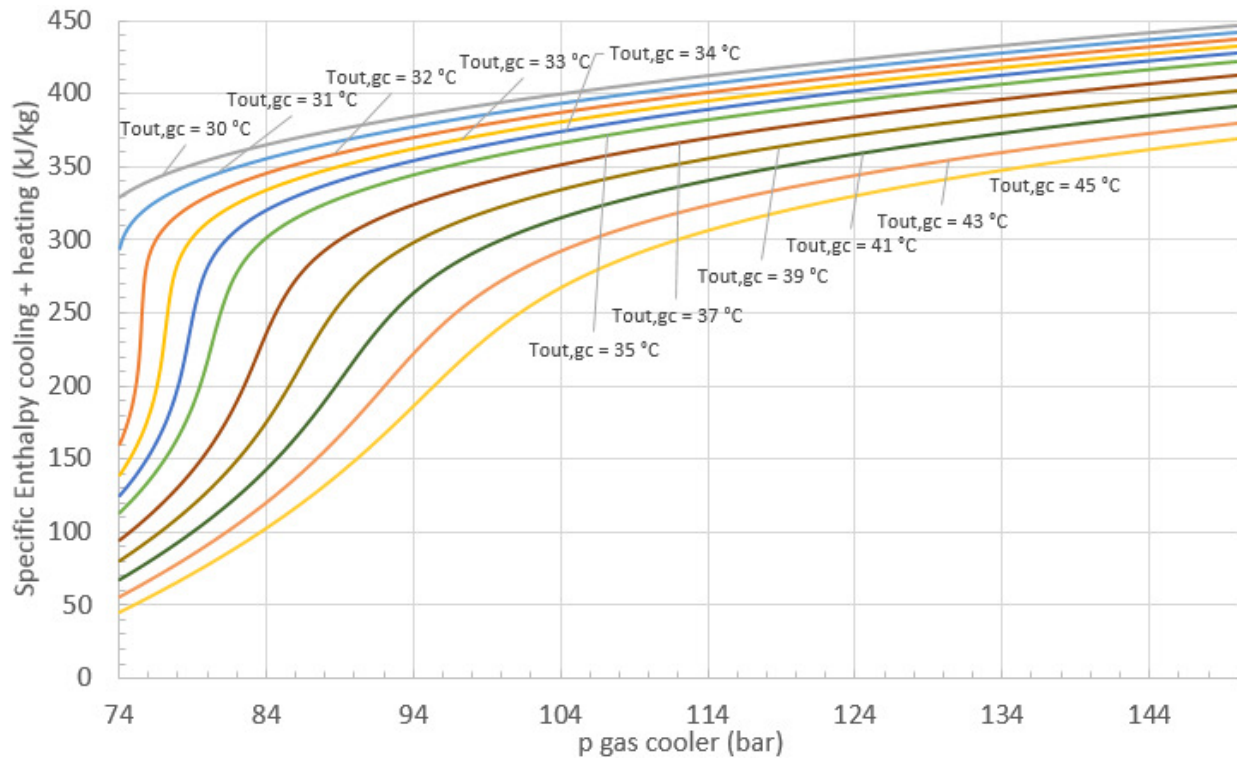


Figure 6. Available total specific enthalpy with $T_{\text{eva}} = -5\text{ }^{\circ}\text{C}$ and $\eta_{\text{is,c}} = 0.7$.

4. Additional cooling with absorption chiller coupling

The feasibility of an absorption chiller coupled to the main cooling machine, driven by the exhaust heat rejected at the gas cooler, may be analyzed with some simplifying and conservative assumptions, given the preliminary nature of the present study: the exhaust CO_2 stream is exploited down to $80\text{ }^{\circ}\text{C}$; the absorption chiller operates with a single effect mode; the COP of the absorption machine, defined as the cooling capacity over the heat power input, is set equal to a mean constant value of 0.6, as if it was independent from the feeding fluid temperature. The performance of such a combined system is evaluated by means of the percentage of cooling COP increase compared with the basic cycle, defined as the total cooling capacity (basic plus absorption cycle) over the compressor power required.

As Figure 7 shows, the additional cooling capacity provides a net positive contribution, which may be very attractive at low evaporation temperatures combined with a high temperature of the heat sink. Quantitatively, if the final heat rejection temperature is $45\text{ }^{\circ}\text{C}$ and the evaporation temperature is $-5\text{ }^{\circ}\text{C}$, a cooling COP increase greater than 30% may be accomplished. Furthermore, a specific trend is shown in the range between 82 and 90 bar, where the cooling COP increase reaches its maximum slope, so that a 20% COP increase can be obtained just at a gas cooler pressure equal to 88 bar. As for a better understanding of the exploitable potential, in the diagram the two investigated evaporation temperatures are depicted, along with the compressor outlet temperatures.

5. Heat recovery specific issues

Whichever the heat recovery system is, an essential tool to understand the relationship between the available heat and its exploitation potential is the T-h diagram, describing the heat recovery profiles.

Figure 8 shows the diagram for an evaporator temperature set at 5 °C and a compressor isentropic efficiency set at 0.7. When the abscissa is 0 kJ/kg, the temperature value indicates the CO₂ temperature at the compressor outlet, so that the influence of the evaporator temperature and the isentropic efficiency are already taken into account. Based on this issue, a minimum value of the high pressure can be evaluated, depending on the heat recovery application. For an absorption cycle heat recovery, as it is the present case, 80 °C is set as the minimum useful value for the heat source at the chiller generator, so that with these cycle parameters the high pressure should be set at least at about 96 bar. When the evaporation temperature is set at the lowest value among those investigated, i.e. -5 °C, better conditions result for the heat recovery capability, as Figure 9 shows, both for the enthalpy difference and for the temperature levels, which are higher. In this case, the minimum value for the high pressure should be fixed at about 82 bar.

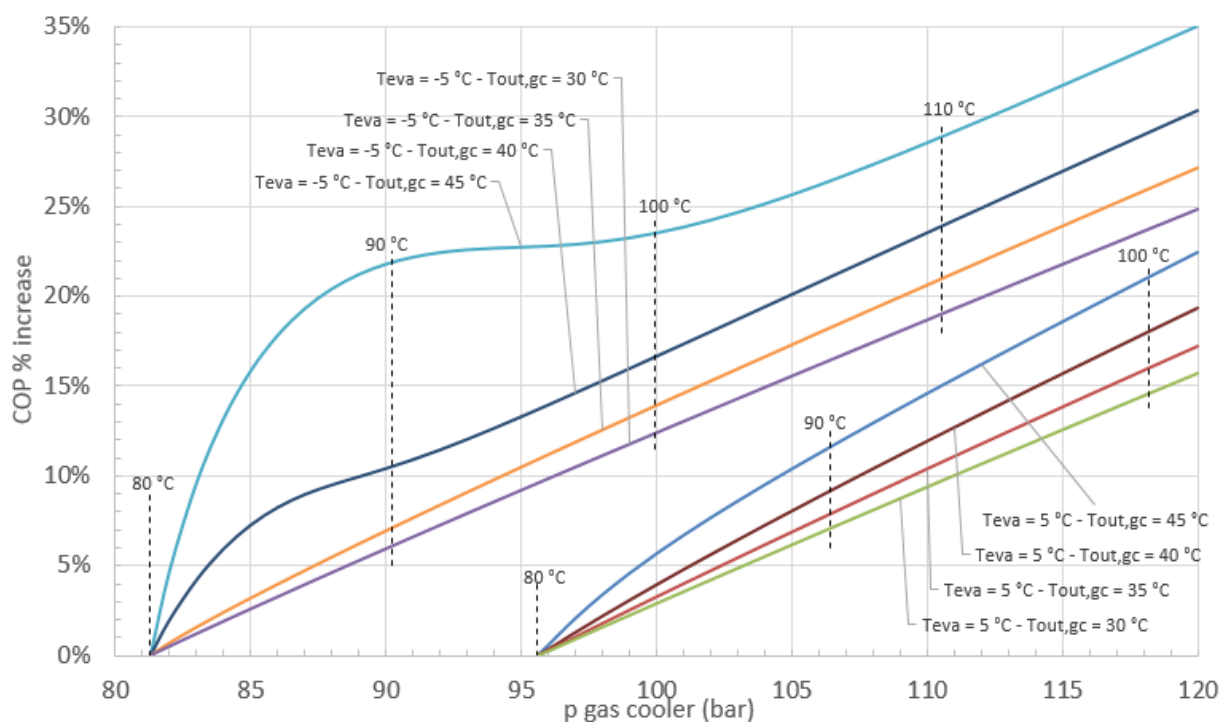


Figure 7. Cooling COP increase for a chiller absorption cycle integration; T out from chiller = 80 °C.

A fundamental issue when dealing with the heat recovery concerns the slope of the depicted curves, which is related to the inverse of the specific heat C_p . When a sharp increase occurs of C_p , especially for pressures close to the critical value, the fluid temperature tends to vary more and more slowly, with noticeable implications as for the heat transfer. The temperature profile of the fluid appointed to the heat recovery will be constrained by the CO₂ one, just as the Second Thermodynamics law states. If the overall waste heat is to be recovered, thus down to the environment temperature, and a rather constant specific heat fluid is used, as the water, its T-h profile would be a straight line that is able to successfully match the CO₂ curves only at the highest pressures. As a lower CO₂ pressure is considered, its behavior deviates more and more from the constant slope trend, so that a higher irreversibility would take place in the heat transfer because of the pinch-point requirements. A viable further option would be to consider many separated system, acting as multiple and variable heat sinks over partial temperature ranges, thus narrower than the entire gas cooler one [14-16]. Such a system would involve a better matching between the cooling and the heating streams even at low gas cooler pressures. On the other hand, it would mean an additional layout complexity, to be carefully evaluated.

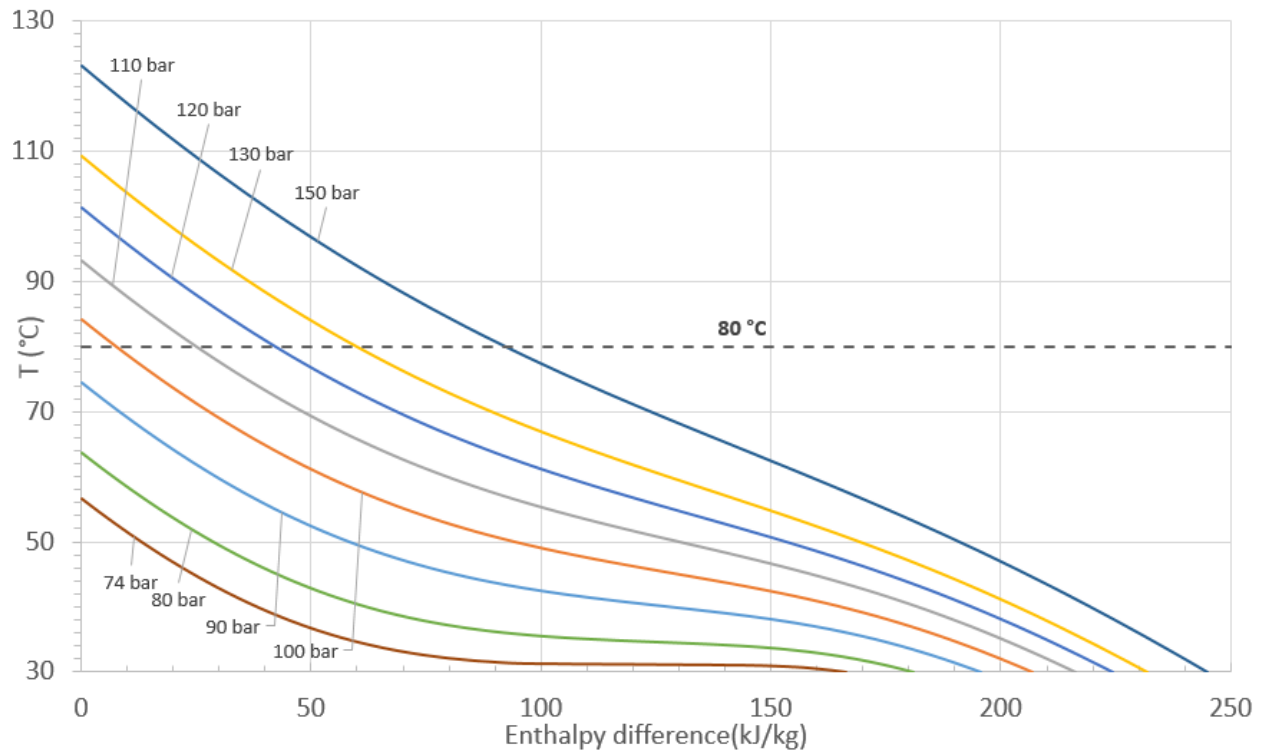


Figure 8. T-h diagram: heat recovery profile when $T_{eva} = 5\text{ °C}$ and $\eta_{is,c} = 0.7$.

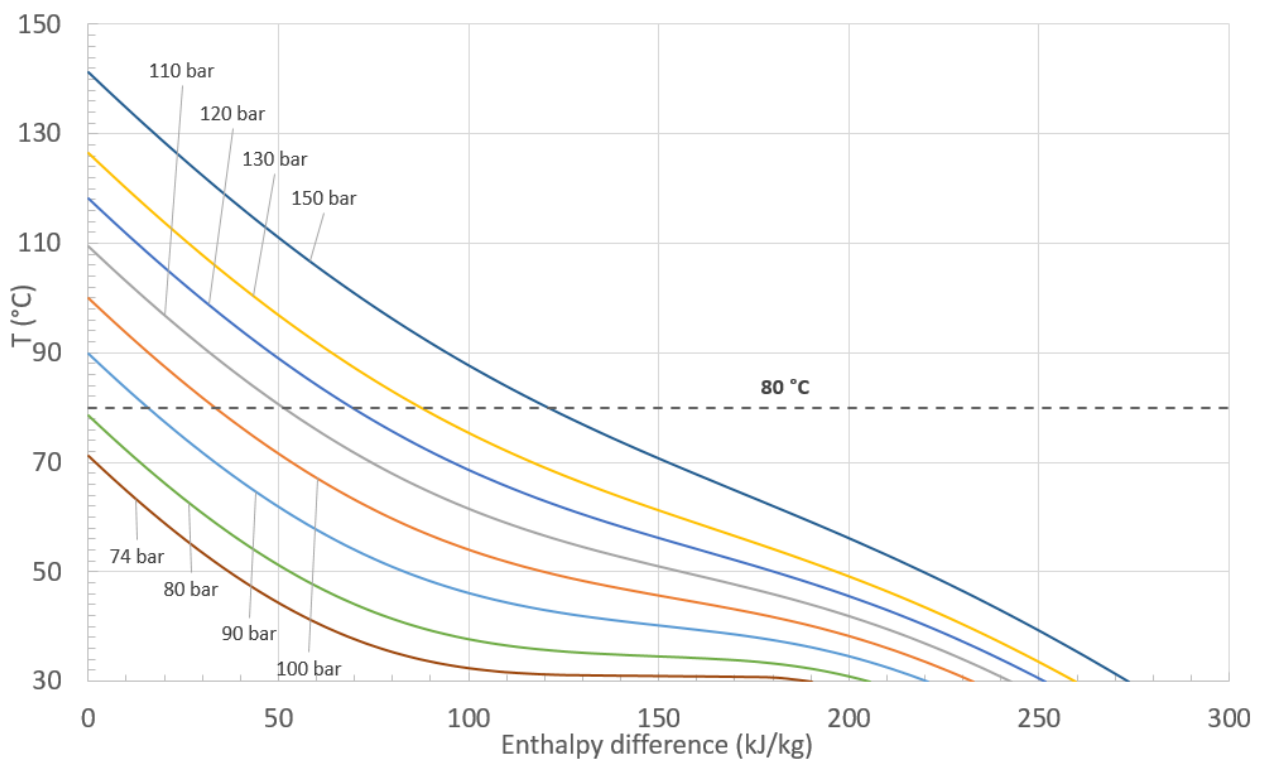


Figure 9. T-h diagram: heat recovery profile when $T_{eva} = -5\text{ °C}$ and $\eta_{is,c} = 0.7$.

6. Conclusions

A preliminary study has been conducted on a basic CO₂ transcritical inverse cycle, in order to evaluate the feasibility of the heat recovery at the gas cooler, where a relatively high-temperature waste heat is available, which usually is rejected to the environment. A general analysis was considered with some reasonable constraints, supposing a cooling machine is operating during summer season for conditioning service, when the environmental heat sink exhibits the worst conditions, assumed between 30 °C and 45 °C. The evaporation temperature was set between -5 °C and 5 °C and the high pressure was varied between 74 bar and 150 bar. As for the theoretically recoverable heat the system COP was calculated, which factors in both the useful effects, cooling and heating. A specific application for the waste heat exploitation was analyzed, that is an absorption chiller machine coupled with the main cycle, so that the latter provides the feeding heat to the former. The total cooling COP, defined as the useful cooling capacity provided by the basic cycle along with the absorption one, has been shown to be capable of exceeding the basic cooling COP alone by more than 30%, being very attractive for the highest temperature values of the heat sink and the lowest evaporation temperatures. Finally, the fundamental key points of the heat transfer have been highlighted related to the influence on the heat recovery capability, namely the T-h profiles matching of the transcritical CO₂ cooling stream to any fluid intended for the heat recovery, and a viable solution has been conceptually outlined.

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