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Yang Chen
Royal Institute of Technology

Per Lundqvist
Royal Institute of Technology

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Yang CHEN*[, 1], Per LUNDQVIST [2]

[1] M.Sc., Royal Institute of Technology, Div. of Applied Thermodynamics and Refrigeration,
Dept. of Energy Technology, Stockholm, Sweden
Tel: +46(0)790 7435, Fax: +46(0)8 20 41 61
E-Mail: yang.chen@energy.kth.se * Author for correspondence

[2] Ph.D., Docent, Royal Institute of Technology, Div. of Applied Thermodynamics and Refrigeration,
Dept. of Energy Technology, Stockholm, Sweden
Tel: +46(0)790 7452, Fax: +46 (0)8 20 41 61
E-Mail: Per.Lundqvist@energy.kth.se

ABSTRACT

Carbon dioxide transcritical cycles have become more and more investigated during the last decade. For all systems operating with such a cycle, there will be at least one heat exchanger to either heat or cool the supercritical carbon dioxide. Unlike in the sub-critical region, the supercritical carbon dioxide's thermophysical properties will have sharp variations in the region close to its critical point. This variation has a significant influence on the shape of the heat exchanger's temperature profile and the heat transfer performance of the heat exchanger. Therefore, the performance of the heat exchanger used for supercritical carbon dioxide cooling or heating process should be evaluated by taking this effect into account. This paper discusses the heat exchangers used for supercritical carbon dioxide refrigeration process including a suction gas heat exchanger in the cycle. Engineering Equation Solver (EES)[1] and Refprop 7.0[2] are used for cycle calculations and for properties calculations.

1. INTRODUCTION

Carbon dioxide is an environmental benign natural working fluid. With the increasing concern for environmental problems caused by the use of synthetic working fluids in different heating and cooling systems, the research on carbon dioxide transcritcal cycles has aroused increasing interest since the last decade. Compared to the systems that operate with other working fluids, systems operating with carbon dioxide have many advantages: Carbon dioxide is inexpensive and abundant in the nature. Compared with other natural working fluids, it is more chemically stable and reliable (i.e. non-explosive, non-corrosive). Further, due to its relatively high working pressure, the carbon dioxide system is more compact than the system operating with other working fluids. Therefore, carbon dioxide has been investigated for use as working fluid in many applications: refrigerators, heat pumps (Kim *et al.* 2004) but also for power production (Feher 1967, Dostal *et al.* 2004, Chen *et al.* 2005) for instance. Unless being used as a secondary working fluid in indirect system for refrigeration applications in supermarkets, the corresponding cycles for carbon dioxide systems in all these applications will be transcritical cycles (for refrigeration/heat pump and power production applications) or entirely supercritical cycles (for power production applications). The main difference between these two types of cycles is weather the cycle is partly located in the supercritical region or totally located in the supercritical region.

Unlike in the sub-critical region, the thermophysical properties of a supercritical working fluid will have sharp variations in the region close to its critical point. This characteristic will greatly influence the heat transfer characteristics and the temperature profile of the heat exchanger if it operates near the critical point. Consequently, this characteristic will have a significant influence on the heat exchanger size in different applications of the carbon dioxide transcritical cycles. Therefore, the performance of the heat exchanger used for supercritical carbon dioxide cooling or heating process should be evaluated by taking this effect into account. The specific heat (Cp), which is

1 Engineering Equation Solver: http://www.fchart.com/ees/ees.shtml
2 Refprop 7.0: http://www.nist.gov/srd/nist23.htm

the main factor that influences the working fluid temperature profile in the heat exchanger, is plotted as a function of temperature for different pressures in Figure 1.

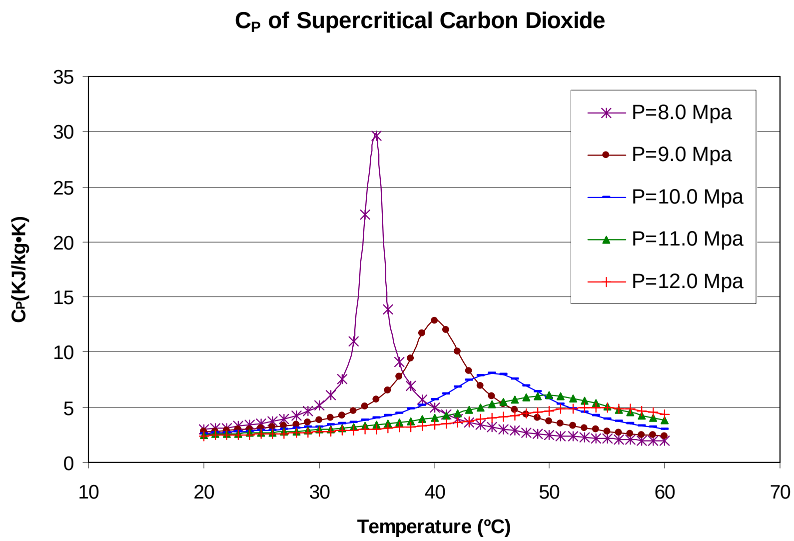


Figure 1. Supercritical carbon dioxide's specific heat *vs.* temperature at different pressures (calculation results from Refprop 7.0)

It can be noticed in Figure 1 that the specific heat of the supercritical carbon dioxide is changing with the temperature with a higher peak when the pressure get close to the critical pressure. Further, it may also be noticed that the temperature corresponding to the peak in specific heat is increasing with the increasing of pressure.

For the transcritical refrigeration cycle, this sharp variation can take place either in the Internal Heat Exchanger (IHX[3]),when transferring the heat to the low pressure side carbon dioxide or in the Gas Cooler (GC) ,while rejecting heat to the heat sink. However, compared to the specific heat variation of supercritical carbon dioxide, the specific heat of heat sink (e.g. cooling air for gas cooler) and low pressure side carbon dioxide are not varying much.

The specific heat variation of the heat sink (air) for the GC and the low pressure side carbon dioxide in the IHX are plotted as a function of temperature in figure 2. Note the differences in scale compared to figure 1.

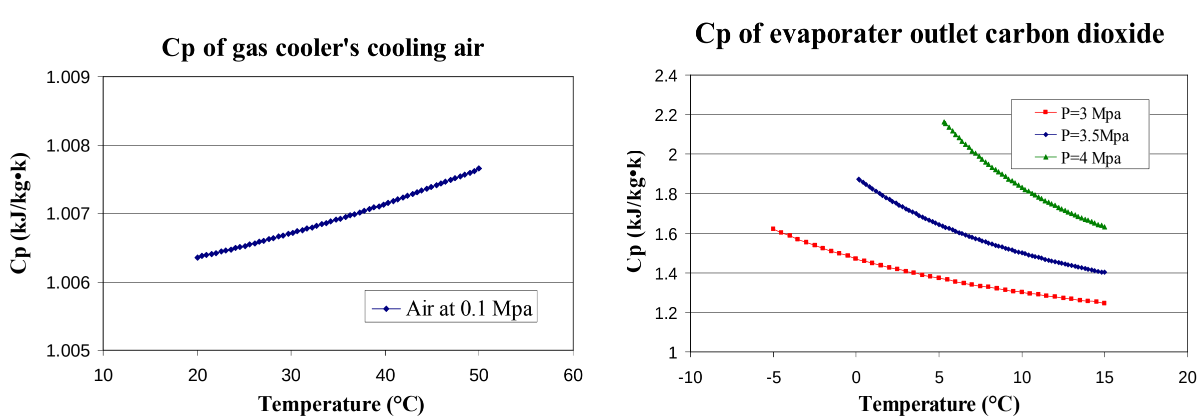


Figure 2. Specific heat of gas cooler's cooling air at atmospheric pressure and of evaporator's outlet carbon dioxide at different pressures (calculation results from EES and Refprop 7.0)

3 Sometimes referred to as suction gas heat exchanger

It can be observed from the above figures that the specific heat of supercritical carbon dioxide is changing over a much bigger range when compared to the heat sink (air) or the CO_2 at the evaporator's outlet. The difference in specific heats will thus influence the heat transfer performance of the heat exchanger and the shape of the heat exchanger's temperature profile (in both GC & IHX). Consequently, so called "pinching", which may limit the performance of the heat exchangers, may also occur in the heat exchanger. Therefore, the specific heat of different heat exchanger working fluids should be carefully examined when evaluating the heat exchangers that operate with supercritical carbon dioxide.

2. BASIC CYCLE ANALYSIS

A typical carbon dioxide transcritical refrigeration cycle can be analyzed as follows to show the influence of supercritical carbon dioxide specific heat's variation on the heat exchanger. The basic carbon dioxide transcritical refrigeration system is composed of five parts, namely: evaporator, compressor, GC, expansion valve and IHX. The schematic system layout is showed in figure 3.

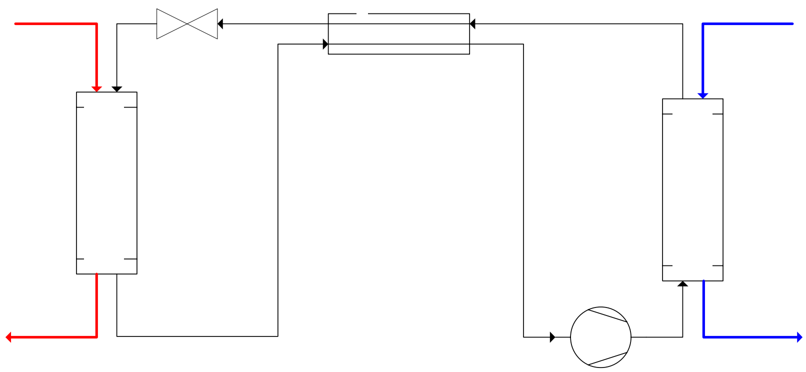


Figure 3. Schematic layout of carbon dioxide transcritical refrigeration system

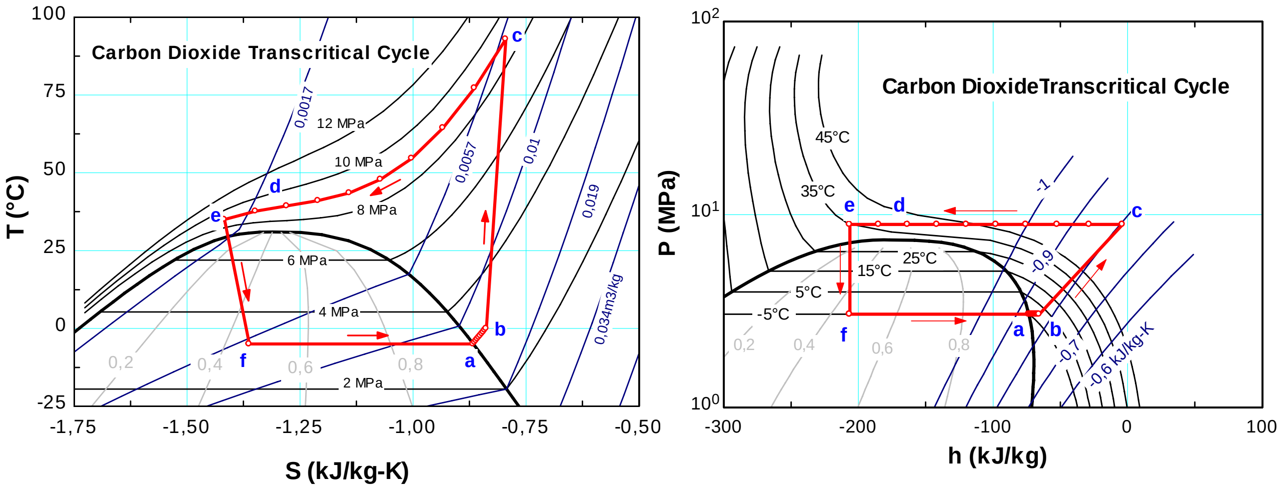
The cycle operating conditions are selected according the most commonly used working condition suggested by other researchers and in CO_2 automobile A/C prototype testing (Kim *et al*., 2004). The evaporation temperature is selected to 5°C and the corresponding pressure will be 3.97 Mpa. The compressor's isentropic efficiency is assumed to 75% according to the research done by Rozhentsev and Wang (2001). The gas cooler is assumed to be cooled by air with 20 ºC inlet temperature and 0.5kg/s available mass flow rate. For the heat rejection pressure, Liao *et al.* (2000) proposed a correlation to predict the optimum heat rejection pressure in terms of evaporation temperature and the GC's outlet temperature, which is expressed as equation (1).

$$p_{opt} = (2.778 - 0.0157 t_e) t_c + (0.381 t_e - 9.34) \qquad (1)$$

Based on equation (1), the optimum heat rejection pressure for the proposed working condition will be 8.7 Mpa Moreover, a 5 ºC superheat after the evaporator is assumed as a fixed value to ensure that there is no moisture at the compressor inlet. The cycle operating conditions are given in table 1 and the corresponding cycle T-S chart and logP-H chart are also plotted in figure 4.

Table 1. Basic combined cycle working condition

Items	Value	Unit
Evaporator pressure	3.97	Mpa
Evaporation temperature	5	ºC
Cooling capacity	10	KW
Superheat after Evaporator	5 (fixed value)	K
Gas cooler pressure	8.705	Mpa
Gas cooler outlet temperature	35	ºC
Compression efficiency	75%	-



(a) (b)

Figure 4. (a)T-S chart of carbon dioxide transcritical refrigeration cycle (b) logP-H chart of carbon dioxide transcritical refrigeration cycle (from EES)

The model implanted in EES shows that under the pre-described working condition, the air temperature after passing the condenser will be 46.3 °C and the supercritical carbon dioxide's temperature at the IHX's inlet will be 36.21 °C. Based on the calculations, a Cp-Δh chart is developed for the integrated total heat exchanger length, which includes both IHX and the GC[4], to show the specific heat variations of all the working fluids (i.e. supercritical CO_2, evaporator's outlet CO_2 and the GC's cooling air) along the two heat exchangers (figure 5). In figure 5, the left part of the curves shows the CP variation along the IHX (d-e for supercritical CO_2, a-b for evaporator's outlet CO_2) and the right part shows the CP variation along the GC (c-d for supercritical CO_2, g-h for gas cooler's cooling air). The arrows show the direction of the fluid flow. It can be noticed from figure 5 that after being compressed to the supercritical region, the supercritical CO_2 enters the GC with a moderate Cp value (c). Inside the GC, the Cp value of supercritical CO_2 increases slightly at the beginning and then increases sharply until it reaches its peak value. After this point, the Cp value starts to decrease until the supercritical CO_2 reaches the outlet of GC (d). After entering the IHX, the CP value of supercritical CO_2 keeps decreasing until it reaches the IHX's outlet (e). On the other side, the Cp value of evaporator's outlet CO_2 shows a slightly decreasing along the IHX (a-b), while the Cp value of GC's cooling air shows an almost "constant" value along the GC (g-h).

As mentioned before, the difference in the trend of Cp variations for different fluids will influence the shape of the temperature profile in both IHX and GC, which is showed in figure 6. As showed in figure 6, the supercritical CO_2's temperature has a more obvious drop near the inlet of GC (c) due to its relatively moderate Cp increment. After that, the temperature profile becomes relatively flat and then slightly drops again before the supercritical CO_2 exits the GC due to its Cp variation. On the other side, the temperature of GC's cooling air is increasing steadily due to its almost constant Cp value. In the IHX, the temperature of supercritical CO_2 decreases while the temperature of evaporator's outlet carbon dioxide increases respectively. It can be also seen from figure 6 that the supercritical CO_2's Cp variation has its main influence on the shape of temperature profile in the GC, which causes the temperature profile to show a concave shape. Due to this shape, the temperature differences at the heat exchanger ends are much bigger than inside the heat exchanger, thus the so-called "pinching" may occur inside the GC. Meanwhile, the temperature difference, which is the "driving force" for heat transfer to take place, is much smaller inside the GC than at its ends. Therefore, the required heat transfer area for the GC to remove a certain amount of heat will be much larger than the one without such a shape of temperature profile. Further, the logarithmic mean temperature difference, which is calculated by the measured temperature difference at both heat exchanger ends (equation 2), will over predict the real temperature difference of the heat exchanger (GC). Consequently, the UA value of the heat exchanger, which calculated by equation 3, will be under-estimated.

[4] All the heat exchangers analyzed in this paper are referring to counter flow heat exchangers.

$$\vartheta_{\ln} = \frac{(t_a - t_f) - (t_b - t_e)}{\ln \dfrac{(t_a - t_f)}{(t_b - t_e)}} \tag{2}$$

$$Q = UA \times \vartheta_{\ln} \tag{3}$$

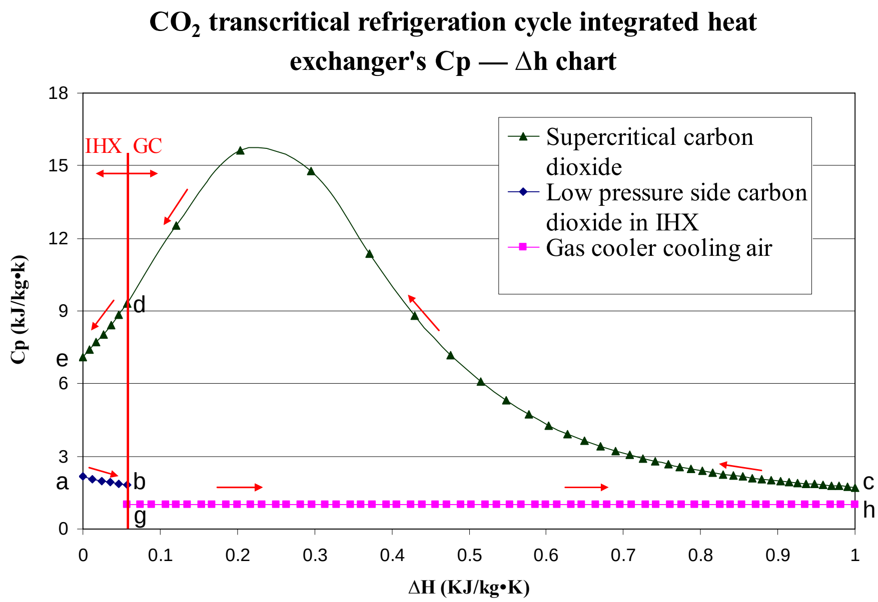


Figure 5. Integrated heat exchanger's CP-ΔH chart: (a) Carbon dioxide transcritical refrigeration cycle (10 kW cooling capacity, gas cooler's cooling air flow=0.5 kg/s)

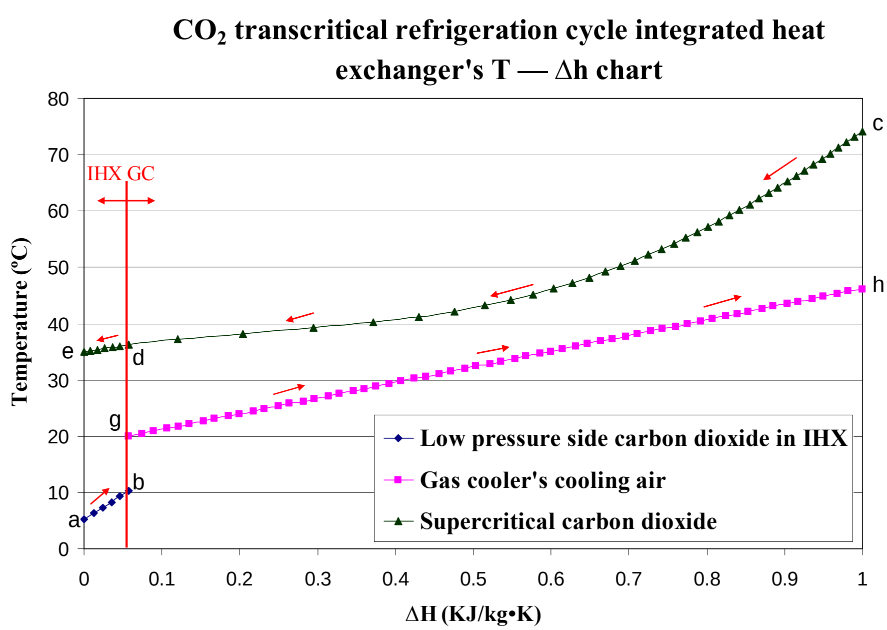
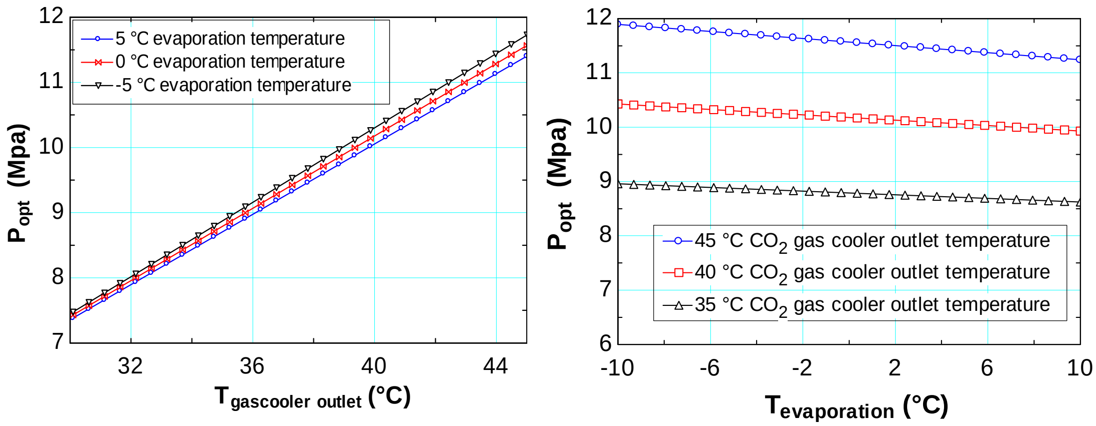


Figure 6. Integrated heat exchanger's T-ΔH chart: (a) Carbon dioxide transcritical refrigeration cycle (10 kW cooling capacity, gas cooler's cooling air flow=0.5 kg/s)

5. DISCUSSION

The optimum GC pressure for carbon dioxide transcritical refrigeration cycle obtained by equation 1 is related to both evaporation temperature and the GC's outlet temperature. The calculated optimum GC pressure is plotted against different GC's outlet temperatures for different evaporation temperatures as well as against different evaporation temperatures for different GC outlet temperatures in figure 7 respectively.



(a) (b)

Figure 7. Carbon dioxide transcritical refrigeration cycle (a) optimum GC pressure against different GC outlet temperature at different evaporation temperatures (b) optimum GC pressure against different evaporation temperatures at different GC outlet temperatures

It can be noticed in figure 7 that for a certain evaporation temperature, the higher the GC's outlet temperature is, the higher the optimum GC pressure will be. While for a certain GC's outlet temperature, the higher the evaporation temperature is, the lower the optimum GC pressure will be. Further, the GC's outlet temperature has much more influence on the cycle's optimum GC pressure than the evaporation temperature has.

Maintaining the evaporation temperature as 5 °C, the temperature profile of supercritical carbon dioxide in the GC and the IHX is plotted in a ΔT-ΔH chart (figure 8) for different GC's outlet temperatures. For every GC outlet temperature, the pressure is kept at optimum GC pressure that calculated by equation 1.

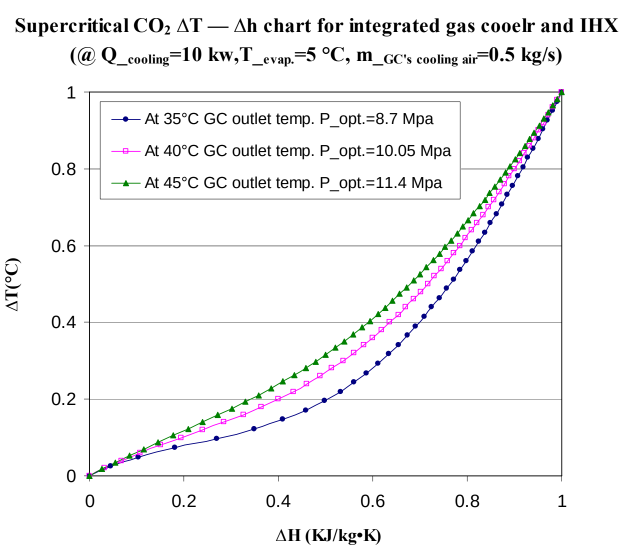


Figure 8. Supercritical carbon dioxide's ΔT--ΔH chart for the integrate heat exchanger length (includes GC and IHX) at different gas cooler outlet temperatures

The results clearly show that the lower the GC outlet temperature is (i.e. the lower gas cooler pressure), the more obvious will the concave shape of the temperature profile be. As mentioned above, the concave shaped temperature profile will influence the heat exchanger size (Area). Therefore, the lower GC outlet temperature is, the bigger the required heat exchanger size will be to transfer a certain amount of heat.

However, when selecting the GC's outlet temperature, one also needs to consider its influence on the system COP. As showed in figure 9, the lower the GC's outlet temperature is, the higher the system COP will be, due to the reduction of throttling losses in the expansion valve by reducing the GC's outlet temperature.

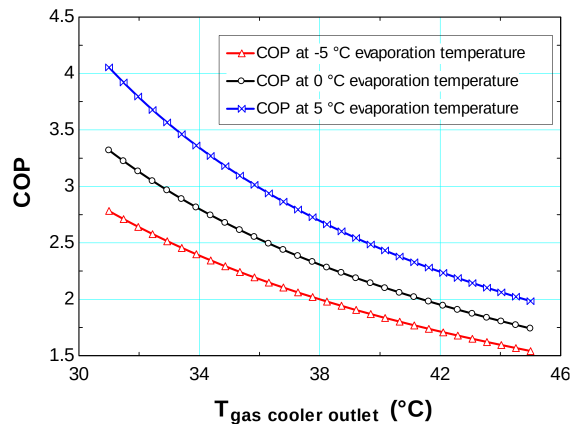


Figure 9. Carbon dioxide transcritical refrigeration cycle's COP vs. GC outlet temperature at different evaporation temperature

6. CONCLUSIONS

In this paper, a basic carbon dioxide transcritical refrigeration cycle has been analyzed to show the influence of supercritical carbon dioxide specific heat's sharp variation on the heat exchanger performance. The results calculated in EES and Refprop 7 show that due to the sharp variation of supercritical carbon dioxide's specific heat, the temperature profile in the gas cooler will show a concave shape. Due to the shape of the temperature profile, the temperature difference, which is the "driving force" for heat transfer to take place, will be much smaller inside the gas cooler than at its ends. Therefore,

- To remove a certain amount of heat from the gas cooler, the required heat exchanger surface will be much bigger than the one without such a shape of temperature profile.
- The logarithmic mean temperature difference, which is calculated by the measured temperature difference at the heat exchanger ends, will over predict the real temperature difference for the heat exchanger (gas cooler).
- The UA value, which is calculated by the measured logarithmic mean temperature difference, will be under estimated.

It is also found that the gas cooler's outlet temperature has a crucial influence on the value of optimum gas cooler pressure and consequently the temperature profile's shape in the gas cooler. The higher the gas cooler outlet temperature is, the less concave shape the temperature profile inside the gas cooler will be, which is an advantage from the heat exchanger design viewpoint. However, higher gas cooler outlet temperature will also leads to a lower COP for the system.

NOMENCLATURE

IHX	Internal Heat Exchanger	(-)	**Subscripts**	
GC	Gas Cooler	(-)	a-f	cycle route point
Cp	specific heat	(kJ/kg•k)	g-h	points for air properties

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