ENERGETIC AND EXERGETIC ANALYSES OF CARBON DIOXIDE TRANSCRITICAL REFRIGERATION SYSTEMS FOR HOT CLIMATES

by

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In the last two decades many scientific papers and reports have been published in the field of the application of the CO_2 as a refrigerant for refrigeration systems and heat pumps. Special attention has been paid to the transcritical cycle. However, almost no papers discussed such cycles for hot climates, i. e., when the temperature of the environment is higher than 40 ºC during a long period of time. This paper deals with the energetic and exergetic evaluation of a CO_2 refrigeration system operating in a transcritical cycle under hot climatic conditions. The performance and exergy efficiency of the CO_2 refrigeration system depend on the operation conditions. The effect of varying these conditions is also investigated as well as the limitations associated with these conditions.

Key words: *refrigeration system, carbon dioxide, transcritical cycle, energy analysis, exergy analysis, hot climatic conditions*

Introduction

The application of carbon dioxide (CO_2) for refrigeration is known from the middle of the 19th century, including operation based on a transcritical cycle. From the middle of 1920's until the middle of 1980's this refrigerant was out of the interest of scientists and practical engineers because of the (a) grooving interest to the newly introduced artificial refrigerants (CFC and HFC refrigerants, or so-called "freons") due to their advantages compared with the "old" refrigerants, and (b) technical limitations related to the large practical application of CO_2. In the 1990's the interest to the so-called "natural refrigerants" (CO_2, ammonia, NH_3, propane, butane, and water) was renewed, especially for CO_2, due to considerations related to ozone depletion potential (ODP) and global warming potential (GWP), which have restricted the use of CFC and HFC refrigerants [1]. CO_2 has some unique properties that make this refrigerant completely different than other "natural refrigerants". The technical developments during the last decades helped overcome many of the barriers for the wide application of CO_2, but still we need to investigate the rational application of this attractive refrigerant.

A large number of scientific publications related to the theoretical and practical investigations of the different refrigeration and heat pump systems using CO_2 followed after publication of the papers published by Lorentzen (for example, [2, 3]). In the 2000s we already had sev-

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eral papers [4-6], where very good detailed reviews of many publications have been reported. However, in all these publications only European (mainly Northern European) climatic conditions are considered for the operation of refrigeration and heat pump systems.

The Middle East is an interesting region of the world to study because this region has experienced impressive increases in economic growth, and energy demand [7]. For countries with hot climates, the energy consumption related to refrigeration processes is much higher than for other countries. It is caused by the expanded application of refrigeration processes (especially for air conditioning systems) and by the higher temperature of the environment (temperature of a cooling medium) that leads to a higher pressure ratio within the refrigeration system, and, therefore to higher energy consumption.

The operation of a CO_2 refrigeration system at a high temperature of the environment can be similar to the operation of a CO_2 heat pump. Therefore, the following publications with corresponding assumptions and results have been considered here. Neksfit *et al.* [8] reported optimal values of the pressure as well as the isentropic and volumetric efficiencies of the CO_2 compressors for heat pump applications in the range of the inlet water temperature between 7 and 20 ºC and corresponding temperature of the evaporation between –25 ºC and 15 ºC, and hot water temperature between 55 and 80 ºC. In this range of temperatures, the pressure ratio is varied between 2 and 5. The isentropic efficiency of the compressor is varied between 0.81 and 0.75 and the volumetric efficiency between 0.9 and 0.78. Schmidt *et al.*, [9] investigated the characteristics of high-temperature heat pumps with a transcritical CO_2 process for drying purposes with a maximal temperature of 60 ºC. The isentropic efficiency of the compressor was varied between 0.65 and 0.7.

An interesting review of CO_2 heat pump systems is published by Neksa [10], however only a relative low temperature for the hot water is considered with a maximal pressure of 90 bar and a minimal pressure of 35 bar. For these operation conditions, the isentropic efficiency of the compressor was 0.92 for the pressure ratio 2.4 and 0.68 for the pressure ratio 3.2. Cecchinato *et al*. [11] reported a similar heat pump system with maximal temperature of the hot water of 45 ºC. The maximal pressure within the gas cooler is assumed to be 115 bar, whereas the isentropic efficiency of the compressor is varied between 0.6 and 0.63.

Yokoyamaa *et al*. [12] investigated the effect of ambient temperatures (inlet water temperature) on the performance of a CO_2 heat pump. At minimal inlet water temperature 5 to 15 ºC, the maximal achieved water temperature was 60 ºC at a pressure in the gas cooler of 102.8 bar.

In the research done by Fernandez *et al*. [13], the operation conditions for the heat pump include a temperature of evaporation of 10 ºC, and a maximal pressure within gas cooler of 110 bar.

Zhang *et al*. [14] reported also experimental studies on the optimum pressure within a heat exchanger for a CO_2 heat pump system. The minimal temperature within the evaporator is 10 ºC and the maximal pressure within the gas cooler is 125 bar. The isentropic efficiency of the compressor is in the range of 0.65 to 7.

An interesting fact is that an exergy analysis has been used in many publications related to CO_2 refrigeration machines and heat pumps (started with [3]). However, only selected data obtained from the exergy analysis are reported in these publications. For example, Bilgen and Takahashi [15] conducted exergy analysis for the CO_2 heat pump systems based on experimental study. The results are given as dimensionless variables (introduced by authors) that are not comparable with other results. Tao *et al*. [16] also conducted an exergy analysis based on experimental results but for a CO_2 residential air-conditioning system. The temperature of the environment was varied between 10 and 14 ºC. The information about the pressure in the gas cooler is missed. The results are: 30% of the total exergy destruction takes place within the expansion valve, 19% within the compressor, 31% within the gas cooler, and 20% within the evaporator. Srinivasan *et al*. [17] used an exergy analysis as an application for a newly developed fundamental equation of state for CO_2. The exergy analysis is conducted in the wide range of the evaporation temperature between –20 ºC and 10 ºC and pressure ratio between 2 and 10. The temperature of the environment is assumed to the equal to 40 ºC. The value of *COP* varies between 0.95 and 1.8 and the exergy efficiency between 0.07 and 0.17. Fartaj *et al*. [18] reported entropy generation and exergy analyses for the comparison of transcritical CO_2 and subcritical R134a refrigeration cycles. For the CO_2 cycle the minimal pressure within the evaporator is 42 bar (temperature is around 9 ºC), and within the gas cooler 111.3 bar. The reported exergetic variables for the components are doubtful because clear definitions are not given. The overall exergy efficiency is 0.34.

Sarkar *et al.* [19] discussed an exergy analysis for transcritical CO_2 heat pump systems with focus on the heat transfer and fluid flow effects. The temperature of the environment is assumed to be 40 ºC. The variation in the isentropic efficiency of the compressor between 0.5 and 0.8 leads to a variation of the exergy efficiency of the overall system between 0.21 and 0.37. In all mentioned publications the exergy analysis was conducted in the term of "exergy inlet/exergy outlet" based on the methodology given by Kotas [20].

The goal of this research is the energetic and exergetic investigation of a transcritical CO_2 refrigeration system for hot climates, particularly for the Middle Eastern countries.

Schematic and simulation

In this paper a simple CO_2 transcritical refrigeration system (fig. 1) is studied. The temperature of evaporation is assumed to be –10 ºC (industrial storage of a wide range of food products). The environmental temperature, *i. e*. the minimal temperature of cooling medium for the gas cooler is assumed to:

– be 25 ºC as a reference condition for Europe (as in many already mentioned publications), and
– vary between 35 ºC and 60 ºC.

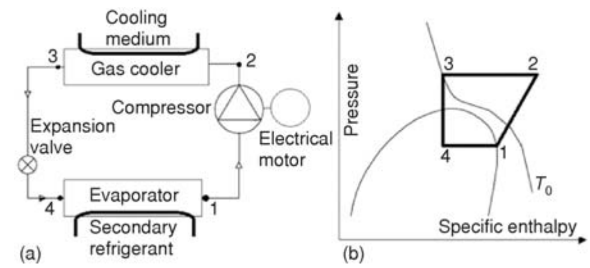

Figure 1. Flow diagram (a) and thermodynamic cycle (b) of a simple CO_2 transcritical refrigeration system

A simple CO_2 transcritical refrigeration system can also contain an internal heat exchanger and/or a semi-hermetic compressor.

Many factors affect the performance of a CO_2 refrigeration system, for example:

– for the compressor
  – isentropic efficiency of the compressor,
  – maximal pressure at the outlet of the compressor,
  – maximal temperature at the outlet of the compressor (related to the thermal stability and non-flamability of a lubricant),
  – superheating of the refrigerant within the electrical motor (if a semi-hermetic compressor is used),

- for the gas cooler
  - pressure drop within the gas cooler,
  - temperature difference between refrigerant at outlet of the gas cooler and temperature of the cooling medium,
- for the evaporator
  - temperature difference within the evaporator,
  - pressure drop within the evaporator,
  - superheating of the refrigerant within the evaporator, and
  - temperature difference within the internal heat exchanger (if this component is used).

The following information was obtained from publications by other authors and from data of companies producing equipment for CO_2 transcritical refrigeration systems, for example [21-25]:

- The efficiency of the CO_2 refrigeration system strongly depends on the pressure ratio in the compressor (simple system or with internal heat exchanger); the effect of including an internal heat exchanger is negligible.
- The superheating process within the evaporator (in the range between 0 and 10 K) does not affect the overall efficiency; a similar conclusion is obtained for the superheating of the refrigerant within the electrical motor of a semi-hermetic compressor.
- The temperature at the outlet of the compressor is, in general, not limited, when synthetic lubricants are used.
- The pressure within the gas cooler depends on the limitations for the compressors. The following information was obtained from refrigeration companies: "Bock" [23] and "Bitzer" [24] $p_{EV}^{\max} = 100$ bar and $p_{GC}^{\max} = 150$ bar; "Adson – Engineering Corporation" [25] – $p_{EV}^{\max} =$ $= 40$ bar and $p_{GC}^{\max} = 135$ bar.

The following assumptions were made for the simulations:

- State 1 is saturated vapor.
- The temperature at state 3 is equal to the minimal temperature of the cooling medium (T_0).
- The cooling medium is unknown and the design of the gas cooler is not specified, therefore, for simplification we can consider only an average temperature for the cooling medium, $T_{cm}^{a} = T_0 + 5$ K.
- The secondary refrigerant is also unknown and the design of the evaporator is also not specified, therefore, for simplification we can consider only an average temperature for the secondary refrigerant, $T_{sr}^{a} = T_{EV} + 5$ K.
- The pressure drop within the evaporator is assumed to be 0.75 bar and within the gas cooler 3 bar.
- The isentropic efficiency of the compressor is equal to 0.8 and the efficiency of the electrical motor is equal to 0.88.
- For the sensitivity analysis the isentropic efficiency of the compressor is varied between 0.7 and 0.85.

For the simulation of the CO_2 refrigeration systems the software CoolPack [26] was used.

Evaluation

Energetic evaluation

The model for the energetic evaluation of the CO_2 transcritical refrigeration system consists on the following equations (fig. 1):

- specific heat of evaporation

$$q_{\mathrm{EV}} = h_1 - h_4 \tag{1}$$

– specific work of the compression process

$$w_{\mathrm{CM}} = h_2 - h_1 \tag{2}$$

– specific heat for gas cooling

$$q_{\mathrm{GC}} = h_2 - h_3 \tag{3}$$

– mass flow rate of CO_2

$$\dot{m}_{\mathrm{CO_2}} = \frac{\dot{Q}_{\mathrm{EV}}}{q_{\mathrm{EV}}} \tag{4}$$

– power of the compressor

$$\dot{W}_{\mathrm{CM}} = w_{\mathrm{CM}} \dot{m}_{\mathrm{CO_2}} \tag{5}$$

– power of the electrical motor

$$\dot{W}_{\mathrm{EM}} = \frac{\dot{W}_{\mathrm{CM}}}{\eta_{\mathrm{EM}}} \tag{6}$$

– gas cooler heat rate

$$\dot{Q}_{\mathrm{GC}} = q_{\mathrm{GC}} \dot{m}_{\mathrm{CO_2}} \tag{7}$$

– *COP*

$$COP = \frac{\dot{Q}_{\mathrm{EV}}}{\dot{W}_{\mathrm{EM}}} \tag{8a}$$

However, in many publications the value of *COP* is calculated as:

$$COP = \frac{\dot{Q}_{\mathrm{EV}}}{\dot{W}_{\mathrm{CM}}} \tag{8b}$$

The CO_2 refrigeration system can be optimized from the thermodynamic point of view: $\dot{W}_{CM} \to \min$ by keeping $\dot{Q}_{\mathrm{EV}} = \mathrm{const.}$

Exergetic evaluation

An exergy analysis identifies the location and magnitude of the thermodynamic inefficiencies [27-29]. Exergy is the maximum theoretical useful work (shaft work or electrical work) obtainable from a system during a process that brings this system into equilibrium with the thermodynamic environment (temperature T_0 and pressure p_0) while interacting only with this environment [28]. The exergy of the system E_{sys} consists of four main components: Physical exergy, chemical exergy, kinetic exergy, and potential exergy:

$$E_{\mathrm{sys}} = E_{sys}^{PH} + E^{\mathrm{CH}} + E^{\mathrm{KN}} + E^{\mathrm{PT}} \tag{9}$$

Usually, the potential and kinetic exergy changes can be neglected. For compression refrigeration systems, the chemical exergy can also be neglected. Physical exergy is the maximum theoretical useful work obtainable as the system passes from its initial state (T, p, x) to the restricted dead state (T_0, p_0, x) while heat transfer takes place only between the system and the environment. Physical exergy for a material stream is:

$$E^{\mathrm{PH}} = \dot{m}e^{\mathrm{PH}} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{10}$$

However, for refrigeration systems, the physical exergy of material streams can be further split into a thermal part and a mechanical part [30]:

$$e^{\mathrm{PH}} = e^{\mathrm{T}} + e^{\mathrm{M}} \tag{11}$$

The exergy analysis is based on exergy balances written for the overall system:

$$\dot{E}_{\rm F,\,tot} = \dot{E}_{\rm P,\,tot} + \sum_k \dot{E}_{\rm D,\,k} + \dot{E}_{\rm L,\,tot} \tag{12a}$$

and for each component of the system:

$$\dot{E}_{\rm F,\,k} = \dot{E}_{\rm P,\,k} + E_{\rm D,\,k} \tag{12b}$$

For the simple CO_2 transcritical refrigeration system $\dot{E}_{\rm F,\,tot} = \dot{W}_{\rm EM}$, $\dot{E}_{\rm P,\,tot} = \dot{E}_{\rm P,EV}$ and $\dot{E}_{\rm L,\,tot} = \dot{E}_{\rm P,GC}$ (exergy transport to the environment [27, 28]). The fuel and product of each component is given by the equations:

- compressor $\dot{E}_{\rm F,CM} = \dot{W}_{\rm CM}$ and $\dot{E}_{\rm P,CM} = \dot{E}_2 - E_1$,
- electrical motor $\dot{E}_{\rm F,EM} = \dot{W}_{\rm EM}$ and $\dot{E}_{\rm P,EM} = \dot{W}_{\rm CM}$,
- gas cooler $\dot{E}_{\rm F,GC} = \dot{E}_2 - \dot{E}_3$ and $\dot{E}_{\rm P,GC} = \dot{Q}_{\rm GC}(1 - T_0/T^{\rm a}_{\rm cm})$,
- expansion valve $\dot{E}_{\rm F,EXV} = \dot{E}^{\rm M}_3 - \dot{E}^{\rm M}_4$ and $\dot{E}_{\rm F,EXV} = \dot{E}^{\rm T}_4 - \dot{E}^{\rm T}_3$,
- evaporator $\dot{E}_{\rm F,EV} = \dot{E}_4 - \dot{E}_1$ and $\dot{E}_{\rm P,EV} = \dot{Q}_{\rm EV}(1 - T_0/T^{\rm a}_{\rm sr})$.

The exergy destruction in the k-th component is $\dot{E}_{\rm D,\,k} = \dot{E}_{\rm F,\,k} - \dot{E}_{\rm P,\,k}$.

The exergy destruction ratio for the k-th component is:

$$y_{\rm k} = \frac{\dot{E}_{\rm D,\,k}}{\dot{E}_{\rm D,\,tot}} \tag{13}$$

The exergy efficiency of the overall refrigeration systems is:

$$\varepsilon_{\rm tot} = \frac{\dot{E}_{\rm P,\,tot}}{\dot{E}_{\rm F,\,tot}} \tag{14}$$

For the exergy analysis T_0 is equal to the minimal temperature of the cooling medium and p_0 =1 bar.

Results

Figure 2 demonstrates the values of *COP* and $\varepsilon_{\rm tot}$ for different assumed temperatures T_0 in order to estimate the optimal pressure within the gas cooler. The character of *COP* curves obtained here is qualitatively similar to the data reported by Kauf [21], *i .e.* flat character of the *COP* functions for $T_0 > 40$ ºC and $COP < 1$ for $T_0 > 50$ ºC. With increasing temperature of the environment, the optimal value of the pressure in the gas cooler increases (tab. 1). Taking into account the equipment limitations associated with the pressure at the outlet of the compressor, we can conclude, that for the range T_0 = 34 to 50 ºC the possibility to optimize the system thermodynamically

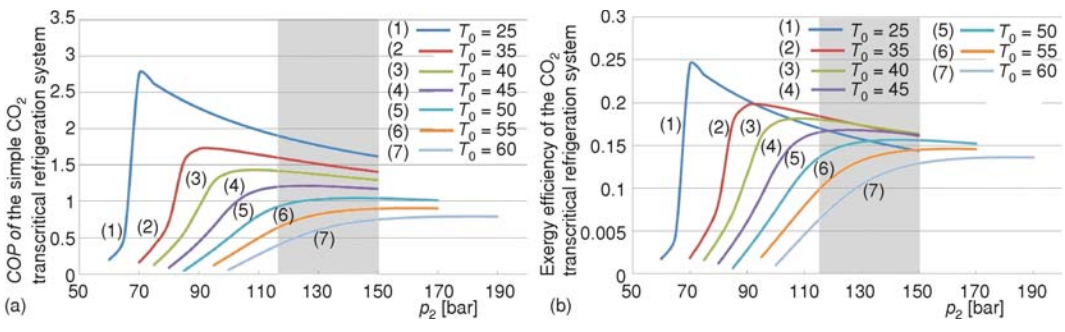

Figure 2. *COP* and exergy efficiency ($\varepsilon_{\rm tot}$) of the simple CO_2 transcritical refrigeration system

is limited (shaded zone in fig. 2) and for $T_0 > 50$ ºC we cannot achieve thermodynamically optimal operation conditions.

For European operation conditions (T_0 is around 25 ºC) the value of *COP* at optimal operation conditions is around 2.7. With increasing temperature of the environment in hot climates – assuming $T_0 > 40$ ºC, the values of *COP* tend to 1 or even lower. This makes less attractive the use of CO_2 transcritical refrigeration systems for hot climates; however a one-stage compression process is practically possible for both European operation conditions and hot climates operation conditions because $p_2/p_1 = 2.72$ for $T_0 = 25$ ºC and $p_2/p_1 = 7.0$ for $T_0 = 60$ ºC [23-25].

Table 1. Optimal pressure in the gas cooler

Temperature of the environment, T_0 [ºC]	Optimal pressure, p_{GC}^{opt} [bar]
25	70
35	95
40	110
45	125
50	140
55	160
60	180

Figure 3 shows two characteristics of the thermodynamic cycle that are usually not reported in the literature: (a) The temperature at the outlet of the compressor, fig. 3(a) and (b) mass flow rate of the refrigerant, fig. 3(b). Based on data from [24], for $p_{GC}^{opt} > 120$ bar, problems with the lubricant may arise, *i. e*., the standard lubricants for the CO_2 transcritical refrigeration systems should be replaced with special ones adapted to the higher pressures.

It is obvious that during the operation the value of p_{GC}^{opt} cannot be kept constant. Small variations always take place. Based on data in fig. 2, two values of pressure ratio ($p < p_{GC}^{opt}$ and $p > p_{GC}^{opt}$) correspond to the same value of the *COP* or ε_{tot}. The question is: which value is preferable if p_{GC}^{opt} is not used. To answer this question we should consider the value of the mass flow rate of the refrigerant. This value affects the power supplied to the compressor as well as the size and, therefore, the cost of the components. Thus, the value of $\dot{m}_{CO_2}$ should be as small as possible. Figure 3(b) demonstrates the interdependences between p_{GC} and $\dot{m}_{CO_2}$ for different values of T_0. For all values of T_0, the mass flow rates of the refrigerant drop significantly during increasing p_{GC} until the optimum value p_{GC}^{opt} is achieved, and after that become almost constant for $p_{GC} > p_{GC}^{opt}$. Therefore, the case $p > p_{GC}^{opt}$ is better for practical applications.

The exergy analysis shows that the exergy efficiency is relative low and it varies between $\varepsilon_{tot} = 0.25$ for $T_0 = 25$ ºC and $\varepsilon_{tot} = 0.14$ for $T_0 = = 60$ ºC. The values of ε_{tot} reported in other publications are higher because a different definition of the exergy efficiency was used there.

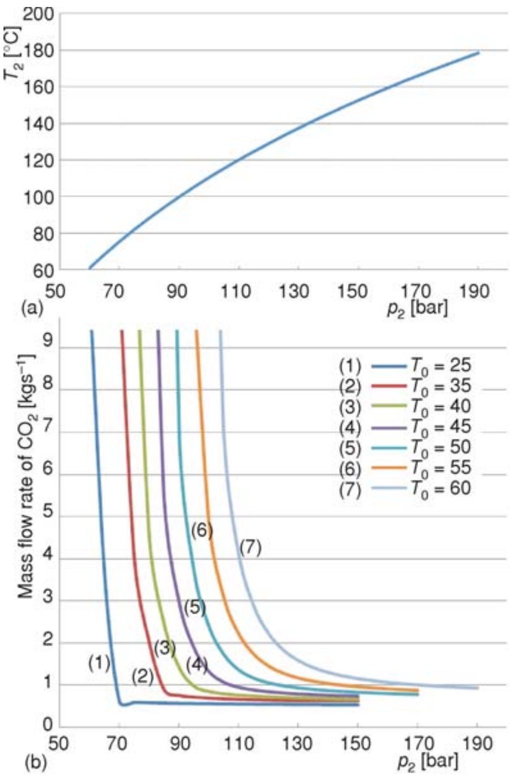

Figure 3. Temperature at the outlet of the compressor (a) and the mas flow rate of the refrigerant (b) as a function of p_2.

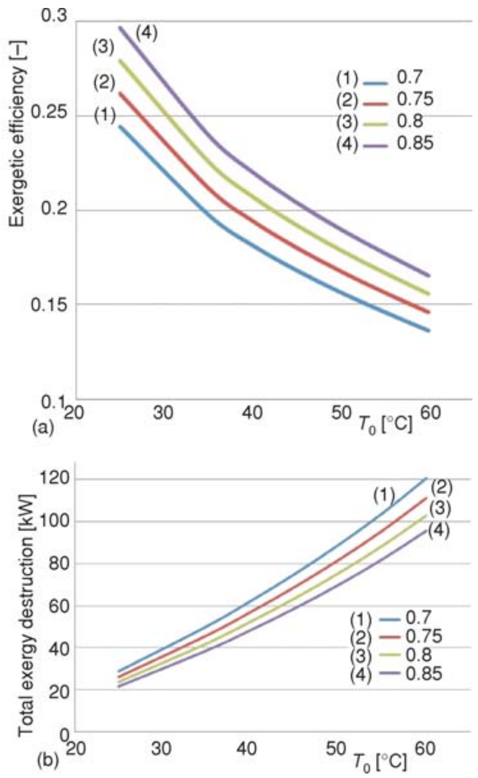

Figure 4. Effect of the isentropic efficiency of the compressor on the exergy efficiency of the refrigeration system ε_{tot} (a) and the total exergy destruction $\dot{E}_{D,tot}$ (b)

The influence of the isentropic efficiency of the compressor is studied using exergy analysis. Figure 4(a) shows the values of ε_{tot} as a function of $\eta_{CM}(\eta_{CM} = 0.7$ to $0.85)$, and fig. 4(b) shows how the isentropic efficiency of the compressor affects the total exergy destruction within the overall refrigeration system. The reader can see that the curves of ε_{tot} are almost parallel, but the curves of $\dot{E}_{D,tot}$ diverge at higher T_0 values.

Finally the values of the exergy destruction ratios (y_k) are given as a function of T_0 at the optimal pressure in the gas cooler (fig. 5). With increasing temperature of the environment (T_0) the ratio of exergy destruction within the compressor decreases from 23% down to 17%, whereas the ratio for the electrical motor almost remains constant, for the evaporator decreases from 17% down to 9%, for the gas cooler increases from 11% up to 20%, and for the expansion valve increases from 31% up to 39%. Based on these data we can conclude that in order to improve the CO_2 transcritical refrigeration system especially for hot climates we should focus on the irreversibilities within the expansion valve. As we know, this process cannot be improved by itself, therefore the structure of the refrigeration system and/or the operation conditions for other components should be modified.

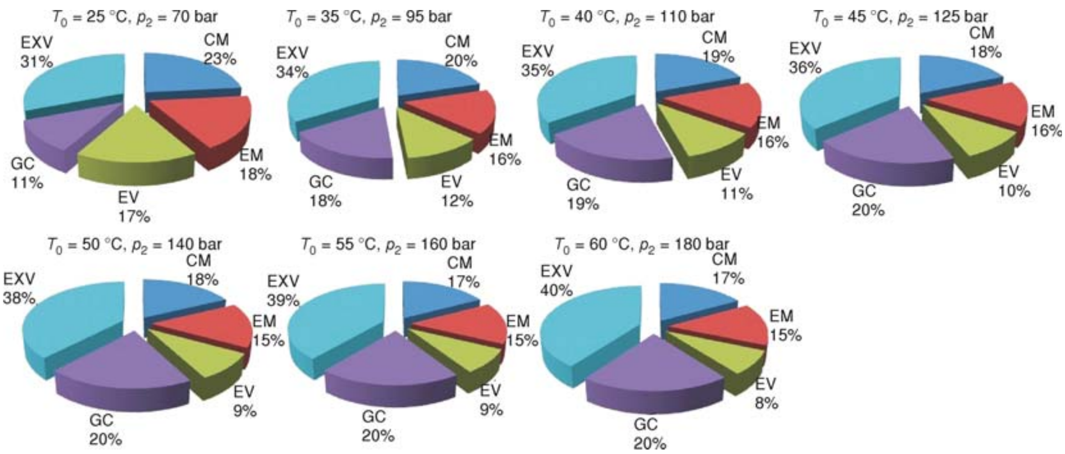

Figure 5. Exergy destruction ratios for the different operation conditions of the CO_2 transcritical refrigeration system (for color image see journal web site)

Conclusions

In this paper the transcritical CO_2 refrigeration system is evaluated based on energetic and exergetic criteria. Special attention is given to the operation conditions for this system in hot climates, *i. e*. when the temperature of the environment is higher than 40 ºC. The optimal pressure in the gas cooler is defined for each considered value of T_0 assuming that the temperature of the generated cold remains constant (–10 ºC). The analysis of the optimal pressure in the gas cooler and the limitations associated with the pressures for modern available compressors for transcritical CO_2 refrigeration systems demonstrate that optimal operation conditions cannot always be achieved. Based on exergetic evaluations, a very important conclusion in obtained. The most important issue for the improvement of the transcritical CO_2 refrigeration system is to find an engineering solution for decreasing the exergy destruction (irreversibilities) within the expansion valve. In this way the transcritical CO_2 refrigeration system will become also attractive for the countries with hot climates. Further work will focus on the economic evaluation of the transcritical CO_2 refrigeration system.

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Nomenclature

COP – coefficient of performance, [–]
$\dot{E}$ – exergy rate, [MW]
e – specific exergy, [kJkg^{-1}]
h – specific enthalpy, [kJkg^{-1}]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
p – pressure, [MPa]
$\dot{Q}$ – heat rate, [MW]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [ºC]
$\dot{W}$ – power, [MW]
0 – reference state for the exergy analysis

Greek symbols

ε – exergy efficiency, [–]
η – isentropic efficiency, [–]

Aronyms

CM – compressor
EM – electrical motor
EV – evaporator
EXV – expansion valve
GC – gas cooler

Subs- and superscripts

a – average
cm – cooling medium
D – exergy destruction
F – fuel
k – *k*-th component
P – exergy of product
sr – secondary refrigerant
tot – total

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