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Thermodynamic Performance Assessment of a CO_2 Supermarket Refrigeration System with Auxiliary Compression Economization by using Advanced Exergy Analysis

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Abstract

Supermarkets are currently one of the most vital service facilities, whose number of installations is ever-growing in both developed and developing countries. On the other hand, these applications feature a copious indirect contribution to the ongoing climate change, as well as a massive use of potent greenhouse gases. In an attempt to promote climate-friendlier technologies in commercial refrigeration sector, "CO_2 only" (transcritical CO_2 or pure CO_2) refrigeration systems have become the mainstream of new food retails worldwide. In particular, in the last few years parallel (or auxiliary) compression has taken root in food retails as a means to enhance the energy efficiency of pure CO_2 units. The thermodynamic performance of such a promising solution can be suitably assessed with the aid of the advanced exergy analysis. The results obtained at the design outdoor temperature of 40 °C showed that the main compressor has the highest priority of enhancement, whereas the high-pressure expansion valve needs to be replaced with a device for expansion work recovery. Also, close attention had to be paid to both the gas cooler and the auxiliary compressor. The former can be improved mainly by enhancing the other components, whereas the irreversibilities related to the latter can be decreased by improving both the compressor itself and the remaining components. Finally, the implemented sensitivity analysis revealed that the improvement in the efficiency of the main compressor should be seriously considered on the part of the manufacturers.

Keywords: *Exergy destruction; parallel compressor; R744; transcritical refrigeration system.*

1. Introduction

The strict regulations aimed at protecting the environment have revived the interest in natural refrigerants. Carbon dioxide (R744) seems to be one of the best long-term alternatives to synthetic refrigerants, since it is an environmentally harmless (GWP = 1 $kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$) and safe working fluid (i.e. non-flammable and non-toxic), as well as it features both good thermo-physical properties [1] and a low cost. On the other hand, basic "CO_2 only" refrigerating systems perform much worse than hydro fluorocarbon (HFC)-based units at high outdoor temperatures. In fact, these running modes imply the occurrence of transcritical operations due to the low critical temperature of CO_2 (30.98 °C). Therefore, high temperature lifts and enormously depreciated performance are experienced on the part of the aforementioned systems in warm/hot climate conditions. This leads to the fact that the greatest amount of irreversibilities occurring in a transcritical R744 refrigeration plant are associated with the expansion valve, as proved by Falzelpour and Morosuk [2]. The authors also assessed that the subsequent adoption of an economizer allows increasing the global exergy efficiency by 7%, although no energy improvements can be observed. Gullo et al. [3] showed that the exergy destruction related to the expansion device can be halved by adopting the solution suggested in this study, involving an auxiliary compressor and a flash tank at intermediate pressure. This allows also increasing the Coefficient of Performance (COP) by 18.7% over a single-stage compression machine at cooling medium temperatures ranging from 30 °C to 50 °C. The adoption of an ejector rather than a conventional expansion valve brings the final cost of the product down to 24.4% in relation to a one-stage R744 cycle at high heat sink temperatures, as revealed by Gullo and Cortella [4].

Many researchers [5-15] demonstrated that the adoption of the state-of-the-art technologies permits R744 supermarket solutions to perform similarly to or better than conventional HFC configurations in any climate context. In particular, Sarkar and Agrawal [16] implemented a theoretical study on various parallel compression-based R744 refrigeration systems. The authors concluded that the solution investigated in this study is the most promising solution. According to Javerschek et al. [17] and Gullo et al. [8], the transcritical R744 refrigeration systems outfitted with parallel compression are defined as the current benchmark (as well as the 2nd generation) of "CO_2 only" refrigerating units for supermarket applications. As a consequence, many theoretical and experimental efforts are being devoted to the evaluation the performance of such a solution [10,16-24].

The real enhancements achievable by any energy system can be properly assessed through the application of advanced thermodynamic tools, like the advanced exergy analysis [25-27]. Such an evaluation allows estimating the actual preventable irreversibilities occurring in the investigated

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solution, as well as it permits the designer to adopt the most effective techniques to enhance its performance.

Morosuk et al. [28] applied an advanced exergy evaluation to a R717 Voorhees refrigeration machine. The results pointed out that close attention has to be given to the evaporator, whose enhancement would imply the improvement of both the heat exchanger itself and the other components. Gullo et al. [24] claimed that promising results can be obtained by applying the aforementioned analysis to "CO_2 only" refrigeration systems. The investigation by Bai et al. [29] suggested that 43.44% of the total irreversibilities occurring in an ejector expansion transcritical R744 refrigerating unit can be reduced by enhancing its components. Sarkar and Joshi [30] experimentally showed that close attention has to be paid to the compressor improvement to enhance the performance of a R744 heat pump system for simultaneous water cooling and heating.

In this study the advanced exergy analysis has been applied to a transcritical one-stage CO_2 refrigeration system equipped with auxiliary compression, being a promising HFC-free technology for food retail applications. The design outdoor temperature has been taken as 40 °C. Unlike the work currently available in the open literature [24], in this investigation the effect of the most influential operating parameters, such as the design outdoor temperature, the main compressor efficiency and the gas cooler approach temperature, on the performance of the evaluated system has also been studied. In Section 2 the methods including the investigated system description, the assumption made and the main concepts related to the conventional and advanced exergy analysis are presented, whereas all the results are summarized in the following section. Finally, the computed outcomes are discussed in Section 4 and the main conclusions are drawn in Section 5.

2. Methods

2.1 Investigated Solution

The investigated solution is schematized in Figure 1 and pointed out as PC. In comparison with a conventional one-stage "CO_2 only" refrigeration unit, this configuration presents an additional compressor (AUX) and a liquid receiver (REC). The former is used for removing the vapor generated by the high-pressure expansion valve (TV(HP)) from the REC. It is worth remarking that the higher the outdoor temperature, the more vapor is produced [5]. Two main benefits can be associated with this solution in relation to a conventional transcritical R744 unit, i.e. the reduction in the mass flow rate drawn by the main compressor (CM) and an increase in the refrigerating effect. The system also relies on an air-cooled gas cooler (GC), which rejects heat to surrounding air. As previously mentioned, this can take place in both subcritical and transcritical running modes, depending on the outdoor temperature.

Table 1. Operating conditions of the investigated solutions.

Approach temperature of GC	5	K
Cooling capacity	100	kW
Evaporating temperature	-10	°C
Internal superheating	5	K
Superheating in the suction lines	5	K
Pressure drop in GC and EV	1	bar

The main operating conditions are listed in Table 1, which were mainly based on the ones presented by Hafner et al. [31].

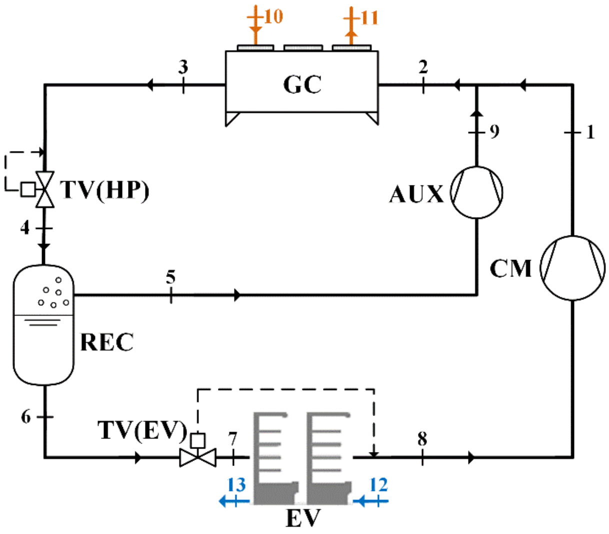

Figure 1. Schematic of a R744 refrigeration system with parallel compression economization (PC).

All the models were implemented by using Engineering Equation Solver (EES) (F-Chart Software, 2015) [32]. The global efficiency of the compressors (i.e. the ratio of the power input computed at isentropic conditions to the power input suggested by the manufacturer) was derived from Bitzer Software [33]. This was assessed with the aid of some correlations represented in the form of quadratic equations as shown by Eq. (1)

$$\eta_{glob} = a \cdot \beta^2 + b \cdot \beta + c \quad (1)$$

and listed as a function of the pressure ratio (β) in Table 2.

Table 2. Global efficiency of the selected compressors.

Compressor	a	b	c
CM	-0.0021	-0.0155	+0.7325
AUX	-0.0788	+0.3708	+0.2729

In accordance with Cavallini and Zilio [34], an optimal high pressure as a function of the gas cooler outlet temperature has to be identified in order to maximize the COP of a CO_2 machine operating in transcritical running modes. An optimization procedure similar to that implemented by Sawalha [35] was applied to the investigated system. As far as its intermediate pressure was concerned, it was set to 40 bar [31]. The reason for this lies in the fact that this value is typically adopted in order to have more stable expansion processes and low pressures inside the supermarket [36].

2.2 Exergy Analysis

2.2.1 Conventional Exergy Analysis

Exergy is the maximum useful work which can be computed after bringing the considered system into equilibrium with the surroundings by interacting only with this. The application of the exergy rate balance at steady state (Eq. (2) by [37]) to each component led to the calculation of the exergy destruction rates related to all the implemented evaluations. The kinetic, chemical and potential exergy variations were assumed negligible.

$$\sum_j \left(1 - \frac{T_0}{T_j}\right) \cdot \dot{Q}_j - \dot{W}_{CV} + \sum_{in} \dot{m}_{in} \cdot e_{in} - \sum_{out} \dot{m}_{out} \cdot e_{out} - \dot{E}_D = 0 \quad (2)$$

in which T_o refers to the temperature of the environment state. The exergy destruction ($\dot{E}_D$) represents the source of thermodynamic irreversibilities by means of which the inefficiencies in the investigated system can be estimated.

The exergy efficiency (η_{exergy}) of a refrigeration machine can be defined as:

$$\eta_{exergy} = 1 - \frac{\dot{E}_D + \dot{E}_L}{\dot{W}_{tot}} \quad (3)$$

in which the exergy losses ($\dot{E}_L$) are due to the interaction associated with transfers of matter, heat and work between the surroundings and the system.

By overlooking the kinetic, chemical and potential exergy changes, the physical exergy per unit of mass (e^{PH}) is equal to:

$$e^{PH} = [h(T,p) - h(T_0,p_0)] - T_0 \cdot [s(T,p) - s(T_0,p_0)] \quad (4)$$

in which the temperature T and the pressure p identify a generic thermodynamic state, whereas p_o denote the pressure of the environment state.

Some of the parameters employed for implementing both the conventional and advanced exergy analysis are summarized in Table 3. The air temperatures flowing through EV and that of the air leaving GC were respectively chosen the same as those presented by Ommen and Elmegaard [38] and 5 K higher than the outdoor temperature [24]. The GC pinch point temperature was assumed equal to the approach one [24], whereas the gas cooler pressure was kept constant in all the performed assessments [24]. Also, the intermediate pressure was not varied in the present work.

Table 3. Additional parameters.

Air pressure	1.01	Bar
Air temperature entering EV	5.00	°C
Air temperature leaving EV	-5.00	°C
Dead state temperature (T_0 in K/°C)	$T_{outdoor}$	K/°C
Dead state pressure (p_0)	1.01	Bar

As suggested by Ommen and Elmegaard [38], the outcomes of the exergy analysis are not substantially affected by the chosen dead state conditions.

2.2.2 Advanced Exergy Analysis

The conventional exergy analysis can assess neither the real improvement potential nor the mutual interdependencies among the components of the system under investigation. The implementation of the advanced exergy evaluation allows overcoming these drawbacks. In fact, the splitting of the exergy destruction rate associated with the *k-th* component into its avoidable and unavoidable fractions leads to a better understanding of the real enhancements that the selected system can achieve.

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{UN} + \dot{E}_{D,k}^{AV} \quad (5)$$

$\dot{E}_{D,k}^{UN}$ in Eq. (5) represents the irreversibilities associated with the *k-th* component which cannot be avoided because of manufacturing methods and the availability and cost of the materials [28, 39-40]. After the calculation of $\dot{E}_D$ for all the components belonging to the selected system through the application of the conventional exergy analysis, the corresponding unavoidable exergy destruction rates were computed by simulating a cycle in which the parameters in the second column of Table 4 were taken into account. Pressure drop in the unavoidable conditions was halved in comparison with the one selected for the conventional exergy analysis and the external superheating was neglected.

Table 4. Parameters used for the advanced exergy analysis [24].

Compressors	$\eta_{glob}^{UN} = 0.94$	$\eta_{glob}^{EN} = 1.00$
TV	-	$\eta_{glob}^{EN} = 1.00$
EV, GC	$\Delta T_{pp}^{UN} = 0.50\ K$	$\Delta T_{pp}^{EN} = 0.00\ K$

The endogenous exergy destruction occurring in the *k-th* component ($\dot{E}_{D,k}^{EN}$) can be calculated by implementing a thermodynamic cycle in which the component being considered is working in the actual running modes, whereas all the remaining ones are operating in the theoretical conditions. The exogenous exergy destruction within the *k-th* component ($\dot{E}_{D,k}^{EX}$) are caused by the inefficiencies occurring in the other components [28,39,41]. This is evaluated by subtracting $\dot{E}_{D,k}^{EN}$ from $\dot{E}_{D,k}$ as:

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX} \quad (6)$$

According to the third column of Table 4, the endogenous operating conditions for both the expansion valves and the compressors can be reached by replacing them with isentropic devices. As for the heat exchangers, a null value of ΔT_{pp}^{EN} for both EV and GC was chosen.

Further useful information can be collected by combining the previously mentioned splitting approaches, meaning that:

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{UN,EN} + \dot{E}_{D,k}^{UN,EX} + \dot{E}_{D,k}^{AV,EN} + \dot{E}_{D,k}^{AV,EX} \quad (7)$$

The avoidable endogenous ($\dot{E}_{D,k}^{AV,EN}$) and the avoidable exogenous exergy ($\dot{E}_{D,k}^{AV,EX}$) destruction rates of the *k-th* component can be respectively dropped by decreasing the inefficiencies associated with the components themselves and by improving the remaining components [28,39,41].

In this paper, $\dot{E}_D^{UN}$ and $\dot{E}_D^{UN,EN}$ in all the assessed cases were computed in the same way as Morosuk and Tsatsaronis [39], whereas the other fractions of the exergy destruction rate related to the *k-th* component were evaluated as follows:

$$\dot{E}_{D,k}^{UN,EX} = \dot{E}_{D,k}^{UN} + \dot{E}_{D,k}^{UN,EN} \quad (8)$$

$$\dot{E}_{D,k}^{AV,EN} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{UN,EN} \quad (9)$$

$$\dot{E}_{D,k}^{AV,EX} = \dot{E}_{D,k}^{EX} + \dot{E}_{D,k}^{UN,EX} \quad (10)$$

3. Results

3.1 Results of the Conventional Exergy Analysis

The main thermodynamic parameters for PC, such as temperature, pressure, mass flow rate, enthalpy, entropy and physical exergy per unit of mass, are summarized in Table 5 for the design outdoor temperature (T_0) of 40 °C.

Table 5. Thermodynamic parameters of PC calculated in real conditions.

State	Fluid	T [°C]	p [bar]	$\dot{m}$ [kg·s^{-1}]	h [kJ·kg^{-1}]	s [kJkg^{-1}K^{-1}]	e^{PH} [kJ·kg^{-1}]
1	R744	144.2	107.0	0.43	561.0	2.06	266.3
2	R744	117.4	107.0	0.97	524.5	1.97	258.1
3	R744	45.0	106.0	0.97	332.2	1.41	241.2
4	R744	5.3	40.0	0.97	332.2	1.47	222.6
5	R744	5.3	40.0	0.54	437.1	1.85	209.6
6	R744	5.3	40.0	0.43	213.3	1.05	237.4
7	R744	-10.0	26.49	0.43	213.3	1.06	234.5
8	R744	-5.0	25.49	0.43	444.4	1.94	188.9
9	R744	98.1	107.0	0.54	495.4	1.90	253.0
10	Air	40.0	1.01	37.22	313.5	6.91	0.0
11	Air	45.0	1.01	37.22	318.5	6.93	0.04
12	Air	5.0	1.01	9.94	278.3	6.79	2.13
13	Air	-5.0	1.01	9.94	268.2	6.75	3.60

The results of the conventional exergy analysis related to PC are plotted in Figure 2 for T_0 = 40 °C. It was possible to notice that, despite the presence of the intermediate pressure liquid receiver, TV(HP) was accountable for the highest contribution (i.e. 29.2%) to $\dot{E}_{D,tot}$ of the investigated system. Secondly, GC and CM respectively caused 24.3% and 22.1% of $\dot{E}_{D,tot}$. Close attention also had to be given to AUX and EV, since $\dot{E}_{D,AUX}$ and $\dot{E}_{D,EV}$ were equal to 13.0% and 8.1% of $\dot{E}_{D,tot}$. The inefficiencies related to the mixing point amounted to 0.74 kW. Furthermore, the exergy efficiency added up to 0.198 due to the fact that $\dot{W}_{tot}$ and $\dot{E}_L$ were respectively equal to 79.24 kW and 1.48 kW.

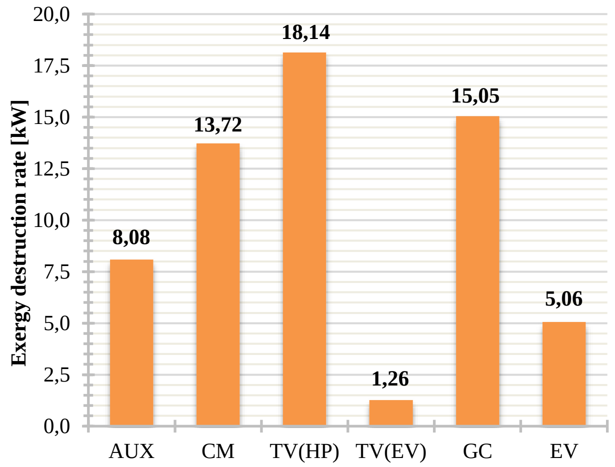

Figure 2. Results of the conventional exergy analysis associated with PC (T_0 = 40 °C).

3.2 Results of the Advanced Exergy Analysis

The outcomes of the advanced exergy analysis related to PC are shown in Figure 3. Over 59% of the total exergy destruction taking place in this configuration was preventable. In addition, the results revealed that the only application of the conventional exergy analysis would have led to enormously misleading results. In fact, the designer should firstly focus on CM, being accountable for 33.9% of $\dot{E}_{D,tot}^{AV}$. Secondly, significant enhancements were necessary to GC, AUX, TV(HP) and EV, as these components were respectively responsible for 21.4%, 20.1%, 16.7% and 6.7% of $\dot{E}_{D,tot}^{AV}$. Furthermore, in both the compressors about 90% of the total irreversibilities could be avoided, whereas only half of those associated with both GC and EV could be decreased. The following splitting of the irreversibilities into their avoidable endogenous and exogenous parts can define the strategies which have to be implemented to improve the considered solution. The overall system showed that about 46% of $\dot{E}_{D,tot}^{AV}$ was endogenous. Also, only one-third of the inefficiencies occurring in TV(HP) could be actually reduced uniquely by enhancing the remaining components. Furthermore, over 80% of $\dot{E}_D^{AV}$ taking place in CM and GC could be respectively decreased by improving the compressor itself and the other components. On the contrary, the enhancements achievable by AUX could be associated with both the improvement of the component itself and that of the remaining components. Finally, EV was improvable only by enhancing the heat exchanger itself in accordance with the results available in the open literature [27-28,39].

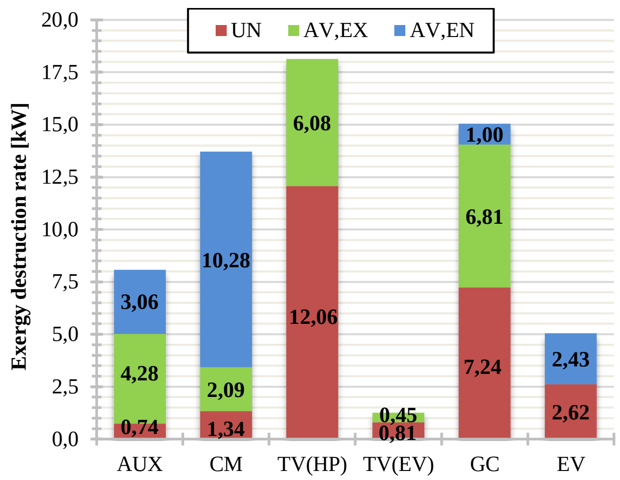

Figure 3. Results of the advanced exergy analysis associated with PC (T_0 = 40 °C).

3.3 Results of the Sensitivity Analysis

In this subsection a sensitivity analysis with respect to the design outdoor temperature (i.e. dead state temperature), the efficiency of the main compressor and the approach temperature of the gas cooler was implemented. In order to properly carry out such an assessment, the same assumptions made in Section 2 were adopted.

3.3.1 Sensitivity Analysis: Design Outdoor Temperature

Figure 4 depicts the effect of the design external temperature (i.e. dead state temperature or T_0) on the avoidable exergy destruction of the overall system. The avoidable irreversibilities associated with both EV and TV(EV) weakly ranged with respect to T_0. With rise in design outdoor temperature, their contribution to $\dot{E}_{D,tot}^{AV}$ in percentage terms became negligible. However, the avoidable exergy destruction rates related to AUX, CM, TV(HP) and GC went up as t_0 increased. AUX and TV(HP) showed an increasing trend in terms of percentage contribution to $\dot{E}_{D,tot}^{AV}$. On the other hand, this tendency was respectively diminishing and slightly variable when it came to CM and GC. It was important to remark that the designer firstly had to pay attention to CM ($\dot{E}^{AV}_{D,CM}$ = 42.3%-33.9% of $\dot{E}^{AV}_{D,tot}$), secondly to GC and then to AUX ($\dot{E}^{AV}_{D,AUX}$ = 12.3%-20.1% of $\dot{E}^{AV}_{D,tot}$) and TV(HP) ($\dot{E}^{AV}_{D,TV(HP)}$ = 10.5%-16.7% of $\dot{E}^{AV}_{D,tot}$) at $T_0 \leq 40$ °C. However, despite the presence of a receiver at intermediate pressure, TV(HP) presented the largest avoidable irreversibilities at $T_0 > 40$ °C (28.3% of $\dot{E}^{AV}_{D,tot}$), followed by AUX (26.4% of $\dot{E}^{AV}_{D,tot}$) and finally by CM (22.5% of $\dot{E}^{AV}_{D,tot}$) and GC (17.6% of $\dot{E}^{AV}_{D,tot}$). These results can be justified by taking into account that at high design outdoor temperatures:

- significant differences in pressure between the gas cooler and the receiver are reached;
- prominent mass flow rates are drawn by the auxiliary compressor.

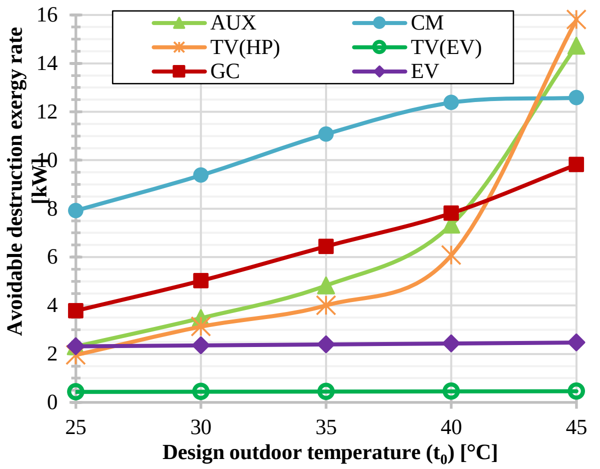

Figure 4. Influence of the design outdoor temperature (i.e. dead state temperature or T_0) on the avoidable exergy destruction rates of the components belonging to PC.

Figure 5 shows the endogenous and exogenous avoidable destruction exergy rates of AUX, CM and GC. As regards those related to the other components:

- the avoidable irreversibilities associated with both TV(HP) and TV(EV) are uniquely exogenous, as showed in Subsection 3.2;
- the avoidable irreversibilities related to EV are uniquely exogenous, as pointed out in Subsection 3.2.

The outcomes revealed that $\dot{E}^{AV,EX}_{D,AUX}$ was comparable to $\dot{E}^{AV,EN}_{D,AUX}$ at $T_0 \leq 40$ °C, whereas $\dot{E}^{AV,EX}_{D,AUX}$ became more noticeable than $\dot{E}^{AV,EN}_{D,AUX}$ at $T_0 > 45$ °C. On the other hand, $\dot{E}^{AV,EN}_{D,CM}$ represented from 78.1% (at $T_0 = 25$ °C) to 83.1% (at $T_0 = 45$ °C) of $\dot{E}^{AV}_{D,CM}$. As for GC, approximately 86% of its avoidable irreversibilities were exogenous independently from t_0.

It could be concluded that, in order to improve the performance of PC, the designer's priorities had to involve:

- the enhancement of CM, which could also lead to improvements for GC;
- the reduction in the approach temperature of the gas cooler so as to drop the irreversibilities of both TV(HP) and AUX.

These were particularly true at high cooling medium temperatures. The irreversibilities related PC could also be dropped by both recovering some expansion work, i.e. replacing TV(HP) with an ejector [4,7] and employing a more efficient auxiliary compressor [7].

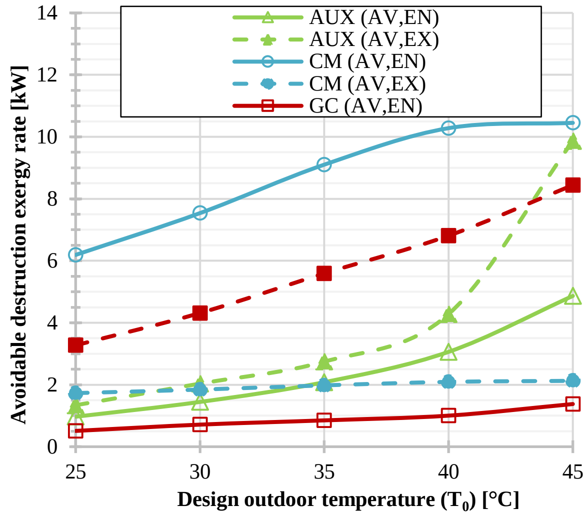

Figure 5. Influence of the design outdoor temperature (i.e. dead state temperature or T_0) on the endogenous and exogenous avoidable exergy destruction rates of the components belonging to PC.

3.3.2 Sensitivity Analysis: Efficiency of Main Compressor

The effect of the increase by 10% in CM compressor on the endogenous and exogenous avoidable exergy destruction rates of PC are shown in Figure 6. AUEV, TV(HP) and TV(EV) were not included as they were not affected by the aforementioned parameter substantially. By adopting this measure, $\dot{E}^{AV}_{D,tot}$ fell from 35.8% to 40.8% in comparison with the results presented in the previous subsection. In particular, thanks to the drop in the discharge temperature, $\dot{E}^{AV,EX}_{D,GC}$ and $\dot{E}^{AV,EN}_{D,CM}$ respectively decreased from 8.2% (at $T_0 = 45$ °C) to 14.9% (at $T_0 = 25$ °C) and from 27.1% (at $T_0 = 45$ °C) to 31% (at $T_0 = 25$ °C).

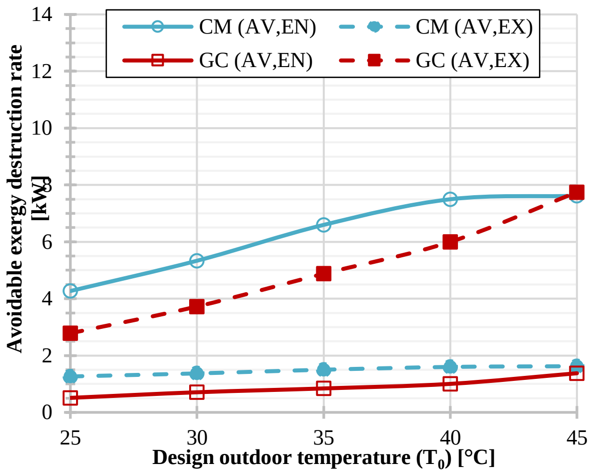

Figure 6. Influence of the increase by 10% of CM efficiency on the endogenous and exogenous avoidable exergy destruction rates of the components belonging to PC at different T_0.

3.3.3 Sensitivity Analysis: Approach Temperature of Gas Cooler

As depicted in Figure 7, the reduction in the approach temperature of GC led to drop in $\dot{E}_{D,tot}^{AV}$ from 8.5% (at T_0 = 25 °C) to 22.1% (at T_0 = 45 °C) in relation to the outcomes summarized in Subsection 3.3.1. Compared to these, the enhancements related to AUX were represented by drops in $\dot{E}_{D,AUX}^{AV,EX}$ ranging from 41% to 58.8% at the investigated range of outdoor temperatures. Great enhancements were quantified for both TV(HP) and GC. Since the approach temperature of GC was brought down, the discharge temperature of AUX was also decreased. As a consequence, $\dot{E}_{D,GC}^{AV,EX}$ reduced from 5% (T_0 = 25 °C) to 22.6% (T_0 = 45 °C). Furthermore, this permitted also lessening $\dot{E}_{D,GC}^{AV,EX}$ from 5% (T_0 = 25 °C) to 22.6% (T_0 = 45 °C). Finally, the largest reduction in irreversibilities was estimated for TV(HP). This was, at worst, equal to 59.5% and, at best, to 78.6% at design outdoor temperatures ranging from 25 °C to 45 °C.

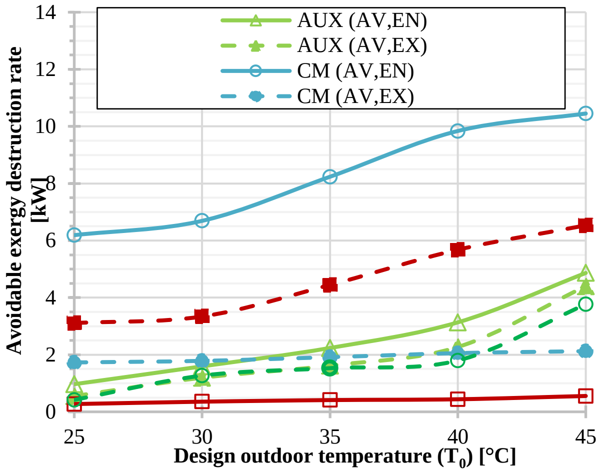

Figure 7. Influence of the approach temperature of GC on the endogenous and exogenous avoidable exergy destruction rates of the components belonging to PC at different T_0.

4. Discussions

The implementation of the advanced exergy analysis has shown the limitations related to the conventional exergy assessment to the investigated solution. At the design outdoor temperature of 40 °C, the results of the latter suggest paying close attention primarily to the high-pressure expansion valve (29.2% to $\dot{E}_{D,tot}$) and secondly to both the gas cooler (24.3% to $\dot{E}_{D,tot}$) and the main compressor (22.1% to $\dot{E}_{D,tot}$). The parallel compressor (13% to $\dot{E}_{D,tot}$) and the evaporator (8.1% to $\dot{E}_{D,tot}$) also contributes to the total irreversibilities significantly.

The application of the more sophisticated exergy evaluation to the selected systems points out that:

- the high-pressure expansion valve is actually accountable for 16.7% of $\dot{E}_{D,tot}^{AV}$. In fact, only one third of $\dot{E}_{D,TV(HP)}$ can be diminished by uniquely enhancing the remaining components;
- an enormous contribution (33.9%) to $\dot{E}_{D,tot}^{AV}$ on the part of the main compressor has been revealed. Such a component can be mainly enhanced by replacing it with a more efficient component, although the contribution of its exogenous avoidable irreversibilities to $\dot{E}_{D,CM}^{AV}$ is also noteworthy;
- the auxiliary compressor and the gas cooler cause about 20% of $\dot{E}_{D,tot}^{AV}$. In particular, only half of the irreversibilities occurring in the gas cooler can be actually decreased. In addition, its preventable inefficiencies are almost completely exogenous. As regards the auxiliary compressor, this component can be equally enhanced by improving the compressor itself and the other components;
- only half of the exergy destruction of the evaporator can be dropped. This result can be attained uniquely through the enhancement of the heat exchanger itself, accordingly with the results available in the open literature [27-28,39].

In accordance with the results by Bai et al. [29], the closest attention has to be addressed to the main compressor at outdoor temperatures up to 40 °C. However, different conclusions can be drawn when it comes to the other components. Also, unlike the ejector expansion transcritical CO_2 refrigeration system, the avoidable irreversibilities occurring in the investigated solution are mainly due to the other components.

Further valuable information can be derived from the implemented sensitivity analysis:

- at more severe operating conditions, the designer has to focus on both the high-pressure expansion valve and the auxiliary compressor. They are respectively improvable by uniquely and mainly the other components.
- reductions in $\dot{E}_{D,tot}^{AV}$ between 35.8% and 40.8% can be accomplished by increasing the efficiency of the main compressor by 10%. This result is mainly related to the enhancement of the gas cooler;
- the decrease in the approach temperature of the gas cooler allows reducing the irreversibilities from 8.5% (at T_0 = 25 °C) to 22.1% (at T_0 = 45 °C) over the investigated range of outdoor temperatures. This is mostly beneficial to the high-pressure expansion valve.

5. Conclusions

In this paper, the thermodynamic performance evaluation of a commercial R744 refrigeration system using parallel compression has been conducted. The typical running modes of a supermarket, which have involved the assumption of an evaporating temperature of -10 °C and a cooling capacity of 100 kW, have been chosen. Furthermore, the computed results have firstly been carried out at the design outdoor temperature of 40 °C. The global efficiency of the compressors has been derived from some manufacturer' selection software. At a later time, the implemented investigation has also involved a sensitivity analysis based on the design outdoor temperature, the main compressor efficiency and the gas cooler approach temperature.

It can be concluded that:

- the advanced exergy analysis leads to a better understanding of the design procedure to be implemented to the investigated solutions;
- the total avoidable exergy destruction rate is higher than the total unavoidable one, being in particular the exogenous avoidable irreversibilities slightly more relevant than the endogenous avoidable inefficiencies;
- the high-pressure expansion valve needs to be replaced with a device for expansion work recovery;
- enhancements in the efficiency of the main compressor should be seriously taken into account on the part of the manufacturers. This expedient is much more beneficial than the reduction in the approach temperature of the gas cooler;

- the gas cooler and the auxiliary compressor also have a substantial contribution to the total avoidable inefficiencies, especially with rise in outdoor temperature. The former is mainly improvable by enhancing the other components, whereas the latter can be improved through the enhancement of both the compressor itself and the other components;

Nomenclature

o	*Dead state*
AUX	*Auxiliary compressor*
AV	*Avoidable*
CM	*Main compressor*
compr	*Compressor*
COP	*Coefficient of Performance [-]*
CV	*Control volume*
D	*Destruction*
e	*Exergy per unit of mass* $[kJ\cdot kg^{-1}]$
$\dot{E}$	*Exergy rate [kW]*
EES	*Engineering Equation Solver*
EN	*Endogenous*
EV	*Evaporator*
EX	*Exogenous*
GC	*Air-cooled gas cooler*
glob	*Global*
GWP	*Global Warming Potential* $[kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}]$
h	*Enthalpy per unit of mass* $[kJ\cdot kg^{-1}]$
HFC	*Hydro fluorocarbon*
HP	*High-pressure*
in	*Inlet*
j	*j-th stream*
k	*k-th component*
L	*Loss*
$\dot{m}$	*Mass flow rate* $[kg\cdot s^{-1}]$
out	*Outlet*
p	*Pressure [bar]*
PC	CO_2 *refrigeration system with parallel compression*
PH	*Physical*
pp	*Pinch point*
$\dot{Q}$	*Heat transfer rate [kW]*
REC	*Liquid receiver*
s	*Entropy per unit of mass* $[kJ\cdot kg^{-1}\cdot K^{-1}]$
T	*Temperature [K]*
tot	*Total*
TV	*Throttling valve*
UN	*Unavoidable*
$\dot{W}$	*Power input [kW]*

Greek symbols

β	*Pressure ratio [-]*
Δ	*Difference*
η	*Efficiency [-]*

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