Energetic, Exergetic and Exergoeconomic Analysis of CO_2 Refrigeration Systems Operating in Hot Climates

Paride Gullo[a], Brian Elmegaard[b] and Giovanni Cortella[c]

[a] *University of Udine, Udine, Italy, gullo.paride@spes.uniud.it, CA*
[b] *DTU Technical University of Denmark, Kgs. Lyngby, Denmark, be@mek.dtu.dk*
[c] *University of Udine, Udine, Italy, giovanni.cortella@uniud.it*

Abstract:

Carbon dioxide is one of the most interesting refrigerants in terms of both environmental impact and efficiency at low outdoor temperatures. Its use in warm climates still needs for some improvements as well as some economic evaluations in order to analyse its real potential in such weather conditions. CO_2 refrigeration system with parallel compression represents one of the solutions which have been proposed in the last few years in order to enhance the performance of a single-stage refrigeration system.
The main target of this study is to compare the thermodynamic efficiency and the final cost of the product of a R744 refrigeration solution with auxiliary compressor with those of a R744 conventional system, both of them operating in transcritical conditions.
The results pointed out that the adoption of an auxiliary compressor resulted in an increase of the COP by approximately 18.7% over the investigated temperatures range. Furthermore, the final cost of the product associated with this solution was on average 6.7% lower than the one of the conventional solution.

Keywords:

COP, Exergoeconomic Analysis, Hot Climates, Parallel Compression, R744.

1. Introduction

CO_2 is getting more and more popular among refrigerants due to its almost negligible Global Warming Potential (GWP), non-flammability and non-toxicity. However, because of its low critical temperature, its performance is worse than that of a single-stage system using a synthetic refrigerant at high external temperatures. R744 transcritical refrigeration systems are characterized by an optimum high pressure as a function of the gas cooler outlet temperature, as explained in [1]. The interest in R744 has been involving a large number of researchers in the last decade in order to determine the most suitable configuration for employing in warm climates.

Sarkar and Agrawal [2] concluded that the system with parallel compression is more efficient than both the solution with parallel compression and subcooler and that with multi-stage compression and vapour by-pass at low evaporating pressures. They also showed that the optimum intermediate pressure is mainly affected by the evaporating temperature, while the latter does not affect the optimum gas cooler pressure strongly.

Minetto et al. [3] pointed out that a CO_2 refrigeration solution with auxiliary compressor has good Coefficient of Performance (COP) values and good refrigerating capacities in comparison with a conventional one. The authors also stated that the issue associated with the oil return can be worked out.

An optimization of two different solutions with vapour removal was fulfilled by Da Ros [4], who concluded that a two-stage system is characterized by better performance than the one with auxiliary compressor over the chosen operating range.

The results of the experimental campaign carried out by Chiarello et al. [5] proved the feasibility of the evaluated system. Sarkar and Agrawal [2], Minetto et al. [3] and Chiarello et al. [5] highlighted the importance of the intermediate pressure optimization in order to improve COPs of a R744 refrigeration system with auxiliary compressor.

Hafner et al. [6] concluded that a system with parallel compressor can improve the performance of a transcritical refrigeration machine over a standard booster in warm climates, but the adoption of an ejector is necessary at very high external temperatures. They also showed that the adoption of parallel compression allows the system to operate at lower high pressures than a conventional one, improving its performance.

A performance evaluation of different cycles was made by Minetto et al. [7] who showed that, in warm climates, the solution with auxiliary compressor has similar performance to that of the basic one with overfed evaporators.

Chesi et al. [8] carried out both an experimental evaluation and a theoretical analysis. The latter evaluated the parameters, such as separator efficiency and volumetric flow rate of the compressors, which influence the performance of the system. Furthermore, it allowed also identifying the best operating conditions of the system under investigation. From the experimental results point of view, liquid receiver efficiency, unintentional superheating and pressure drop can strongly worsen the performance of the real cycle.

Publications regarding the application of the exergoeconomic analysis to CO_2 refrigeration systems include the ones made by Rezayan and Behbahaninia [9], Fazelpour and Morosuk [10] and Aminyavari et al. [11]. Rezayan and Behbahaninia [9] implemented an optimization in terms of minimization of the annual costs of a cascade system using ammonia and carbon dioxide as refrigerants. Fazelpour and Morosuk [10] concluded that the throttling valve of a R744 transcritical refrigeration system is characterized by the highest exergy destruction rate and the adoption of an economizer leads to a lower total cost of the final product. On the other hand, it does not improve the performance of the system. An optimization by using TOPSIS decision-making method setting the total cost and the exergetic efficiency of a CO_2/NH_3 cascade as objective functions was carried out by Aminyavari et al. [11]. The evaluation also took into account the influence of the variation of the cooling capacity on the exergy destruction and that of the unit cost of electricity on the optimal conditions.

The goal of this paper is to compare the energetic efficiency and the cost effectiveness of a R744 refrigeration system with auxiliary compressor with the ones of a R744 conventional solution. The systems are intended for refrigerated foods and will operate in transcritical mode.

It is important to point out that, considering a system with parallel compression, the intermediate pressure becomes an additional key performance parameter, as previously mentioned. Displacement ratio of the auxiliary compressor to the main one and the variation of this parameter affect the optimum intermediate pressure value [3].

The following solutions have been evaluated:

- conventional R744 refrigeration system (Figure 1);
- R744 refrigeration system with parallel compression (Figure 2).

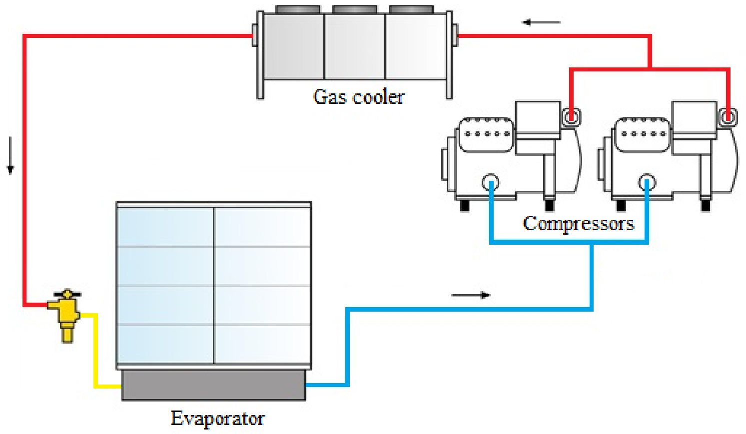

Fig. 1. Schematic of a conventional R744 refrigeration system.

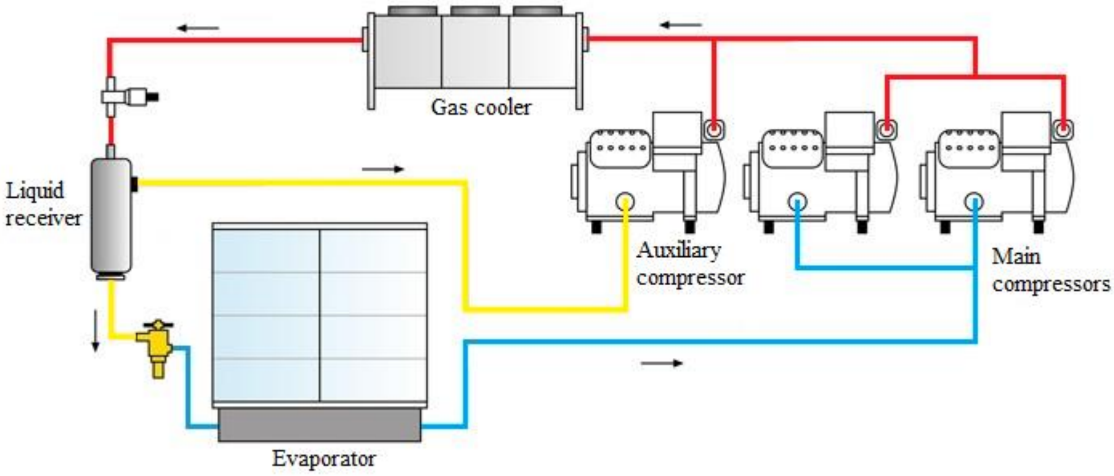

Fig. 2. Schematic of a R744 refrigeration system with parallel compression.

2. Methods

Numerical models of the cycles in steady state operation were developed based on the application of fundamental relations of thermodynamics, i.e. mass and energy balances. The cycles were modelled in design mode, which implies that actual component characteristics in off-design load were not taken into account.

The following assumptions were made in order to carry out the energetic analysis:

- cooling capacity equal to 100 kW [5];
- evaporating temperature equal to -10 °C [5];
- useful superheating within the evaporator equal to 5 K [5];
- superheating in the suction line of 5 K [5];
- isentropic efficiency of all compressors and efficiency of all electrical motors equal to 0.8 and 0.88, respectively [10];
- the cooling medium is unknown and thus it is only possible to evaluate its average temperature $t_{cm} = t_0 + 5$ °C [10];
- the secondary fluid is unknown and thus it is only possible to evaluate its average temperature $t_{sf} = 0$ °C;
- approach temperature of the gas cooler of 2 K [5];
- the pressure drop within the gas cooler and within the evaporator were assumed constant and equal to 3 bar and 0.75 bar, respectively [10];

- separation process was isobaric and it was assumed perfect;
- mixing process was isobaric;
- auxiliary consumption was neglected;
- the same procedure of heat exchangers design as in [10] was implemented.

The reference temperature t_0, which corresponds to the dead state temperature for the exergetic analysis and to the cooling medium inlet one for the energetic evaluation, was varied from 30 °C to 50 °C.

The analyses were carried out by using Engineering Equation Solver (EES) [12].

The gas cooler pressure of the basic solution was optimized as a function of t_0 by using the Golden Section search Method. As far as the optimization procedure of the other evaluated system is concerned, the Direct Algorithm Method was used assuming the intermediate pressure and the high one as optimization variables [13]. The target for both optimization procedures was to minimize the energy consumption.

The economic analysis was based on the following assumptions:

- the costs associated with the additional equipment and the cost of installation were set to 15% of the capital investment [10, 14];
- the operation and maintenance costs were not taken into account [10];
- the average cost of money was set to 10% [10, 14];
- the plant economic life was evaluated equal to 15 years [10, 14];
- the equations associated with the purchased equipment costs were the same as in [9];
- the purchased equipment cost (PEC) of all throttling valves and that of the liquid receiver were chosen equal to 100 € [10, 14] and 1000 € (from manufacturer catalogue), respectively;
- the global heat transfer coefficients of the gas cooler and of the evaporator were chosen equal to 180 $W \cdot (m^{-2} \cdot K^{-1})$ and 950 $W \cdot (m^{-2} \cdot K^{-1})$, respectively [10, 14];
- the cost of the electricity was assumed equal to 0.12 $€ \cdot (kW \cdot h)^{-1}$ [10, 14].

The economic analysis was conducted in the same way as in [10].

3. Results

In the following section, the results coming from the analyses previously mentioned are presented. Table A.1. and Table A.2. in Appendix A summarize the equations used in the simulation.

3.1. Results of the energetic analysis

Table 1 compares the two selected solutions in terms of Coefficient of Performance and exergetic efficiency. COP of the enhanced system is on average 18.7% higher than the one of the basic solution.

Table 1. Comparison in terms of COP and exergetic efficiency between the systems under investigation.

	CONVENTIONAL SYSTEM		IMPROVED SYSTEM	
t_0 [°C]	**COP**	**η_{ex}**	**COP**	**η_{ex}**
30	1.90	0.21	2.29	0.25
35	1.55	0.20	1.88	0.24
40	1.30	0.19	1.58	0.23
45	1.01	0.17	1.30	0.21
50	0.70	0.13	0.87	0.16

The solution with parallel compression has 17% and 21.8% higher COP values than the conventional one at the reference temperatures of 30 °C and 45 °C, respectively. Due to the large amount of vapour that has to be sucked at very high reference temperatures, the enhanced solution shows an increase in COP by 19.8% in comparison with that of the basic one at 50 °C. Since the cooling capacity was kept constant, a similar trend was also found in terms of compressors power input.

As previously mentioned, an additional benefit associated with the adoption of the parallel compression is the reduction in optimal high pressure. This result is highlighted by means of Figure 3. Due to the technological constraints in terms of maximum discharge pressure of the compressors, the difference in optimal gas cooler pressure decreases at t_0 equal to 40 °C and it becomes almost null at higher temperatures.

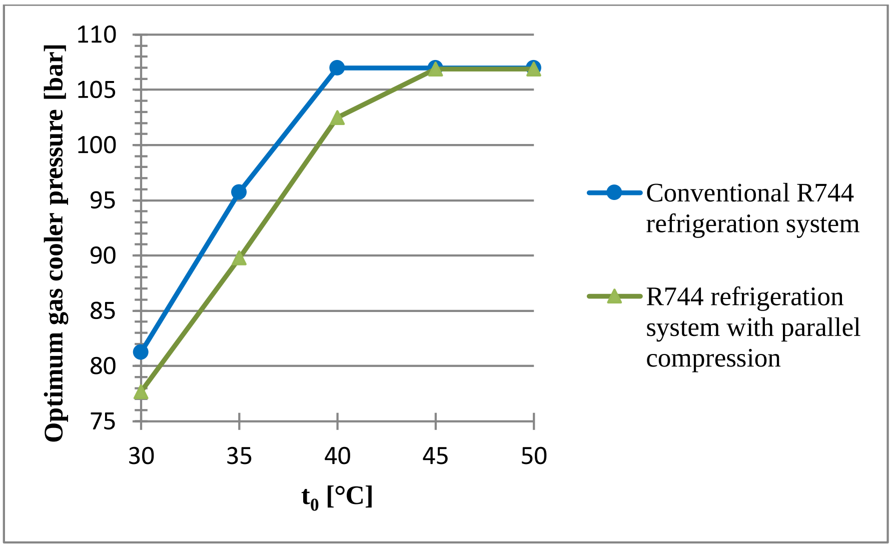

Fig. 3 Comparison in terms of optimum gas cooler pressure between the investigated solutions.

3.2. Results of the exergetic analysis

The exergetic analysis was carried out in accordance with [15]. The enhancements obtained by using a parallel compression can also be showed in terms of exergetic efficiency, which resemble the ones encountered for the coefficient of performance (Table 1).

A reduction in the sum between the exergy destruction and the exergy loss rates by 22.7% is achieved by employing the auxiliary compressor, as showed in Figure 4.

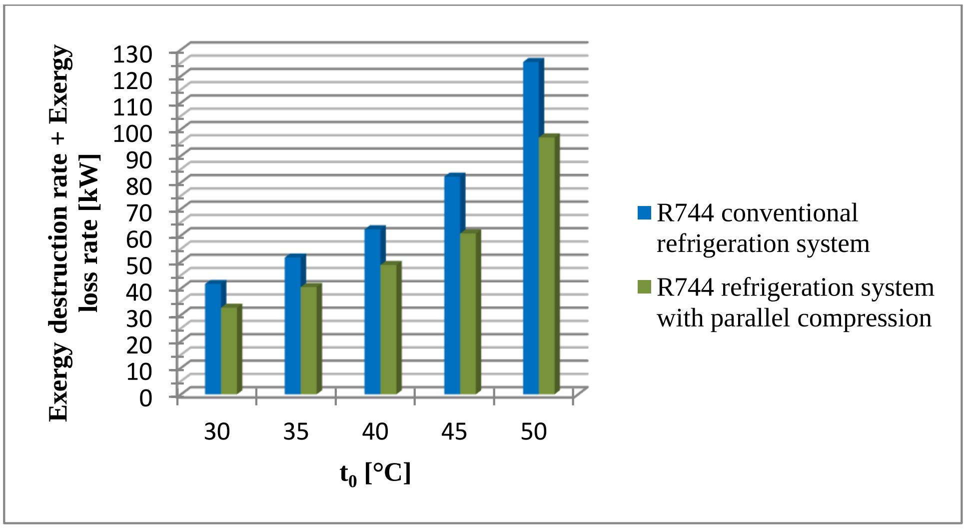

Fig. 4. Comparison in terms of total exergy destruction and exergy loss rates between the investigated solutions.

As mentioned in [10], the throttling valve is characterized by the highest exergy destruction rates in a transcritical R744 refrigeration machine. Figure 5 compares the exergy destruction rates of the throttling valve of the conventional system with the ones of the expansion valve upstream of the liquid receiver in the improved system. In the latter case, they are on average halved over the chosen operating range.

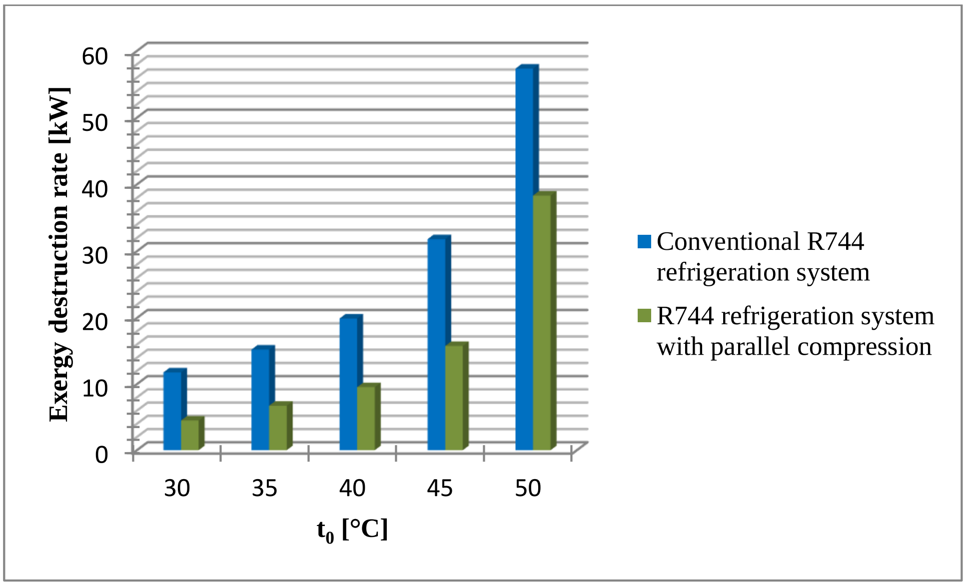

Fig. 5. Comparison in terms of exergy destruction rate associated with the throttling valves of the investigated systems.

3.3. Results of the economic analysis

The benefits in terms of energetic analysis clash with the results referring to the economic evaluation (Figure 6 - 7). The system with auxiliary compressor shows on average 23.4% higher total purchased equipment cost than that of the conventional solution. This result can mainly be attributed to the additional compressor in the improved system. In fact, compressors are usually characterized by the highest cost in a refrigeration system and thus the increase in their number leads to a growth in total purchased equipment cost.

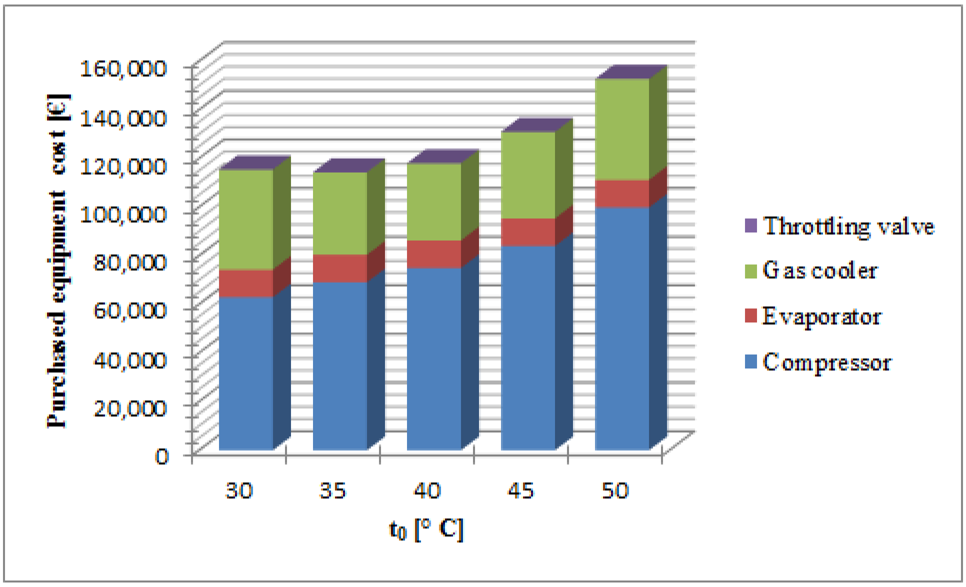

Fig. 6. Total purchased equipment cost of the conventional R744 refrigeration system.

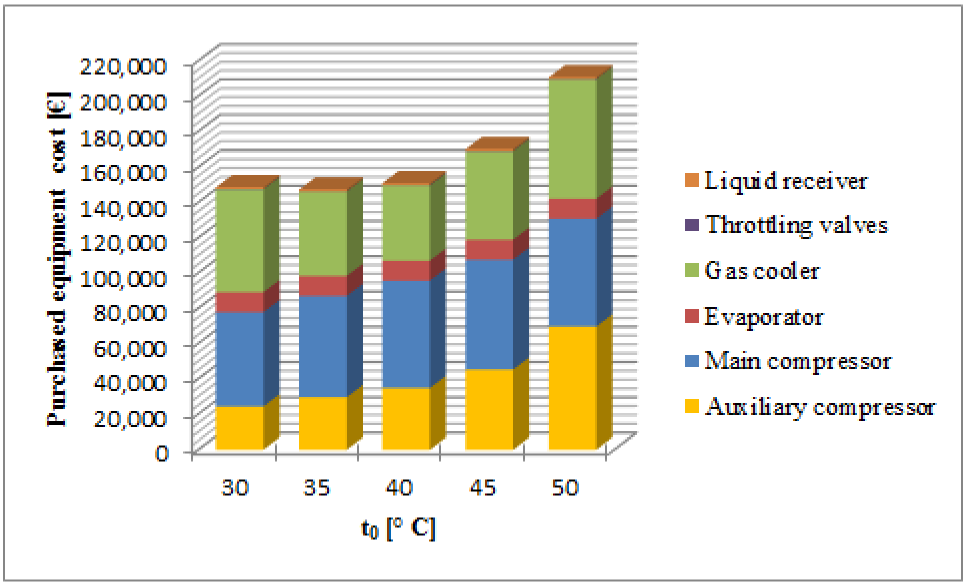

Fig. 7. Total purchased equipment cost of the R744 refrigeration system with parallel compression.

For the conventional system, PEC associated with the compressor has a growing trend over the investigated temperatures range, while that of the gas cooler decreases for reference temperatures up to 40 °C and then it starts increasing (Figure 6). This trend was also found in the enhanced system (Figure 7). Furthermore, in the latter case, PEC associated with the main compressor grows for reference temperatures up to 45 °C and then it drops. The cost of the auxiliary compressor has a growing trend and it shows the highest PEC value of the overall improved system at 50 °C.

3.4. Results of the exergoeconomic analysis

The exergoeconomic analysis was fulfilled in accordance with [15] in order to compare the cost effectiveness of the chosen systems.

Figure 8 shows that the adoption of an additional compressor leads to a drop in the total cost of the final product by 6.7% in comparison with that of the single-stage solution. The difference ranges from 2.6% to 10.5% for reference temperatures up to 45 °C, becoming equal to 9.3% at 50 °C.

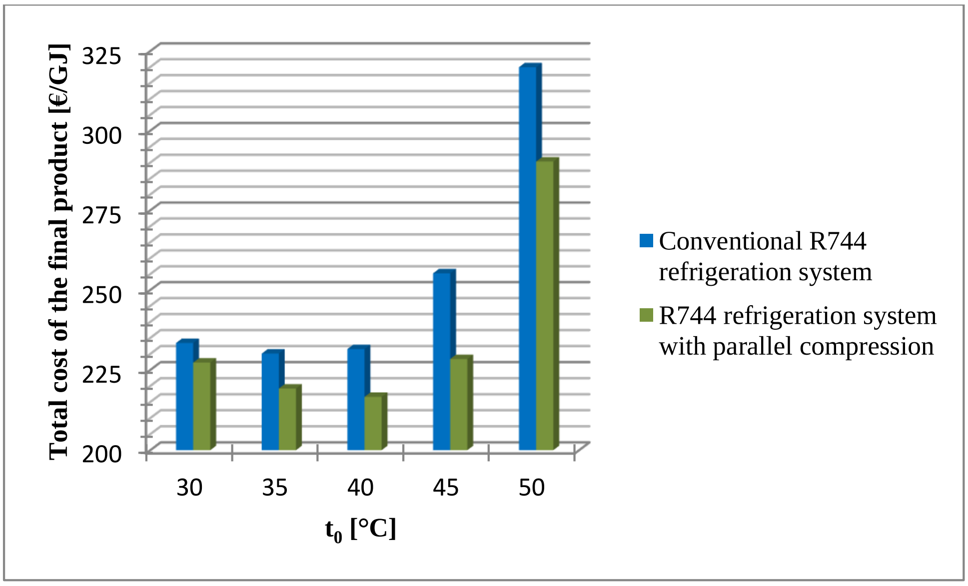

Fig. 8. Comparison in terms of total cost of the final product between the investigated solutions.

4. Discussion

The results obtained suggest that the solution with auxiliary compressor can improve the performance of a conventional system achieving an increase in COP on average by 18.7% and a drop in the total cost of the final product on average by 6.7%. From the economic point of view, the enhanced solution leads to an increase in total purchased equipment cost on average equal to 23.4%. Because of the latter outcome, the adoption of the previously mentioned solution in hot climates is mainly justifiable with a view to both the achievable energy saving over the plant economic life and to the progressive mandatory replacement of the refrigerants with high GWP with natural ones, imposed by European Union [16]. Therefore, an annual energy consumption analysis and a following evaluation in terms of payback period between the selected systems could further bear out its use.

Due to the different assumptions adopted, a direct comparison with the results obtained in [10] cannot be made.

5. Conclusions

In this paper, the energetic, exergetic and exergoeconomic analysis of two different R744 transcritical refrigeration systems have been carried out. The performance of a simple system has been optimized as a function of the gas cooler outlet temperature, while the optimization procedure of the improved system has also involved the intermediate pressure. Both the energetic and the exergetic analysis confirm that the installation of an auxiliary compressor can improve the performance in comparison with that of the basic cycle. The improvements vary from 17% to 21.8% for reference temperatures ranging from 30 to 50 °C. Furthermore, it allows dropping the exergy destruction rates associated with the expansion valve on average by 51%.

From the economic point of view, the system with parallel compression is characterized by higher total purchased equipment cost over the chosen operating range. The reason of this result lies in the adoption of an additional compressor, which is usually the most expensive component in a refrigeration system.

The exergoeconomic analysis reveals that the final cost of the product of the improved system is on average 6.7% lower than that of the basic solution. The trend of this reduction is growing for dead state temperatures up to 45 °C and then it decreases.

It can be concluded that this result, along with that obtained in terms of COP, confirm that the adoption of an auxiliary compressor is more beneficial for reference temperatures up to 45 °C even though it leads to an increase in total purchased equipment cost. On the other hand, the adoption of an auxiliary compressor can lead to a reduction in indirect CO_2 emissions.

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Appendix A

Table A.1. Data for the energetic, exergetic and exergoeconomic analysis of the conventional R744 refrigeration system.

Component	Energetic analysis	Exergetic analysis	Exergoeconomic analysis
Compressor	$\dot{W}_{compr} = \dot{m}_{refr} \cdot (h_{out,compr} - h_{in,compr})$	$\dot{E}_{D,compr} = \dot{W}_{compr} - (\dot{E}_{out,compr} - \dot{E}_{in,compr})$	$\dot{C}_{out,compr} = c_{el} \cdot \dot{W}_{compr} + \dot{C}_{in,compr} + \dot{Z}_{compr}$
Gas cooler	$t_{out,gc} = t_o + 2$ $\dot{Q}_{gc} = \dot{m}_{refr} \cdot (h_{out,compr} - h_{out,gc})$	$\dot{E}_{D,gc} = \dot{m}_{refr} \cdot (\dot{E}_{out,compr} - \dot{E}_{out,gc}) - \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)$	$\dot{C}_{P,gc} + \dot{C}_{out,gc} = \dot{C}_{out,compr} + \dot{Z}_{gc}$ $\dot{C}_{P,gc} = c_{P,gc} \cdot \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)$ $c_{out,gc} = c_{out,compr}$
Expansion valve	$h_{out,gc} = h_{in,evap}$	$\dot{E}_{D,ev} = \dot{E}_{out,gc} - \dot{E}_{in,evap}$	$\dot{C}_{in,evap} = \dot{C}_{out,gc} + \dot{Z}_{ev}$
Evaporator	$\dot{Q}_{evap} = \dot{m}_{refr} \cdot (h_{in,compr} - h_{in,evap})$	$\dot{E}_{D,evap} = \dot{m}_{refr}(\dot{E}_{in,compr} - \dot{E}_{in,evap}) + \dot{Q}_{evap}\left(1 - \frac{T_0}{T_{sf}}\right)$	$\dot{C}_{P,evap} + \dot{C}_{in,compr} = \dot{C}_{in,evap} + \dot{Z}_{evap}$ $\dot{C}_{P,evap} = c_{P,evap} \cdot \dot{Q}_{evap} \cdot \left\lvert\left(1 - \frac{T_0}{T_{sf}}\right)\right\rvert$ $c_{in,evap} = c_{in,compr}$
Overall system	$COP = \frac{\dot{Q}_{evap}}{\dot{W}_{compr}}$	$\eta_{ex} = 1 - \frac{\dot{E}_{D,tot} + \dot{E}_{L,tot}}{\dot{W}_{compr}}$ $\dot{E}_{L,tot} = \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)$	$c_{P,tot} = \frac{\dot{C}_{P,evap} + \dot{C}_{P,gc}}{\dot{Q}_{evap} \cdot \left\lvert\left(1 - \frac{T_0}{T_{sf}}\right)\right\rvert}$

Table A.2. Data for the energetic, exergetic and exergoeconomic analysis of the R744 refrigeration system with parallel compression.

Component	Energetic analysis	Exergetic analysis	Exergoeconomic analysis
Main compressor	$\dot{W}_{main} = \dot{m}_{refr,main} \cdot (h_{out,main} - h_{in,main})$	$\dot{E}_{D,main} = \dot{W}_{main} - (\dot{E}_{out,main} - \dot{E}_{in,main})$	$\boldsymbol{\dot{C}_{out,main} = c_{el} \cdot \dot{W}_{compr} + \dot{C}_{in,main} + \dot{Z}_{main}}$
Gas cooler inlet	$\dot{m}_{refr,main} h_{out,main} + \dot{m}_{refr,aux} h_{out,aux} = \dot{m}_{refr,tot} h_{in,gc}$	$\dot{E}_{D,in,gc} = \dot{m}_{refr,main}\dot{E}_{out,main} + \dot{m}_{refr,aux}\dot{E}_{out,aux} - \dot{m}_{refr,tot}\dot{E}_{in,gc}$	$\boldsymbol{\dot{C}_{in,gc} = \dot{C}_{out,main} + \dot{C}_{out,aux}}$
Gas cooler	$t_{out,gc} = t_o + 2$ $\dot{Q}_{gc} = \dot{m}_{refr,tot} \cdot (h_{in,gc} - h_{out,gc})$	$\dot{E}_{D,gc} = \dot{m}_{refr} \cdot (\dot{E}_{in,gc} - \dot{E}_{out,gc}) - \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)$	$\boldsymbol{\dot{C}_{P,gc} + \dot{C}_{out,gc} = \dot{C}_{in,gc} + \dot{Z}_{gc}}$ $\boldsymbol{\dot{C}_{P,gc} = c_{P,gc} \cdot \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)}$ $\boldsymbol{c_{out,gc} = c_{in,gc}}$
Expansion valve (upstream of liquid receiver)	$h_{out,gc} = h_{in,lr}$	$\dot{E}_{D,ev,HP} = \dot{E}_{out,gc} - \dot{E}_{in,lr}$	$\boldsymbol{\dot{C}_{in,lr} = \dot{C}_{out,gc} + \dot{Z}_{ev,HP}}$
Liquid receiver	$\dot{m}_{refr,main} h_{sat,L} + \dot{m}_{refr,aux} h_{in,AUX} = \dot{m}_{refr,tot} h_{out,gc}$	-	$\boldsymbol{\dot{C}_{in,aux} + \dot{C}_{out,sat,L} = \dot{C}_{out,gc} + \dot{Z}_{lr}}$ $\boldsymbol{c_{in,aux} = c_{out,sat,L}}$
Expansion valve (downstream of liquid receiver)	$h_{sat,L} = h_{in,evap}$	$\dot{E}_{D,ev,LP} = \dot{E}_{sat,L} - \dot{E}_{in,evap}$	$\boldsymbol{\dot{C}_{in,evap} = \dot{C}_{sat,L} + \dot{Z}_{ev,LP}}$
Evaporator	$\dot{Q}_{evap} = \dot{m}_{refr,main} \cdot (h_{in,main} - h_{in,evap})$	$\dot{E}_{D,evap} = \dot{m}_{refr.main}(\dot{E}_{in,main} - \dot{E}_{in,evap}) + \dot{Q}_{evap}\left(1 - \frac{T_0}{T_{sf}}\right)$	$\boldsymbol{\dot{C}_{P,evap} + \dot{C}_{in,main} = \dot{C}_{in,evap} + \dot{Z}_{evap}}$ $\boldsymbol{\dot{C}_{P,evap} = c_{P,evap} \cdot \dot{Q}_{evap} \cdot \left\lvert\left(1 - \frac{T_0}{T_{sf}}\right)\right\rvert}$ $\boldsymbol{c_{in,evap} = c_{in,main}}$
Auxiliary compressor	$\dot{W}_{aux} = \dot{m}_{refr,aux} \cdot (h_{out,aux} - h_{in,aux})$	$\dot{E}_{D,aux} = \dot{W}_{aux} - (\dot{E}_{out,aux} - \dot{E}_{in,aux})$	$\boldsymbol{\dot{C}_{out,aux} = c_{el} \cdot \dot{W}_{aux} + \dot{C}_{in,aux} + \dot{Z}_{aux}}$
Overall system	$\boldsymbol{\dot{m}_{refr,main} + \dot{m}_{refr,aux} = \dot{m}_{refr,tot}}$ $\boldsymbol{COP = \frac{\dot{Q}_{evap}}{\dot{W}_{compr} + \dot{W}_{aux}}}$	$\boldsymbol{\eta_{ex} = 1 - \frac{\dot{E}_{D,tot} + \dot{E}_{L,tot}}{\dot{W}_{compr} + \dot{W}_{aux}}}$ $\boldsymbol{\dot{E}_{L,tot} = \dot{Q}_{gc} \cdot \left(1 - \frac{T_0}{T_{cm}}\right)}$	$\boldsymbol{c_{P,tot} = \frac{\dot{C}_{P,evap} + \dot{C}_{P,gc}}{\dot{Q}_{evap} \cdot \left\lvert\left(1 - \frac{T_0}{T_{sf}}\right)\right\rvert}}$

Nomenclature

$\dot{C}$	Cost rate associated with an exergy stream, €·s^{-1}
c_{el}	Cost per unit of electricity, €·kJ^{-1}
$\dot{C}_L$	Cost rate associated with exergy loss, €·s^{-1}
c	Cost per unit of exergy, €·kJ^{-1}
COP	Coefficient of Performance
$\dot{E}$	Exergy rate, kW
$\dot{E}_L$	Exergy loss rate, kW
EES	Engineering Equation Solver
GWP	Global Warming Potential
h	Enthalpy per unit of mass, kJ·kg^{-1}
$\dot{m}$	Mass flow rate, kg·s^{-1}
PEC	Purchased Equipment Cost, €
$\dot{Q}$	Heat rate, kW
t	Temperature, °C
$\dot{W}$	Power input, kW
$\dot{Z}$	Cost rate associated with capital investments, €·s^{-1}

Subscripts and superscripts

0	Reference state for the exergetic analysis
aux	Auxiliary compressor
cm	Cooling medium
compr	Compressor
D	Destruction
ex	Exergetic
ev	Expansion valve
evap	Evaporator
gc	Gas cooler
HP	High Pressure
in	Inlet
L	Liquid
LP	Low Pressure
lr	Liquid receiver
main	Main compressor
out	Outlet
P	Exergy of product
refr	Refrigerant
sf	Secondary fluid
sat	Saturated
tot	Total

Greek symbols

η	Efficiency

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