Comparative Exergoeconomic Analysis of Various Transcritical R744 Commercial Refrigeration Systems

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Abstract:
The interest in carbon dioxide (R744) as a refrigerant for supermarket applications has been remarkably rising thanks to its favourable thermo-physical, safety and environmental properties. Furthermore, R744 refrigerating plants are still prone to accomplishing large improvements in terms of cost-effectiveness, capability of heat recovery and energy efficiency.

The main target of this paper is to compare the final cost of the product of different enhanced CO\textsubscript{2} refrigeration supermarket configurations, with a marked emphasis on the one employing a two-phase ejector as an expansion device. The outcomes showed that the use of such technology leads to a drop in the final cost of the product at least by 22.7\% over a basic single-stage solution for cooling medium temperatures ranging from 30 °C to 42 °C. The assessment was based on the assumption of three different values of the purchased equipment cost associated with the ejector. The efficiency of the diffuser was observed to influence the cost-effectiveness of the system quite substantially. It could be concluded that the higher the heat sink design temperature, the more cost-effective an ejector as a refrigerant expander is.

Keywords:
CO\textsubscript{2}, Cost-effectiveness, Parallel Compression, Supermarket, Thermoeconomics, Transcritical Ejector Refrigeration Cycle.

1. Introduction

The challenge represented by the lowering of the so-called “CO\textsubscript{2} equator” has been involving many researchers in the last few years. This efficiency limit distinguishes the locations in which the use of the R744 is energetically acceptable, i.e. the ones with an annual average temperature below 15 °C, from the places where HFC-based solutions perform much better. The deterioration in the performance of the R744 refrigerating machines can be attributed to the low critical temperature (30.98 °C) of the CO\textsubscript{2}, which causes the appearance of the transcritical operating conditions. Although in these running modes the high pressure can be optimized as a function of the gas cooler outlet temperature \cite{1} to maximize the Coefficient of Performance (COP), large temperature lifts are reached. Several reasons can justify the interest in the use of the carbon dioxide as a refrigerant in warm climates. First of all, the R744 is cheap and characterized by advantageous thermo-physical characteristics, besides being one of the few long-term replacements to environmentally harmful refrigerants thanks to its negligible Global Warming Potential (GWP), non-flammability and non-toxicity. Furthermore, the use of natural working fluids has been being considerably fostered by both the European Union, by means of the issuing of strict regulations on the subject of the environment preservation, and some countries, by way of both taxes on the HFCs purchase and the adoption of eco-friendly technologies. Different layouts derived from the basic R744 configuration have been currently suggested \cite{2-5}, demonstrating that similar energy saving to the one performed by the systems employing high-GWP refrigerants can be achieved even in mild/warm climates. The spread of the R744 in locations with high outdoor temperatures for a long period of time over the year could also be promoted by implementing evaluations based on advanced thermodynamic methods. In \cite{6}, the authors proved that the expansion valve is accountable for the highest total cost in a one-stage
transcritical CO₂ cycle. The use of an economizer leads to an increment in the purchased equipment cost (PEC) by 4-5% and to a decrement in the total cost of the final product by 13-14%. On the other hand, insignificant energetic improvements were accomplished. Gullo et al. [7] implemented a comparative thermodynamic analysis between a basic transcritical CO₂ refrigerating plant and the one including an auxiliary compressor. In comparison with the conventional cycle, the latter exhibited a reduction in the power input by 18.7% and a decrease in the irreversibilities taking place in the throttling valve by 50%, respectively. An increase in the total PEC by 23.4% and a decrement by 6.7% of the final cost of the product were also observed for cooling medium temperatures ranging from 30 to 50 °C. An ejector can take over the expansion valve in order to drop the irreversibilities occurring in the system and rise the COP. It is a well-known fact that any throttling valve operates (approximately) as an isenthalpic device. Such thermodynamic process causes a reduction in the refrigerating effect since the refrigerant kinetic energy generated by the pressure drop is converted into friction. In order to recover some of such energetic content, either an expander or an ejector can be employed. Since the former is expensive and easily damageable due to presence of a large amount of liquid, the researchers’ attention has been considerably focusing on the ejectors thanks to its cheapness, absence of moving parts and ability to handle two-phase flows without no damage. Kornhauser [8] implemented a one-dimensional model of a constant-pressure mixing ejector in which the conventional components performed ideally. The outcomes underlined that the efficiency of the diffuser has the highest influence on the increase in COP. Furthermore, an optimal mixing pressure, which can be ascribed to a tradeoff between mixing process irreversibilities and the inefficiencies occurring in the nozzle and in the diffuser, has to be identified. The use of such device as an expander in CO₂ refrigerating cycles has been comprehensively investigated [9-11] from both the energetic perspective and the exergetic one. Bilir and Ersoy [12] claimed that this technology is particularly promising for refrigerating plants running in tropical countries and desert area. Hafner et al. [13] showed that high potentials of enhancement can be attained on the part of a transcritical ejector refrigeration cycle operating at high outdoor temperatures. The replacement of the expansion valve with a two-phase ejector has experimentally driven to performance improvements varying from 7% [14] to 26% [15] over a basic R744 solution. A R600a refrigeration system using an ejector as a substitute of the compressor was optimized by Chen et al. [16] as a function of the pinch point temperatures of the generator, the condenser and the evaporator. The sum of the costs associated with the brine, the steam, the electricity, the capital cost and the operation and maintenance expense was employed as the objective function to be minimized. The results highlighted a reduction in the latter by 7.5-8.1% in comparison with the non-optimized cases.

The main target of this study is to compare the cost-effectiveness of a transcritical ejector refrigerating cycle with that of both the basic configuration and two configurations with parallel compression for temperatures of the cooling medium varying from 30 °C to 42 °C. Furthermore, the energetic enhancements and the influence of the diffuser efficiency on the final cost of the product have also been quantified. All the investigated solutions have been supposed to be for commercial refrigeration applications.

1.1 Investigated solutions

Fig. 1a depicts a CO₂ refrigeration system which employs the so-called “parallel compression” (PCRC), i.e. an auxiliary compressor which is used to draw the vapour generated in the flash tank due to the high-pressure (HP) expansion valve. The layout sketched in Fig. 1b features the presence of a subcooler downstream of the gas cooler (PCSRC). The refrigerant leaving the latter is split into two flows and either of them is vaporized to cool down the other part to promote an increase in the refrigerating effect. The saturated vapour at intermediate pressures continues to be cooled and then compressed to the high pressure.

Fig. 2 refers to a typical schematic of a transcritical R744 refrigerating plant using an ejector (TERC) as a substitute for a conventional expansion device with the purpose of reducing its irreversibilities. The refrigerant exiting the gas cooler, defined as the primary fluid, is expanded and accelerated
through the motive nozzle. Due to the pressure difference between the expanded stream and the refrigerant leaving the evaporator (secondary fluid), the low-pressure fluid is entrained into the suction nozzle. Both streams are then blended in the mixing section and a part of the remaining kinetic energy of the refrigerant is converted into a pressure increase through the diffuser. Two main energetic benefits can be associated with the adoption of an ejector as a refrigerant expander: (i) increase in the refrigerating effect since the working fluid goes into the evaporator at almost saturated conditions; (ii) reduction in the compressor power input as the refrigerant is compressed from an intermediate pressure to the high one.

A conventional single-stage CO\textsubscript{2} cycle was chosen as the baseline (CRC). The operating conditions of the investigated solutions were based on the ones presented in [13]. The cooling capacity was chosen equal to 100 kW, while the pressure drop of all the heat exchangers added up to 1 bar. The evaporating temperature was assumed to be -10 °C for CRC, PCRC and PCSRC, whereas the one related to TERC was increased by 3 K to consider the advantages associated with the use of an overfed evaporator. In the latter, in fact, a large amount of liquid flows through it and the refrigerant leaving
the heat exchanger has a high quality. Consequently, the degree of superheating (which is usually imposed) vanishes, the refrigerant-side heat transfer enhances and the evaporating temperature can be risen. The approach temperature of the gas cooler was supposed to be equal to 5 K. As for the effectiveness of the subcooler, it was set to 0.7. A constant pressure of 40 bar was selected for the flash tank in PCRC, whereas the one related to the separator in TERC was optimized according to the operating conditions of the ejector. The model of the latter was based on the Kornhauser’s iteration method [8]. The energetic, economic and cost-effectiveness assessments were carried out by setting the efficiencies of the ejector components to 0.8 [14]. At a later time, the influence of the diffuser efficiency on the final cost of the product was also taken into account. The upper and the lower limit of the discharge pressure were respectively imposed equal to 75 bar and 135 bar, whereas the maximum phase separator pressure was chosen equal to 60 bar. The degree of superheating was set to 5 K, while a global efficiency of all the compressors of 0.65 was assumed. All the investigated solutions were optimized as a function of the discharge pressure. The intermediate pressure for PCSRC and the mixing pressure for TERC became additional independent variables for their corresponding optimization procedures. Simulation models of the cycles were implemented in Engineering Equation Solver (EES) [17]. All the components were considered well-insulated and the expansion valves were supposed to perform an isenthalpic process.

2. Methods

2.1. Energetic and exergetic analyses

Although the main goal of this paper is to assess the cost-effectiveness of the investigated transcritical R744 refrigerating plants, the energetic benefit associated with each of them was firstly quantified by computing their COP. At a later time, an exergetic evaluation was implemented in order to be able to carry out the exergoeconomic analysis. Exergy is the maximum useful work which can be obtained after bringing the investigated system into equilibrium with the surroundings. The exergetic analysis allows calculating and detecting the inefficiencies occurring in the selected system [18]. For a given control volume, the exergy rate balance at steady state [19] was applied as follows:

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

The kinetics, chemical and potential exergy contributions were assumed negligible. The dead state was represented by setting the pressure $p_0$ and the temperature $T_0$ to 1 bar and to the cooling medium inlet temperature [6], respectively. Furthermore, the cooling medium and the secondary fluid were assumed unknown [6] and therefore only their average temperature could be defined: $T_{cm} = T_0 + 2.5$ K, $T_{sm} = T_{out,EV} + 5$ K for the conventional evaporators and $T_{sm} = T_{EV} + 5$ K for the overfed evaporator.

2.2. Economic and exergoeconomic analyses

The economic assessment was grounded on a simplified total revenue requirement method [18]. The total capital investment (TCI) was calculated as the product of the purchased equipment cost and the factor ($\gamma$) taking into account the expenditure related to pipes, control system, etc.

$$TCI = \gamma \cdot \sum_n PEC_n = = 1.15 \cdot \sum_n PEC_n$$  \hspace{1cm} (2)

The capital recovery factor (CRF) and the levelized carrying cost (CCL) were respectively computed by means of (3) and (4):

$$CRF = \frac{i (1+i)^n}{(1+i)^n - 1}$$  \hspace{1cm} (3)
\[
CC_L = TCI \cdot CRF
\]  

(4)

where \(i\) and \(n\) refer to the annual effective interest rate (10% [6]) and the plant economic life (15 years [6]), respectively.

The following assumptions were made to implement the economic analysis:

- the purchased equipment cost of all the selected compressors was computed as \(PEC_{COM} = 10167.5 \cdot W_{COM}^{0.46} [6]\);
- the overall heat transfer coefficient of the gas cooler and that of the evaporator were assumed equal to 180 W·m\(^{-2}\)·K\(^{-1}\) and to 950 W·m\(^{-2}\)·K\(^{-1}\) [6], respectively. The cost of all the heat exchangers was evaluated as \(PEC_{HX} = 1397 \cdot A_{HX}^{0.89} [6]\);
- the PEC associate with all the expansion valve was chosen equal to 100 € [6];
- the cost of the flash tank and the one related to the phase separator were selected equal to 2500 € and 3000 € (from a manufacturer’s catalogue), respectively;
- the operating and maintenance cost was not considered [6].

To the best of the authors’ knowledge, a suitable cost function for any ejector is still difficult to be found. Therefore, three scenarios involving the values of \(PEC_{ejec}\) of 600 €, 1300 € and 2000 € were taken into account.

The exergoeconomic analysis allows assessing the location, the magnitude, the sources and the costs of the irreversibilities in an energy conversion system. The evaluation was carried out by applying the cost balance to each component of the investigated system [18] as follows:

\[
\sum \dot{C}_{out,k} + c_{W,k} \cdot \dot{W}_k = c_{Q,k} \cdot \dot{Q}_k + \sum \dot{C}_{in,k} + \dot{Z}_k^{CI}
\]  

(5)

or

\[
\sum c_{out,k} \cdot \dot{E}_{out,k} + c_{W,k} \cdot \dot{W}_k = c_{Q,k} \cdot \dot{Q}_k + \sum c_{in,k} \cdot \dot{E}_{in,k} + \dot{Z}_k^{CI}
\]  

(6)

The term \(\dot{Z}_k^{CI}\), which is the cost rate related to the capital investment, was computed as [18]:

\[
\dot{Z}_k^{CI} = \frac{CC_L \cdot \frac{PEC_K}{\sum n PEC_n}}{\tau}
\]  

(7)

where \(\tau\) (6000 h·year\(^{-1}\) [6]) is the amount of hours in which the considered plant operates at full load.

The cost of the electricity was assumed equal to 0.12 €·kWh\(^{-1}\) [6].

According to [6], the final cost of the product can be calculated by using (8):

\[
c_p,_{tot} = \frac{c_p,_{EV} + c_p,_{GC}}{E_p} = \frac{c_p,_{EV} \cdot \dot{Q}_{EV} \left(1 - \frac{T_0}{T_{cm}}\right) + c_p,_{GC} \cdot \dot{Q}_{GC} \left(1 - \frac{T_0}{T_{cm}}\right)}{\dot{Q}_{EV} \left(1 - \frac{T_0}{T_{cm}}\right)}
\]  

(8)

All the equations used to carry out the thermoeconomic evaluation are summarized in Table A.1.

In this study, the outcomes associated with the exergoeconomic analysis were not significantly affected by the fact that the uncertainty related to the economic evaluation was higher than ±10% [6].
3. Results

3.1. Comparison in terms of COP

According to Fig. 3, all the investigated solutions could outperform CRC. In particular, PCRC exhibited on average 17.3% higher COPs than the baseline for cooling medium temperatures ranging from 30 to 42 °C. Slightly greater values could be accomplished by employing PCSRC, which reached on average 21.2% higher energy saving than the baseline. Promising results associated with TERC were computed since it consumed on average 37.7% less power than CRC. Furthermore, due to the fact that the amount of vapour generated in the flash tank goes up with rise in cooling medium temperature, it was observed that the COP percentage difference trend of PCRC was decreasing as the heat sink temperature increased. Larger energy saving could be obtained by optimizing its intermediate pressure \[2-3\]. On the contrary, both PCSRC and TERC were more beneficial than CRC with rise in cooling medium temperature. The configuration with the ejector showed 36% at \(t_0 = 30 \) °C and 38.8% at \(t_0 = 42 \) °C higher COPs than CRC. As for PCSRC, its power consumption was on average 13.6% greater than that exhibited by TERC over the investigated temperatures range.

![Fig. 3 Percentage difference in COP in comparison with CRC.](image)

3.2. Comparison in terms of TCI

The energetic advantages associated with the adoption of PCRC and PCSRC clashed with the outcomes of the economic analysis. As showed in Fig. 4, the use of an additional compressor led to a substantial increase in the TCI since such components usually have the largest contribution to the economic investment related to a refrigerating plant.

The cost of PCRC and that of PCSRC were on average 29.3% and 38% higher than that associated with the basic cycle, respectively. As aforementioned, the usage of an ejector as a refrigerant expander allows reducing the pressure ratio of the compressor. Therefore, at the same boundary conditions, the size of the latter component is smaller and then cheaper than the one operating in CRC. Thanks to the cheapness of the ejector, its cost over the investigated PEC\(_{e_j}\) range did not affect the TCI significantly. The higher value of TCI for TERC in comparison with CRC could be attributed to the large expenditure associated with both the evaporator and the gas cooler in all the studied cases employing an ejector. As for the former, the result could be justified by taking into account the lower temperature difference observable through the overfed evaporator in relation to the conventional one. As regards the gas cooler, the amount of heat rejected into the surroundings went up more remarkably than the temperature of the R744 leaving the heat exchanger with rise in cooling medium temperature in all the investigated scenarios. This was significantly depending on the compressor discharge temperature, which entailed a larger gas cooler heat transfer area and therefore a higher PEC\(_{GC}\). In the selected scenarios, TERC exhibited on average from 11.2% to 12.5% higher TCI values than the basic configuration. The results showed that, unlike the other investigated configurations, the
percentage difference in TCI associated with TERC had a decreasing trend with rise in heat sink temperature.

3.3. Comparison in terms of cost-effectiveness

Fig. 5 compares the final cost of the product of all the studied configurations. No significant difference was noticed between PCRC and PCSRC, which exhibited on average about 4% higher cost-effectiveness than CRC over the selected temperatures range. In particular, the adoption of the former was slightly more beneficial for cooling temperatures up to 34 °C, otherwise PCSRC was preferable. As regards the three scenarios with the ejector, all of them justified the adoption of such device on the part of a R744 refrigeration systems operating in warm climates. The drop in the final cost of the product varied from 20.7% to 24.4% in comparison with the baseline over the investigated temperatures range. In all the studied scenarios, the higher the heat sink temperature, the more effective TERC was.

3.4. Influence of the diffuser efficiency on the cost-effectiveness of TERC

As demonstrated by Kornhauser [8], the diffuser efficiency has a great influence on the performance of an ejector-expansion refrigeration cycle. In this subsection, such parameter was varied in order to study its impact on the cost-effectiveness of TERC. Fig. 6 depicts these outcomes taking into account the scenario related to $\text{PEC}_{\text{eject}} = 1300$ € and different diffuser efficiency values. In comparison with CRC and for heat sink temperatures ranging from 30 °C to 42 °C, an average rise in the cost-effectiveness by 20.5% and by 25.2% was observed with $\eta_{\text{eject}} = 0.7$ and $\eta_{\text{eject}} = 0.9$, respectively. The
variation of the diffuser efficiency implied a change in the final cost of the product belonging to the order of magnitude of ±3% in relation to the scenario involving $\eta_{\text{ejec}} = 0.8$.

4. Discussion

The COP evaluation suggests that the configuration with the ejector consumes on average 37.7% less energy than the basic cycle over the cooling temperatures range varying from 30 °C to 42 °C. The energy saving related to the other two evaluated solutions ranges from 16.5% to 21.6%. Both the ejector-based configuration and the system with the subcooler are more beneficial at high heat sink temperatures. Although all the investigated solutions lead to good energetic performance, the ejector is the only technology which is also justifiable from both the economic viewpoint and the thermoeconomic one. In fact, the systems with parallel compression exhibits a total capital investment which is at least 28.1% higher than that of the baseline. On the contrary, the solution with the ejector shows at worst 15.9% larger TCI than the conventional solution due to a larger heat transfer area on the part of both the heat exchangers. Due to the current lack of cost equations related to the ejectors, values of 600 €, 1300 € and 2000 € have been taken into account. The results of the exergoeconomic analysis have demonstrated that a transcritical ejector refrigerating cycle has at worst 20.7% lower final cost of the product than the baseline over the investigated cooling medium temperatures range. On the other hand, the adoption of either of the other studied configurations would lead at best to a decrement of 5.9%. An additional drop by 3% can be accomplished by adopting a diffuser with an efficiency of 0.9.

5. Conclusions

In this paper, the cost-effectiveness of three different CO$_2$ refrigeration systems, i.e. two solutions employing the so-called “parallel compression” and the one using a two-phase ejector as an expansion device, has been compared with that of a basic R744 cycle. The typical running modes of a supermarket have been considered by adopting a cooling capacity of 100 kW and an evaporating temperature of -10 °C. The latter has been increased by 3 K for the system with the ejector to consider the energetic advantages associated with the use of overfed evaporators. A cooling temperature ranging from 30 °C to 42 °C has been selected. The implementation of the ejector has been based on the model proposed by Kornhauser.

The replacement of an expansion valve in a transcritical R744 refrigerating plant is particularly reasonable from the energetic point of view. The application of the economic and exergoeconomic analyses provides a further push towards the employment of this technology, especially at high heat sink design temperatures. In fact, it has been observed that the higher the gas cooler outlet temperature, the more competitive this solution becomes from the energetic, economic and
exergoeconomic viewpoints. It is worth remarking that, in subcritical conditions, an auxiliary compressor is usually employed in configurations similar to TERC in order to avoid unsuitable values of entrainment ratio, i.e. the ratio between the mass flow rate coming from the evaporator and the mass flow rate coming from the gas cooler. On the other hand, the significant improvement in the overall heat transfer coefficient of the overfed evaporators compensates the need for increasing their heat transfer area caused by the reduction in the temperature difference through the heat exchanger. Furthermore, the sensitive analysis carried out as a function of the diffuser efficiency has confirmed that close attention should be paid to this parameter. It can be concluded that a transcritical R744 refrigeration system operating in warm climatic conditions should include a two-phase ejector as a substitute of the expansion valve.

Appendix A

Table A.1. Cost balance equations for the exergoeconomic analysis of the investigated solutions.

<table>
<thead>
<tr>
<th>PCRC</th>
<th>PCSRC</th>
</tr>
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<tbody>
<tr>
<td>COM</td>
<td>[\dot{C}<em>{\text{out,COM}} = \dot{C}</em>{\text{out,EV}} + c_{el} \cdot W_{\text{COM}} + 2\dot{\phi}_{\text{Cl,COM}}]</td>
</tr>
<tr>
<td>GC</td>
<td>[\dot{C}<em>{\text{out,GC}} + \dot{\phi}</em>{\text{P,GC}} = \dot{C}<em>{\text{in,GC}} + 2\phi</em>{\text{GC}}] [c_{\text{out,GC}} = c_{\text{in,GC}}]</td>
</tr>
<tr>
<td>EX (HP)</td>
<td>[\dot{C}<em>{\text{in,TANK}} = \dot{C}</em>{\text{out,GC}} + 2\phi_{\text{Cl}}]</td>
</tr>
<tr>
<td>TANK</td>
<td>[\dot{C}<em>{\text{out,TANK vap}} + \dot{C}</em>{\text{out,TANK liq}} = \dot{C}<em>{\text{in,TANK}} + 2\phi</em>{\text{TANK}}] [c_{\text{out,TANK vap}} = c_{\text{in,TANK liq}}]</td>
</tr>
<tr>
<td>EX (MT)</td>
<td>[\dot{C}<em>{\text{in,EV}} = \dot{C}</em>{\text{out,TANK liq}} + 2\phi_{\text{Cl}}]</td>
</tr>
<tr>
<td>EV</td>
<td>[\dot{C}<em>{\text{out,EV}} + \dot{\phi}</em>{\text{P,EV}} = \dot{C}<em>{\text{in,EV}} + 2\phi</em>{\text{Cl}}] [c_{\text{out,EV}} = c_{\text{in,EV}}]</td>
</tr>
<tr>
<td>AUX</td>
<td>[\dot{C}<em>{\text{out,AUX}} = \dot{C}</em>{\text{out,TANK vap}} + c_{el} \cdot W_{\text{AUX}} + 2\phi_{\text{Cl}}]</td>
</tr>
<tr>
<td>MIX</td>
<td>[\dot{C}<em>{\text{in,EV}} = \dot{C}</em>{\text{out,AUX}} + \dot{C}_{\text{out,COM}}]</td>
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TERC

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<td>COM</td>
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<td>SEP</td>
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<td>EX</td>
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<td>EV</td>
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<td>EJEC</td>
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</table>
Nomenclature

A  heat transfer area, m²
AUX auxiliary
c  cost per unit of exergy, €·GJ⁻¹

C  cost rate of exergy, €·h⁻¹
CCₐ levelized carrying cost, €
COP coefficient of performance
CRC conventional refrigeration cycle
CRF capital recovery factor
e  exergy per unit of mass, kJ·kg⁻¹

Ê  exergy rate, kW
EES engineering equation solver
EJEC ejector
EV evaporator
EX expansion valve
GWP global warming potential, kgCO₂·kg⁻¹refrigerant
HFC hydro-fluorocarbon
HP high-pressure
i  average effective discount rate, %
m  mass flow rate, kg·s⁻¹
MIX mixing point
MT medium temperature
p  pressure, bar
PCRC refrigeration cycle with parallel compression
PCRSC refrigeration cycle with subcooler and parallel compression
PEC purchased equipment cost, €

Q  heat transfer rate, kW
SEP separator
SUB subcooler
TCI total capital investment, €
TERC transcritical ejector refrigeration cycle
T  temperature, K
W  power input, kW

Z  cost rate, €·h⁻¹

Greek symbols

η  efficiency
τ  annual number of operating hours, h·year⁻¹

Subscripts and superscripts

0  Dead state
References


