

Exergoeconomic and environmental analyses of CO₂/NH₃ cascade refrigeration systems equipped with different types of flash tank intercoolers

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Research Highlights

- CO₂/NH₃ cascade refrigeration cycles with flash intercoolers are investigated.
- Exergoeconomic factors of components are determined to assess their relative significances.
- An environmental analysis is applied to determine the penalty cost of GHG emission.
- The effects of operating parameters on COP, exergy efficiency and total cost rate are investigated.
- An optimization is applied based on the maximum COP and the minimum total cost rate.

1 **Exergoeconomic and environmental analyses of CO₂/NH₃ cascade**
2 **refrigeration systems equipped with different types of flash tank**
3 **intercoolers**

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11 **Abstract:**

12 Exergoeconomic and environmental analyses are presented for two CO₂/NH₃ cascade
13 refrigeration systems equipped with 1) two flash tanks, and 2) a flash tank along with a flash
14 intercooler with indirect subcooler. A comparative study is performed for the proposed
15 systems, and optimal values of operating parameters of the system are determined that
16 maximize the coefficient of performance (COP) and exergy efficiency and minimize the total
17 annual cost. The operating parameters considered include condensing temperatures of NH₃ in
18 the condenser and CO₂ in the cascade heat exchanger, the evaporating temperature of CO₂ in
19 the evaporator, the temperature difference in the cascade heat exchanger, the intermediate
20 pressure of the flash tank in the CO₂ low-temperature circuit, the mass flow rate ratio in the
21 flash intercooler and the degree of superheating of the CO₂ at the evaporator outlet. The total
22 annual cost includes the capital, operating and maintenance costs and the penalty cost of
23 GHG emission. The results show that, the total annual cost rate for system 1 is 11.2% and

24 11.9% lower than that for system 2 referring to thermodynamic and economic optimizations,
 25 respectively. For thermodynamic and cost optimal design condition the COP and exergy
 26 efficiency of both systems are almost the same. Finally, in order to obtain the best balance
 27 between exergy destruction cost and capital cost, the exergoeconomic factor is defined for
 28 each component of proposed systems, for cases in which the system operates at the best
 29 performance conditions.

30 **Keywords:** Cascade refrigeration system; CO₂/NH₃; Exergoeconomic analysis;
 31 Environmental analysis; Optimization; Flash tank.

32 **Nomenclature**

A	area (m ²)
c	unit cost of exergy (\$ kJ ⁻¹)
\dot{C}	cost rate (\$ s ⁻¹)
CO ₂ e	carbon dioxide equivalent
COP	coefficient of performance
CRF	capital recovery factor
E	electrical energy consumption (kWh)
$\dot{E}x$	exergy rate (kW)
f	exergoeconomic factor
F	correction factor
FT	flash tank
FIS	flash intercooler with indirect subcooler
GHG	greenhouse gas
GWP	global warming potential
h	specific enthalpy (kJ kg ⁻¹)

HTC	high-temperature compressor
i	annual interest rate
LTC	low-temperature compressor
\dot{m}	mass flow rate (kg s^{-1})
m	mass (kg)
n	system life time (year)
N	operational hours in a year (h)
ODP	ozone depletion potential
P	pressure (kPa)
PR	pressure ratio
\dot{Q}	heat rate (kW)
r	mass flow rate ratio
s	specific entropy ($\text{kJ kg}^{-1} \text{K}^{-1}$)
T	temperature ($^{\circ}\text{C}$ or K)
TV	throttling valve
ΔT_{lm}	logarithmic mean temperature difference (K)
U_o	overall heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
\dot{V}	volumetric flow rate ($\text{m}^3 \text{s}^{-1}$)
\dot{W}	electrical power (kW)
\dot{Z}	capital cost rate ($\text{\$ s}^{-1}$)
Z	capital cost ($\text{\$}$)

Greek symbols

α_{el}	unit electricity cost ($\text{\$ kWh}^{-1}$)
ϕ	maintenance factor

η	energy efficiency
$\mu_{\text{CO}_2\text{e}}$	emission conversion factor (kg kWh^{-1})
ψ	exergy efficiency

Subscripts

0	ambient
ca	cooled air
CAS	cascade heat exchanger
CD	condenser
CM	compressor
D	destruction
e	exit
env	environment
el	electricity
EV	evaporator
F	fuel
i	inlet
int	intermediate
k	<i>k</i> th component
m	mechanical
OP	operation
P	product
s	isentropic
sup	superheating
t	thermal

33 **1. Introduction**

34 The use of CO₂ as a working fluid in refrigeration cycles has expanded notably in recent
35 years, because it has low global warming potential (GWP) and no ozone depletion potential
36 (ODP). It is also non-flammable, inexpensive and abundant in nature. Moreover, CO₂ (R744)
37 has advantages in use as a refrigerant in low temperature applications such as storage of
38 frozen food and rapid freezing systems. Despite of these advantages of CO₂ as a working
39 fluid in refrigeration cycles, using carbon dioxide as the working fluid in a single stage
40 refrigeration cycle is normally not economical due to the high pressure difference between
41 evaporator and condenser. In single stage refrigeration systems using CO₂ as a refrigerant, a
42 high pressure ratio and condensation close to the critical conditions lead to a low coefficient
43 of performance (COP) in comparison with the refrigeration cycles working with HFC
44 refrigerants [1].

45 Two-stage compression systems and cascade refrigeration cycles can be used for these
46 applications to overcome the aforementioned problem [2–7]. A cascade refrigeration cycle
47 involves two refrigeration circuits which are thermally coupled through an internal cascade
48 heat exchanger. The internal cascade heat exchanger plays the role of condenser for the low
49 temperature circuit and evaporator for the high temperature circuit. The CO₂/NH₃ cascade
50 refrigeration cycle uses two natural refrigerants, NH₃ (R717) in the high temperature circuit
51 and CO₂ in the low temperature circuit, and is a well-known system in refrigeration industry.
52 Research on CO₂/NH₃ cascade refrigeration has been reported by several authors. Lee et al.
53 [8] thermodynamically assessed a CO₂/NH₃ cascade refrigeration to determine the optimal
54 condensing temperature of the cascade heat exchanger to maximize the COP and minimize
55 the exergy destruction of the system. Getu and Bansal [9] thermodynamically analyzed a
56 CO₂/NH₃ cascade refrigeration system and optimized several cycle operating parameters:
57 condensing, evaporating, subcooling and superheating temperatures and temperature

58 difference in the cascade heat exchanger. They showed that an increase in subcooling before
59 expansion to the evaporator increased the COP of the system while an increase in
60 superheating and condensing temperature decreased the COP. Dopazo et al. [10] analyzed a
61 CO₂/NH₃ cascade refrigeration system and identified the optimum CO₂ condensing
62 temperature based on energy and exergy points of view. Bingming et al. [11] experimentally
63 investigated the effects of operation parameters on the performance of a CO₂/NH₃ cascade
64 refrigeration system, and showed that the system COP is greatly affected by evaporating and
65 condensing temperatures and temperature difference in cascade heat exchanger while it is
66 only slightly sensitive to the degree of superheating. Dopazo and Fernandez-Seara [12]
67 experimentally evaluated a CO₂/NH₃ cascade refrigeration system for an industrial freezer
68 with a -50 °C evaporating temperature. They also investigated the influence of the operating
69 parameters on system performance and compared the results with those for common NH₃
70 two stage refrigeration systems under the same operating conditions. They concluded that the
71 COP of the cascade system is similar to the COP of an ammonia double stage with
72 intercooler and about 20% higher when an economizer is applied. Ma et al. [13]
73 thermodynamically analyzed a CO₂/NH₃ cascade refrigeration system using a falling film
74 evaporator–condenser as the cascade heat exchanger, and showed that the use of such a heat
75 exchanger improved the system COP by providing a smaller temperature difference.
76 After a technical feasibility study, the thermodynamic analysis must be completed with
77 considerations about the costs of systems incorporated. Therefore, an economic analysis
78 should also be considered for analyzing a refrigeration plant. Mitishita et al. [14] developed
79 an optimization methodology to reduce power consumption and costs for frost-free
80 refrigerators. This methodology was used to determine the compressor size and efficiency,
81 the number of condenser and evaporator fins and the evaporator air flow rate in order to
82 minimize energy consumption. Various studies based on exergy and thermoeconomic

83 concepts in relation to heat pumps [15–17] and refrigeration systems have been previously
84 published. Rezayan and Behbahaninia [18] presented a thermoeconomic optimization for a
85 simple CO₂/NH₃ cascade refrigeration system without considering environmental analysis.
86 They investigated the influence of design parameters on total annual cost of the system when
87 ambient temperature, cooling capacity and cold space temperature are constraints.
88 Exergoeconomic analysis plays a key role in determining the optimal performance of a
89 thermodynamic system. By combining exergy analysis and economic principles in a cost-
90 effective method, exergoeconomic analysis can be used to identify the optimum system
91 design via exergy-aided cost minimization. Moreover, due to the consumption of fossil fuels
92 to generate electricity, an environmental analysis that determines the amount of greenhouse
93 gas (GHG) emission is important for analyzing and optimizing such thermodynamic systems.
94 In the present study, exergoeconomic and environmental analyses are applied to the different
95 multistage CO₂/NH₃ cascade refrigeration systems. Ammonia is the preferred refrigerant.
96 However, since ammonia is toxic, it is common practice to use carbon dioxide to distribute
97 refrigeration at low temperatures while the high temperatures are served by ammonia in a
98 restricted area. In this study two multistage CO₂/NH₃ cascade refrigeration systems equipped
99 with 1) two flash tanks, 2) a flash tank along with flash intercooler with indirect subcooler are
100 proposed. Typically, exergoeconomic and environmental analyses of such systems have not
101 been reported, but are needed to provide a more comprehensive view. Furthermore, the
102 effects on performance and total annual cost for each system are investigated for operational
103 parameters such as evaporator, condenser and cascade heat exchanger outlet temperatures,
104 pressures of the flash tank (FT) or flash intercooler with indirect subcooler (FIS) of the low-
105 temperature circuit, mass flow rate ratio of the FIS and degree of superheating of CO₂ at the
106 evaporator outlet. Also an optimization is performed based on maximum COP and exergy
107 efficiency and the minimum total cost rate (including capital, operating and maintenance

108 costs as well as the penalty cost of GHG emission). The objective is to improve
109 understanding of CO₂/NH₃ cascade refrigeration systems equipped with flash tanks with or
110 without an indirect subcooler and the benefits that their use can provide.

111 2. System description

112 Fig. 1(a) provides a schematic of the CO₂/NH₃ cascade refrigeration cycle equipped with two
113 flash tanks (system 1). The system consists of the two loops: a high-temperature cycle with
114 NH₃ as the working fluid and a low-temperature cycle with CO₂ as the working fluid. Both
115 loops are equipped with flash tanks while the one in the CO₂ loop has also an intercooler
116 function. A flash intercooler cools the discharge vapor exiting the low-temperature
117 compressor (LTC I) before it enters the LTC II. The vapor cooling is performed within the
118 flash tank by vaporizing some liquid at the pressure maintained in the tank. In the high-
119 temperature cycle, the saturated liquid NH₃ from the flash tank flows to the cascade
120 condenser. At the same time, the superheated CO₂ vapor from the LTC II enters the cascade
121 condenser. In the cascade heat exchanger, NH₃ evaporates to a saturated vapor while CO₂
122 condenses to a saturated liquid. Then, the NH₃ vapor from the cascade condenser enters the
123 flash tank, from which saturated NH₃ vapor flows to the high-temperature compressor
124 (HTC). In the low-temperature cycle, the saturated CO₂ liquid from the cascade condenser,
125 after isenthalpic expansion in throttling valve (TV II), returns to the CO₂ flash tank and
126 partially vaporizes due to flashing and cooling of the superheated CO₂ vapor from LTC I.
127 The residual CO₂ saturated liquid then flows to TV I. The condenser in the high-temperature
128 cycle rejects the heat to the environment at inlet temperature $T_{env,i}$ and the evaporator in the
129 low-temperature cycle absorbs heat from the cold air at inlet temperature $T_{ca,i}$. Fig. 1(b)
130 shows the processes occurring in both the high- and low-temperature cycles on a $T-s$
131 diagram.

132 Fig. 2(a) shows a schematic of the CO₂/NH₃ cascade refrigeration cycle equipped with a
 133 flash tank and a flash intercooler with an indirect subcooler (system 2). The CO₂ after the
 134 cascade heat exchanger is divided into two streams. One is throttled down to the intermediate
 135 pressure through TV II and flows into the FIS. Then the CO₂ flashes to a vapor, cools the
 136 residual stream of high pressure liquid, mixes and exchanges heat with the discharged high
 137 temperature CO₂ from LTC I. Then the resulting saturated vapor is drawn in to LTC II. The
 138 cooled high pressure liquid is expanded in the TV I and then fed to the evaporator. Fig. 2(b)
 139 shows the processes on a $T-s$ diagram.

140 3. Thermodynamic, economic and environmental analyses

141 For the thermodynamics and economics analyses of the proposed CO₂/NH₃ cascade
 142 refrigeration system it is assumed that pressure and heat losses in all system components and
 143 connections are negligible and that all components operate under steady-state conditions. It is
 144 also assumed that nuclear, electric, electromagnetic and surface tension effects are absent and
 145 that changes in kinetic and potential energy are negligible. Moreover, there is no subcooling
 146 at the outlet of the condenser and cascade heat exchanger.

147 3.1. Energy analysis

148 Applying the first law of thermodynamics, a steady-state form of the energy rate balance for
 149 the k th component of system can be expressed as follows:

$$\dot{Q}_k + \sum_i (\dot{m}h)_k = \sum_e (\dot{m}h)_k + \dot{W}_k \quad (1)$$

150 The cooling load of the system is equal to the heat transfer rate absorbed by the CO₂
 151 evaporator and is defined as:

$$\dot{Q}_{EV} = \dot{m}_1 (h_1 - h_8) \quad (2)$$

152 The electric power consumption of the compressor is obtained as:

$$\dot{W}_{\text{CM}} = \frac{\dot{m}(h_{\text{es}} - h_1)}{\eta_s \eta_{\text{el}} \eta_m} = \frac{\dot{m}(h_{\text{es}} - h_1)}{\eta_{\text{total}}} \quad (3)$$

153 where η_s , η_{el} and η_m respectively are the isentropic, electrical and mechanical efficiencies
 154 of the compressor. The total isentropic efficiency of the considered compressors, η_{total} , is
 155 defined as:

156 For the HTC (ammonia screw compressor) (J.S. Bahamonde, personal communication,
 157 February 5, 2012):

$$\eta_{\text{total}} = \begin{cases} 0.0071PR^5 - 0.1264PR^4 + 0.9023PR^3 - 3.2277PR^2 & \text{for } PR < 4.3 \\ + 5.7871PR - 3.3429 & \\ -0.0261PR + 0.9069 & \text{for } PR \geq 4.3 \end{cases} \quad (4)$$

158 For the LTC (carbon dioxide piston compressor) (L. Shi, personal communication, October
 159 19, 2015):

$$\eta_{\text{total}} = \begin{cases} -0.1234PR^4 + 1.1251PR^3 - 3.8902PR^2 + 6.0433PR - 2.8860 & \text{for } PR < 2.7 \\ -0.0237PR^4 + 0.3051PR^3 - 1.4740PR^2 + 3.1348PR - 1.7978 & \text{for } PR \geq 2.7 \end{cases} \quad (5)$$

160 where PR is the pressure ratio of the compressor. Defining the mass flow rate ratio of the
 161 flash intercooler as $r = \dot{m}_7 / \dot{m}_6$ in system 2, the energy balance equation for the flash
 162 intercooler can be written as follows:

$$h_6 + r(h_2 + h_5) = r(h_3 + h_7) + h_3 \quad (6)$$

163 The power consumptions of the evaporator and condenser fans are approximated as follows
 164 [19]:

$$\dot{W}_{\text{Fan I}} = 0.075(\dot{Q}_{\text{EV}}) \quad (7)$$

$$\dot{W}_{\text{Fan II}} = 0.027 \left(\dot{Q}_{\text{EV}} + \sum_j \dot{W}_{\text{CM},j} \right) \quad (8)$$

165 where $\sum_j \dot{W}_{\text{CM},j}$ denotes the sum of the electrical power consumptions of the compressors.

166 The total electrical power consumption of the system can be written as:

$$\dot{W}_{\text{total}} = \dot{W}_{\text{LPC I}} + \dot{W}_{\text{LPC II}} + \dot{W}_{\text{HPC}} + \dot{W}_{\text{Fan I}} + \dot{W}_{\text{Fan II}} \quad (9)$$

167 The COP of the system is defined as:

$$\text{COP} = \frac{\dot{Q}_{\text{EV}}}{\dot{W}_{\text{total}}} \quad (10)$$

168 The total heat transfer area of the heat exchangers is calculated as follows:

$$A = \frac{\dot{Q}}{U_o F \Delta T_{\text{lm}}} \quad (11)$$

169 where U_o and ΔT_{lm} are the overall heat transfer coefficient based on external heat transfer
 170 area and the logarithmic mean temperature difference (LMTD) of the heat exchanger,
 171 respectively. A mathematical relationship to determine the LMTD correction factor, F , is
 172 given by Fettaka et al. [20]. For counter-flow heat exchangers and the evaporator, the
 173 correction factor F has a value of 1 but for the condenser the value of F should be calculated.

174 3.2. Exergy analysis

175 When the kinetic and potential energies are neglected, the physical exergy at point j in a
 176 system can be expressed by:

$$\dot{E}x_j = \dot{m}_j \left[(h - h_0)_j - T_0 (s - s_0)_j \right] \quad (12)$$

177 where T_0 is the thermodynamic averaged temperature of the ambient environment defined as
 178 follows [21]:

$$T_0 = \frac{(T_e - T_i)_{\text{env}}}{\ln(T_e/T_i)_{\text{env}}} \quad (13)$$

179 Applying an exergy balance to the k th system component, the exergy destruction rate can be
 180 defined as follows:

$$\dot{E}x_{\text{D},k} = \dot{E}x_{\text{F},k} - \dot{E}x_{\text{P},k} \quad (14)$$

181 where the subscripts 'F' and 'P' indicate fuel (or driving input) and product (or desired
 182 output), respectively. The exergy efficiency can be expressed as the ratio of product exergy
 183 rate to fuel exergy rate:

$$\psi_k = \frac{\dot{E}x_{P,k}}{\dot{E}x_{F,k}} \quad (15)$$

184 Estimations of fuel and product exergy rates for each component of these proposed systems
 185 are given in Table 1. For the exergy analysis of the throttling valve, it is necessary to split the
 186 physical exergy of the fluid flow into its mechanical and thermal parts [22].

187 The product exergy rate of the system is the exergy rate of heat transferred to the evaporator:

$$\dot{E}x_p = \dot{E}x_{ca,e} - \dot{E}x_{ca,i} \quad (16)$$

188 the fuel exergy rate of the system is the total electrical power input:

$$\dot{E}x_F = \dot{W}_{total} \quad (17)$$

189 Accordingly, the exergy efficiency of the system can be expressed as:

$$\psi = \frac{\dot{E}x_{ca,e} - \dot{E}x_{ca,i}}{\dot{W}_{total}} = 1 - \frac{\dot{E}x_{D,total}}{\dot{W}_{total}} \quad (18)$$

190 3.3. Economic analysis

191 In the economic analysis, a cost rate balance can be expressed for the overall system as
 192 follows:

$$\dot{C}_{total} = \dot{C}_{env} + \dot{Z}_{OP} + \sum_k \dot{Z}_k \quad (19)$$

193 where \dot{C}_{env} is the rate of penalty cost of GHG emission for the k th component (see section
 194 3.4). The operating cost of the system, \dot{Z}_{OP} , including the cost of electricity consumption, can
 195 be defined as follows:

$$\dot{Z}_{OP} = N \times \dot{W}_{total} \times \alpha_{el} \quad (20)$$

196 where N is the yearly number of operation hours of the system and α_{el} is the unit electricity
 197 cost in $\$ \text{kWh}^{-1}$. The rate of capital investment and maintenance costs of each system
 198 component can be estimated as follows [23]:

$$\dot{Z}_k = \frac{Z_k \times \phi}{N \times 3600} \text{CRF} \quad (21)$$

199 where Z_k is the capital cost of the k th component and ϕ is the maintenance factor. The capital
 200 recovery factor (CRF) is defined as [24]:

$$\text{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (22)$$

201 where i and n are the annual interest rate and system life time, respectively.

202 Exergy destructions and capital costs are the real cost sources of a thermodynamic system. In
 203 an exergoeconomic evaluation, the exergoeconomic factor expresses the relative significance
 204 of a component and can be defined as follows [25]:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k} \dot{E}x_{D,k}} \quad (23)$$

205 where $c_{F,k}$ is the unit cost of fuel for the k th component and can be calculated by solving the
 206 exergy cost rate balance for the k th component, which can be expressed in a general form as
 207 [24]:

$$\sum_e (c\dot{E}x)_k = \sum_i (c\dot{E}x)_k + \dot{C}_{env,k} + \dot{Z}_k + \dot{Z}_{OP,k} \quad (24)$$

208 where c is the unit cost of exergy in each flow. In this study, external exergy losses are not
 209 considered and the thermodynamic inefficiencies of a component consist exclusively of
 210 exergy destruction [26]. A low value of f_k calculated for a major component suggests that
 211 cost savings in the entire system might be achieved by improving the component efficiency
 212 even if it increases the capital investment for the component. Conversely, a high value of f_k

213 suggests a decrease in the investment costs of this component at the expense of its exergetic
214 efficiency may be reasonable [24].

215 3.4. Environmental analysis

216 The rate of penalty cost of GHG emission for the considered system can be determined based
217 on the annual amount of GHG emission from the system, $m_{\text{CO}_2\text{e}}$, as follows [27]

$$\dot{C}_{\text{env}} = m_{\text{CO}_2\text{e}} c_{\text{CO}_2} \quad (25)$$

218 where c_{CO_2} is the cost of CO₂ avoided and $m_{\text{CO}_2\text{e}}$ is obtained as:

$$m_{\text{CO}_2\text{e}} = \mu_{\text{CO}_2\text{e}} \times E_{\text{annual}} \quad (26)$$

219 Here, $\mu_{\text{CO}_2\text{e}}$ is the emission factor and E_{annual} is the annual electrical energy consumption of
220 the system in kWh.

221 4. System specifications

222 To determine the investment cost rate of each component, the maintenance factor (ϕ) is 1.06
223 and the investment cost (Z_k) can be estimated based on the cost functions listed in Table 2.

224 In calculating the CRF, the annual interest rate (i), the life time of the system (n) are

225 considered as 14% and 15 years respectively. The average electricity cost is 0.09 \$ kWh⁻¹

226 (Iran's electricity tariff in 2015) and the annual operational hours of the system (N) are

227 considered to be 4266 h [19]. The emission factor of electricity ($\mu_{\text{CO}_2\text{e}}$) is taken to be 0.968

228 kg kWh⁻¹ (Iran's average) and the cost of CO₂ avoided (c_{CO_2}) is considered as 87 \$ ton⁻¹ of

229 CO₂e emissions for the natural gas combined cycle power plants with post-combustion

230 capture technology [28]. Thermodynamic conditions of the system are listed in Table 3.

231 5. Results and discussion

232 The validation for a basic CO₂/NH₃ cascade refrigeration system is shown in Table 4. Good
233 agreement is observed between the obtained results for performance parameters of the system
234 and the corresponding results reported in References [18] and [29]. However, since Ref. [18]
235 does not consider the cost of condenser and evaporator fans, the predicted value of the total
236 cost rate for the present model is 1.3% higher than reported in Ref. [18].

237 Fig. 3 shows the variations of system COP and exergy efficiency with evaporating
238 temperature of the CO₂ for the two proposed CO₂/NH₃ cascade refrigeration systems. In
239 obtaining these results, other operating parameters are kept constant. Since the
240 thermodynamic averaged temperature of the ambient environment and the cooled space are
241 fixed, the trend of COP variation is the same as for that of the exergy efficiency. Due to the
242 minimum allowable temperature difference of 5 K in the flash intercooler with an indirect
243 subcooler, the mass flow rate ratio, r , should be greater than 3.2. As expected, an increase in
244 evaporating temperature decreases the pressure ratio of LTC I and the total electrical power
245 consumption. Therefore, the system COP and exergy efficiency both increase. Moreover, in
246 system 2, an increase of r leads to a rise in mass flow rate through LTC I, which increases
247 the compression work and causes the COP and exergy efficiency to decrease. Under the
248 given conditions for system 1, a 10 K increase in T_{EV} from $-45\text{ }^{\circ}\text{C}$ to $-35\text{ }^{\circ}\text{C}$ leads to
249 increases of 16.9% in both COP and exergy efficiency. At the same condition for system 2, a
250 10 K increase in T_{EV} leads to a maximum increase of 20.8% in both COP and exergy
251 efficiency when $r = 3.6$. Also, it can be seen from Fig. 3 that, due to the lower electrical
252 power consumption, the COP and exergy efficiency of system 1 are greater than for system 2,
253 under the same operating conditions.

254 The effect of varying the evaporating temperature of the CO₂ on the ratio of the penalty cost
255 of GHG emission and the total annual cost rate is shown in Fig. 4. By increasing the CO₂
256 evaporating temperature, the electrical power consumption of LTC I decreases and leads to a

257 reduction in the \dot{C}_{env} . Also, by increasing the CO₂ evaporating temperature, the heat transfer
 258 surface area of the evaporator increases and leads to an increase in the system capital and
 259 total costs above a certain value of T_{EV} . It is also observed that, under the same operating
 260 conditions, the total cost rate of system 2 exceeds that of system 1, as a result of the higher
 261 capital cost of FIS. For instance, using FT in the CO₂ circuit leads to a decrease of up to
 262 14.3% in the total cost rate in comparison with system 2 at $T_{EV} = -45$ °C.

263 Fig. 5 shows the variation of the COP and exergy efficiency of the system with condensing
 264 temperature of the NH₃ for the proposed CO₂/NH₃ refrigeration systems. These results are
 265 obtained with the other operating parameters kept constant. An increase in condensing
 266 temperature is seen to increase the pressure ratio of the HTC and the total electrical power
 267 consumption and leads to decreases in the system COP and exergy efficiency. For system 1, a
 268 10 K increase in T_{CD} leads to a 13.4% decrease in both COP and exergy efficiency. At the
 269 same condition for system 2, a 10 K increase in T_{CD} leads to maximum decrease of 13.3% in
 270 both COP and exergy efficiency when $r = 3.2$. Moreover, using a FIS with $r = 3.2$ in the
 271 system leads to decreases of up to 1.2% for both system COP and exergy efficiency relative
 272 to system 1, at a condensing temperature 35 °C.

273 Fig. 6 shows the effect of varying the condensing temperature of NH₃ on the ratio cost rate
 274 due to GHG emission and the total annual cost rate. By increasing the NH₃ condensing
 275 temperature, the electrical power consumption of the HTC increases and leads to an
 276 increment in \dot{C}_{env} . It can be seen that variations of the ratio of penalty cost of GHG emission
 277 to r is negligible for system 2, due to the small effect of r on the performance of the NH₃
 278 circuit. Also, by increasing the NH₃ condensing temperature, the total costs of the HTC
 279 increase due to the increased electrical power consumption, leading to an increase of the total
 280 cost rate of the system. In systems 1 and 2, a 10 K increase in T_{CD} leads to 4.8% and 4.2%

281 increases respectively in total cost rate. Furthermore, using an open intercooler in the CO₂
282 circuit leads to decreases of almost 13% for the total annual cost rate in comparison to system
283 2.

284 Fig. 7 illustrates the variation of system COP and exergy efficiency with the cascade
285 temperature difference of the systems, $\Delta T_{\text{CAS}} (= T_5 - T_{13})$. An increase in cascade temperature
286 difference raises the pressure ratio of the HTC and the total electrical power consumption,
287 while the condensing temperatures of CO₂ in the cascade heat exchanger and NH₃ in
288 condenser are kept constant. Therefore, system COP and exergy efficiency both decrease.
289 Under the given conditions, a 10 K increase in ΔT_{CAS} from 2 K to 12 K leads to decreases of
290 14.6% and up to 14.5% for systems 1 and 2, respectively, in both COP and exergy efficiency.
291 It can also be seen in Fig. 7 that using a flash intercooler with $r = 3.2$ in the system leads to
292 1.3% decreases in both COP and exergy efficiency when the cascade temperature difference
293 is set to 2 K.

294 The effect of varying the cascade heat exchanger temperature difference on the ratio of GHG
295 emission cost rate and total annual cost rate is shown in Fig. 8. Increasing the cascade heat
296 exchanger temperature difference is seen to raise the ratio GHG emission cost rate and total
297 cost rate, due to an increase of the capital and operating costs of the HTC which in turn is a
298 result of the electrical power consumption increase. For system 1, a 10 K increase in ΔT_{CAS}
299 leads up to 5.7% increase in \dot{C}_{total} . For the system 2, a 10 K increase in ΔT_{CAS} leads to an
300 increase of 5.4% in \dot{C}_{total} when $r = 3.2$. Also using a FT in the system leads to a decrease of
301 up to 13.2% in \dot{C}_{total} in comparison with system 2 at $\Delta T_{\text{CAS}} = 2$ K.

302 Fig. 9 illustrates the variation of the COP and exergy efficiency of the system with
303 condensing temperature of the CO₂ in the cascade heat exchanger, T_5 , for the presented
304 systems. These results are obtained while other operating parameters are kept constant. [Since](#)

305 the minimum temperature difference in the flash intercooler with an indirect subcooler should
 306 be greater than 5 K, the lower limit of T_5 is considered as 0 °C. An increase in T_5 leads to an
 307 increase in pressure ratio of the LTC II and a decrease in pressure ratio of the HTC since
 308 ΔT_{CAS} , T_{CD} and P_{int} are held constant. As long as the reduction in HTC power is greater than
 309 the increase in LTC II power (T_5 is less than 2 °C), COP and exergy efficiency increase.
 310 When the increment in LTC II power is greater than the reduction in HTC power, COP and
 311 exergy efficiency decrease.
 312 The effect of varying the condensing temperature of the CO₂ in the cascade heat exchanger
 313 on the ratio of cost rate due to GHG emission and total annual cost rate is shown in Fig. 10.
 314 The results show that, when T_5 is less than 2 °C, due to the reduction in total electrical power
 315 consumption, the capital and operating costs of the system decline in both systems, while
 316 other operating parameters are kept constant. After that, due to the increment in total
 317 electrical power consumption, \dot{C}_{env} and \dot{C}_{total} increase. Fig. 10 (a) shows that the ratio of cost
 318 rate due to GHG emission decreases with increasing condensing temperature of the CO₂ due
 319 to the decreased power consumption and GHG emission. Although the COP and exergy
 320 efficiency are less sensitive to the use of an FT instead of an FIS in the CO₂ circuit, the total
 321 cost rate depends on this choice. System 1 leads to a 18.4% decrease in the total cost rate at
 322 $T_5=10$ °C in comparison to system 2.
 323 Fig. 11 shows the variation of the COP and exergy efficiency of the system with the
 324 intermediate pressure for the CO₂ low-temperature circuit. Due to the increase and decrease
 325 in the electrical power consumption of LTC I and LTC II respectively, which result from the
 326 variation of pressure ratio, an optimal P_{int} is seen to exist which leads to a maximum COP
 327 and exergy efficiency. The optimum value of P_{int} is sensitive to the evaporating and
 328 condensing temperatures of CO₂ in the low-temperature circuit. However, using system 1

329 instead of system 2 leads to an increase in the optimum value of P_{int} while T_{EV} and T_5 are
330 kept constant. The results also show that the mass flow rate ratio r has a negligible effect on
331 COP and exergy efficiency for system 2.

332 The effect of the intermediate pressure for the CO₂ low-temperature circuit on the ratio of the
333 cost rate due to GHG emission and the total annual cost rate is shown in Fig. 12, where it is
334 observed that, at the optimal P_{int} , the lowest penalty cost of GHG emission and total cost rate
335 are obtained. In this case, the total cost rate for system 1 is 13.3% less than that for system 2.
336 Also, it can be seen that the ratio of cost rate due to GHG emission for system 2 is 2.9% less
337 than that for system 1 when both systems operate at the optimal P_{int} . This indicates that the
338 investment, operating and maintenance costs of system 2 exceed those for system 1.

339 Fig. 13 shows the variation of system COP and exergy efficiency with superheating degree of
340 CO₂ at evaporator outlet. A small decrease of mass flow rate and a large increase in specific
341 consumed work of LPC I lead to decreases in the COP and exergy efficiency. Under the
342 given conditions, a 10 K increase in ΔT_{sup} from 0 to 10 K leads to decreases of 0.9% and up
343 to 2.3% for systems 1 and 2, respectively, in both COP and exergy efficiency. Furthermore,
344 system 1 leads to increases of up to 3.8% for both system COP and exergy efficiency in
345 comparison to system 2, at $\Delta T_{\text{sup}} = 10$ K.

346 Fig. 14 illustrates the effect of varying degree of superheating of CO₂ at the evaporator outlet
347 on the ratio of cost rate due to GHG emission and total annual cost rate. By increasing the
348 CO₂ superheating degree, the electrical power consumption of LTC I and the heat transfer
349 surface area of the evaporator increase, leading to a rise in both the penalty cost rate of GHG
350 emission and the system capital and total costs.

351 Fig. 15 displays the variation of the ratio of GHG emission cost rate and total annual cost rate
352 with the cost of CO₂ avoided. The cost of CO₂ avoided varies significantly for different types

353 of power plants [28]. The results show that the penalty cost rate of GHG emission and total
354 cost rate are sensitive to c_{CO_2} . For system 1, increasing the cost of CO₂ avoided from 30 \$
355 ton⁻¹ of CO₂e to 120 \$ ton⁻¹ of CO₂e (300%) leads to an increase of 29.7% and 25.5% in
356 total cost rate for systems 1 and 2, respectively.

357 In order to optimize the performances of systems 1 and 2, from the thermodynamic and
358 economic viewpoints, the DIRECT algorithm in the EES software has been used. The values
359 of operating parameters for the thermodynamic optimal design case and parameters affecting
360 the total annual cost are for the cost optimal design case are summarized in Tables 5 and 6,
361 respectively. The results of thermodynamic and economic optimizations show that the values
362 of COP and exergy efficiency for the compared systems are almost the same. Yet, the total
363 annual cost rate for system 1 is 11.2% and 11.9% lower than that for system 2 referring to
364 thermodynamic and economic optimizations, respectively. Comparing the thermodynamic
365 and cost optimal design conditions for system 1, an increase of 14.8% in COP and exergy
366 efficiency is achieved at the expense of a 3.0% increment in the total annual cost rate, when
367 the optimization is based on the maximum COP. This comparison for system 2 shows an
368 increase of 11.6% in COP and exergy efficiency and 2.1% in the total annual cost rate.

369 Fig. 16 shows the values of exergy destruction ratio for various components of the proposed
370 cascade cycles at thermodynamically optimal design condition. As can be seen, the highest
371 value of exergy destruction rate is attributable to Fan I in both cycles (44.9 kW). The high
372 mass flow rate and a temperature of cooled air lower than the ambient temperature lead to a
373 high entropy generation and so a high irreversibility for Fan I. After that, LTC II of both
374 cycles has the highest value of exergy destruction rate (31.1 kW in system 1 and 26.8 kW in
375 system 2), due to the compression process. After NH₃ flash tank, the lowest value of exergy
376 destruction rate is associated with the CO₂ flash tank for system 1 (2.7 kW) and TV II for the
377 system 2 (1.6 kW) due to negligible heat losses and a throttling process at low pressure.

378 Fig. 17 displays the values of exergy destruction ratio for various components of cascade
379 systems 1 and 2 at cost optimal design condition. After Fan I with its exergy destruction rate
380 of 44.9 kW, the evaporator of both cycles has the highest value of exergy destruction rate
381 (41.1 kW), due to the large temperature difference between the cooled air and CO₂.
382 The results obtained from the exergoeconomic analysis of the CO₂/NH₃ cascade
383 refrigeration, systems 1 and 2, for the thermodynamic and economic optimum conditions are
384 presented in Tables 7 and 8, respectively. The results of both thermodynamic and economic
385 optimizations show that, after the NH₃ flash tank in the compared cycles, the CO₂ flash tank
386 has the highest value of exergy efficiency for system 1 (about 97%) and the FIS the highest
387 value for system 2 (about 92%). The lowest exergy efficiencies are observed for the Fan II
388 (1.67%) and condenser (about 3%) for both cycles. For thermodynamic and economic
389 optimum conditions, the low value of f for the flash tank of system 1 (about 1.8%) and
390 cascade heat exchanger in the system 2 (about 3%) indicate that the costs associated with
391 these components are almost exclusively due to exergy destructions. The exergoeconomic
392 factor of the NH₃ flash tank of 100% for both cycles and the relatively large value of f for
393 the evaporator in system 1 suggests that the capital investment, operating and maintenance
394 costs dominate.

395 **6. Conclusions**

396 Exergoeconomic and environmental analyses are successfully carried out for two different
397 CO₂/NH₃ cascade refrigeration systems equipped with two flash tanks and a flash tank along
398 with a flash intercooler with an indirect subcooler. To determine the maximum value of COP
399 and exergy efficiency and the minimum cost rate due to GHG emission and total cost rate of
400 the system, the following operating parameters are considered: condensing temperature of
401 NH₃ in condenser and CO₂ in cascade heat exchanger, evaporating temperature of CO₂ in
402 evaporator, temperature difference in the cascade heat exchanger, intermediate pressure in the

403 CO₂ low-temperature circuit and mass flow rate ratio of the FIS. From the energy, exergy,
404 economic and environmental analyses the following results are obtained and conclusions
405 drawn:

- 406 • By using the FIS in CO₂ low-temperature circuit instead of the FT, the performance
407 of the CO₂/NH₃ cascade refrigeration system is decreased.
- 408 • For system 1, a 10 K increase in T_{EV} leads to increases of 16.9% in both COP and
409 exergy efficiency. At the same conditions for system 2, a 10 K increase in T_{EV} leads
410 to maximum increase of 20.8% in both COP and exergy efficiency when $r = 3.6$.
411 Also, using a FIS in the system leads to an increase of 14.3% in the total cost rate.
- 412 • The minimum annual total cost rate is obtained at a CO₂ evaporating temperature of
413 -41.5 °C and -40 °C, respectively for systems 1 and 2 when $r = 3.2$.
- 414 • For system 1, a 10 K increase in T_{CD} leads to a 13.4% decrease in both COP and
415 exergy efficiency. At the same condition for system 2, a 10 K increase in T_{CD} leads to
416 maximum decrease of 13.3% in both COP and exergy efficiency when $r = 3.2$. Also,
417 in systems 1 and 2, a 10 K increase in T_{CD} leads to a 4.8% and 4.2% increase
418 respectively in total cost rate.
- 419 • The maximum COP and exergy efficiency are obtained at a CO₂ condensing
420 temperature of 1.9 °C and 2.1 °C, respectively for systems 1 and 2 when $r = 3.2$.
- 421 • The total annual cost rate for the system 1 is 11.2% and 11.9% lower than that for the
422 system 2 referring to thermodynamic and economic optimizations, respectively.
- 423 • The lowest value of the exergoeconomic factor is 1.73% for CO₂ flash tank in system
424 1 and 3.85% for cascade heat exchanger in system 2, demonstrating that the costs
425 associated with CO₂ flash tank and cascade heat exchanger in systems 1 and 2
426 respectively are almost exclusively due to exergy destruction.

427 • The highest exergoeconomic factor is observed to be 100% for the NH₃ flash tank in
428 both systems, suggesting that the capital investment, operating and maintenance costs
429 of the FT in the high-temperature circuit dominate in such cases.

430 The present study demonstrates the benefits and profitability of CO₂/NH₃ cascade
431 refrigeration systems equipped with a flash tank and a flash intercooler, with and without an
432 indirect subcooler. However, a more detailed system design considering heat and pressure
433 losses in all system components and using more accurate cost functions are suggested for
434 further investigation.

435 References

- 436 [1] I. Dincer, M. Kanoglu, Refrigeration systems and applications, Wiley, 2010.
- 437 [2] H.J. Dakkama, A. Elsayed, R.K. Al-Dadah, S.M. Mahmoud, P. Youssef, Investigation
438 of cascading adsorption refrigeration system with integrated evaporator-condenser heat
439 exchanger using different working pairs, Energy Procedia. 75 (2015) 1496–1501.
440 doi:10.1016/j.egypro.2015.07.285.
- 441 [3] A.M. Dubey, S. Kumar, G. Das Agrawal, Thermodynamic analysis of a transcritical
442 CO₂/propylene (R744–R1270) cascade system for cooling and heating applications,
443 Energy Convers. Manag. 86 (2014) 774–783. doi:10.1016/j.enconman.2014.05.105.
- 444 [4] W. Han, L. Sun, D. Zheng, H. Jin, S. Ma, X. Jing, New hybrid absorption–
445 compression refrigeration system based on cascade use of mid-temperature waste heat,
446 Appl. Energy. 106 (2013) 383–390. doi:10.1016/j.apenergy.2013.01.067.
- 447 [5] V. Jain, S.S. Kachhwaha, G. Sachdeva, Thermodynamic performance analysis of a
448 vapor compression–absorption cascaded refrigeration system, Energy Convers. Manag.
449 75 (2013) 685–700. doi:10.1016/j.enconman.2013.08.024.
- 450 [6] A. Kilicarslan, M. Hosoz, Energy and irreversibility analysis of a cascade refrigeration
451 system for various refrigerant couples, Energy Convers. Manag. 51 (2010) 2947–2954.
452 doi:10.1016/j.enconman.2010.06.037.
- 453 [7] C.A. Infante Ferreira, R.A. Boukens, Carbon dioxide – secondary coolant or
454 refrigerant for cascade systems?, in: Proc. IIR Conf. Appl. Nat. Refrig., Aarhus,
455 Denmark, Denmark, 1996: pp. 185–194.
- 456 [8] T.S. Lee, C.H. Liu, T.W. Chen, Thermodynamic analysis of optimal condensing
457 temperature of cascade-condenser in CO₂/NH₃ cascade refrigeration systems, Int. J.
458 Refrig. 29 (2006) 1100–1108. doi:10.1016/j.ijrefrig.2006.03.003.
- 459 [9] H.M. Getu, P.K. Bansal, Thermodynamic analysis of an R744-R717 cascade
460 refrigeration system, Int. J. Refrig. 31 (2008) 45–54.
461 doi:10.1016/j.ijrefrig.2007.06.014.
- 462 [10] J.A. Dopazo, J. Fernández-Seara, J. Sieres, F.J. Uhía, Theoretical analysis of a CO₂-
463 NH₃ cascade refrigeration system for cooling applications at low temperatures, Appl.

- 464 Therm. Eng. 29 (2009) 1577–1583. doi:10.1016/j.applthermaleng.2008.07.006.
- 465 [11] W. Bingming, W. Huagen, L. Jianfeng, X. Ziwen, Experimental investigation on the
466 performance of NH₃/CO₂ cascade refrigeration system with twin-screw compressor,
467 Int. J. Refrig. 32 (2009) 1358–1365. doi:10.1016/j.ijrefrig.2009.03.008.
- 468 [12] J.A. Dopazo, J. Fernández-Seara, Experimental evaluation of a cascade refrigeration
469 system prototype with CO₂ and NH₃ for freezing process applications, Int. J. Refrig.
470 34 (2011) 257–267. doi:10.1016/j.ijrefrig.2010.07.010.
- 471 [13] M. Ma, J. Yu, X. Wang, Performance evaluation and optimal configuration analysis of
472 a CO₂/NH₃ cascade refrigeration system with falling film evaporator–condenser,
473 Energy Convers. Manag. 79 (2014) 224–231. doi:10.1016/j.enconman.2013.12.021.
- 474 [14] R.S. Mitshita, E.M. Barreira, C.O.R. Negrão, C.J.L. Hermes, Thermoeconomic design
475 and optimization of frost-free refrigerators, Appl. Therm. Eng. 50 (2013) 1376–1385.
476 doi:10.1016/j.applthermaleng.2012.06.024.
- 477 [15] H. Esen, M. Inalli, M. Esen, Technoeconomic appraisal of a ground source heat pump
478 system for a heating season in eastern Turkey, Energy Convers. Manag. 47 (2006)
479 1281–1297. doi:10.1016/j.enconman.2005.06.024.
- 480 [16] H. Esen, M. Inalli, M. Esen, A techno-economic comparison of ground-coupled and
481 air-coupled heat pump system for space cooling, Build. Environ. 42 (2007) 1955–
482 1965. doi:10.1016/j.buildenv.2006.04.007.
- 483 [17] H. Esen, M. Inalli, M. Esen, K. Pihili, Energy and exergy analysis of a ground-
484 coupled heat pump system with two horizontal ground heat exchangers, Build.
485 Environ. 42 (2007) 3606–3615. doi:10.1016/j.buildenv.2006.10.014.
- 486 [18] O. Rezayan, A. Behbahaninia, Thermoeconomic optimization and exergy analysis of
487 CO₂/NH₃ cascade refrigeration systems, Energy. 36 (2011) 888–895.
488 doi:10.1016/j.energy.2010.12.022.
- 489 [19] H. Wijnnga, M. Van Der Hoff, M. Janssen, C.A. Infante Ferreira, Life cycle
490 performance of refrigeration systems in the Dutch food and beverages sector, in: 4th
491 IIR Conf. Thermophys. Prop. Transf. Process. Refrig., Delft, The Netherlands, The
492 Netherlands, 2013.
- 493 [20] S. Fettaka, J. Thibault, Y. Gupta, Design of shell-and-tube heat exchangers using
494 multiobjective optimization, Int. J. Heat Mass Transf. 60 (2013) 343–354.
495 doi:10.1016/j.ijheatmasstransfer.2012.12.047.
- 496 [21] C.A. Infante Ferreira, Refrigeration and heat pumping in food processing plants, in: L.
497 Stougie (Ed.), Energy Effic. Qual. Energy Food Process. Ind., Interduct: Delft
498 University of Technology, Delft, The Netherlands, 2002: pp. 57–79.
- 499 [22] T. Morosuk, G. Tsatsaronis, Advanced exergetic evaluation of refrigeration machines
500 using different working fluids, Energy. 34 (2009) 2248–2258.
501 doi:10.1016/j.energy.2009.01.006.
- 502 [23] M. Navidbakhsh, A. Shirazi, S. Sanaye, Four E analysis and multi-objective
503 optimization of an ice storage system incorporating PCM as the partial cold storage for
504 air-conditioning applications, Appl. Therm. Eng. 58 (2013) 30–41.
505 doi:http://dx.doi.org/10.1016/j.applthermaleng.2013.04.002.
- 506 [24] A. Bejan, G. Tsatsaronis, M.J. Moran, Thermal design and optimization, Wiley, 1996.
- 507 [25] F. Czesla, G. Tsatsaronis, Z. Gao, Avoidable thermodynamic inefficiencies and costs
508 in an externally fired combined cycle power plant, Energy. 31 (2006) 1472–1489.

- 509 doi:10.1016/j.energy.2005.08.001.
- 510 [26] G. Tsatsaronis, M.-H. Park, On avoidable and unavoidable exergy destructions and
511 investment costs in thermal systems, *Energy Convers. Manag.* 43 (2002) 1259–1270.
512 doi:http://dx.doi.org/10.1016/S0196-8904(02)00012-2.
- 513 [27] J. Wang, Z. Zhai, Y. Jing, C. Zhang, Particle swarm optimization for redundant
514 building cooling heating and power system, *Appl. Energy.* 87 (2010) 3668–3679.
515 doi:http://dx.doi.org/10.1016/j.apenergy.2010.06.021.
- 516 [28] E.S. Rubin, J.E. Davison, H.J. Herzog, The cost of CO₂ capture and storage, *Int. J.*
517 *Greenh. Gas Control.* 40 (2015) 378–400. doi:10.1016/j.ijggc.2015.05.018.
- 518 [29] M. Aminyavari, B. Naja, A. Shirazi, F. Rinaldi, Exergetic, economic and
519 environmental (3E) analyses, and multi- objective optimization of a CO₂/NH₃ cascade
520 refrigeration system, *Appl. Therm. Eng.* 65 (2014) 42–50.
521 doi:10.1016/j.applthermaleng.2013.12.075.
- 522 [30] G. Xu, F. Liang, Y. Yang, Y. Hu, K. Zhang, W. Liu, An improved CO₂ separation and
523 purification system based on cryogenic separation and distillation theory, *Energies.* 7
524 (2014) 3484–3502. doi:10.3390/en7053484.
- 525 [31] S. Sanaye, A. Shirazi, Four E analysis and multi-objective optimization of an ice
526 thermal energy storage for air-conditioning applications, *Int. J. Refrig.* 36 (2012) 1–14.
527 doi:10.1016/j.ijrefrig.2012.10.014.
- 528 [32] A.H. Mosaffa, L. Garousi Farshi, Exergoeconomic and environmental analyses of an
529 air conditioning system using thermal energy storage, *Appl. Energy.* 162 (2016) 515–
530 526. doi:10.1016/j.apenergy.2015.10.122.
- 531

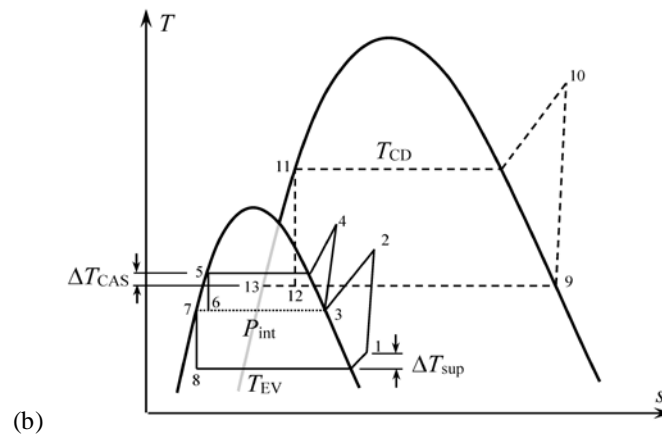
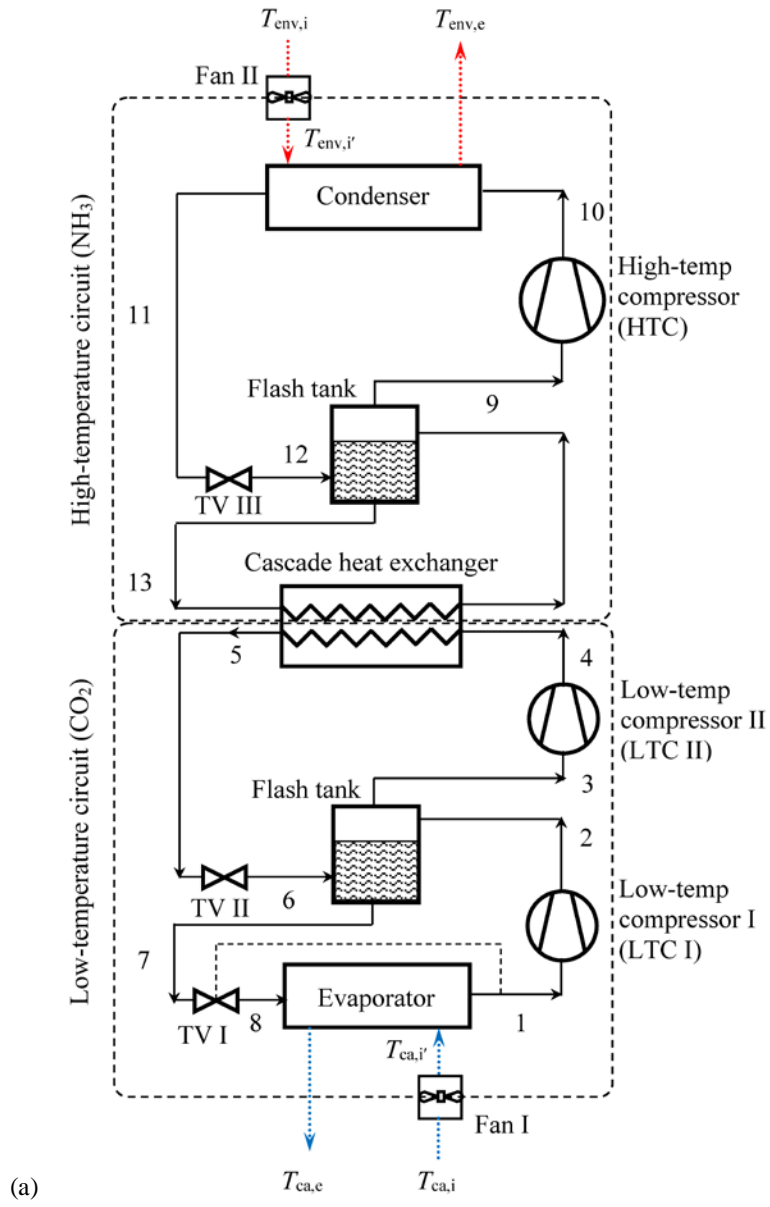


Fig. 1. (a) Schematic and (b) T - s diagram for the CO_2/NH_3 cascade refrigeration cycle equipped with two flash tanks (system 1).

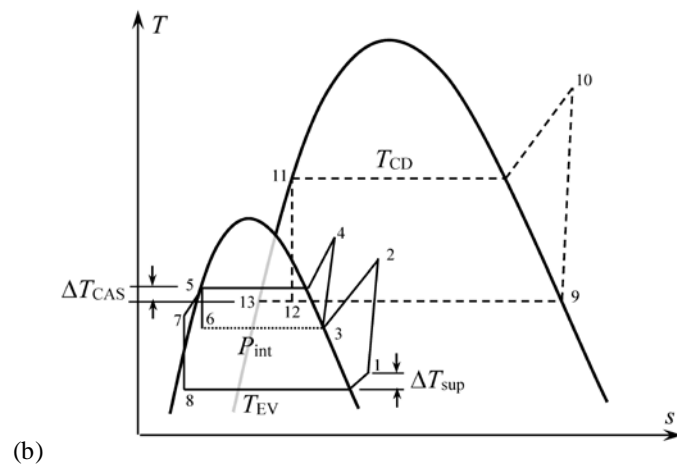
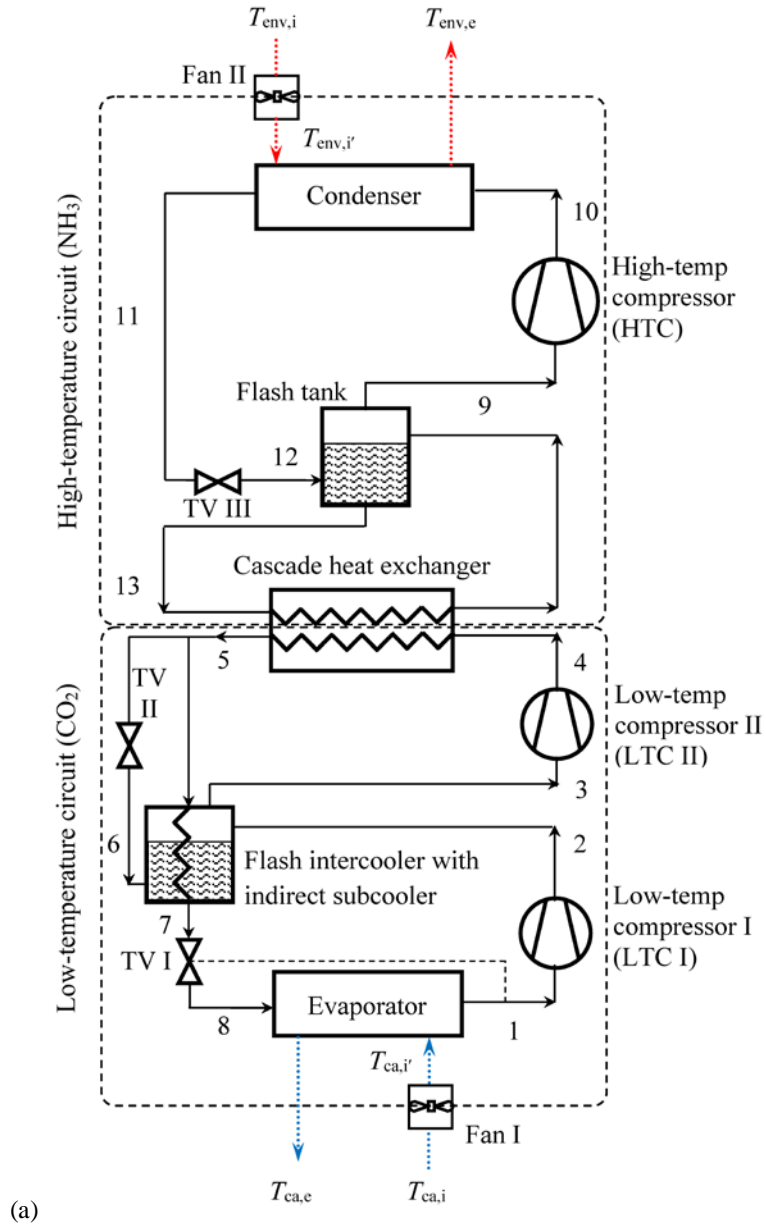


Fig. 2. (a) Schematic and (b) T - s diagram for the CO_2/NH_3 cascade refrigeration cycle equipped with a flash tank and a flash intercooler with an indirect subcooler (system 2).

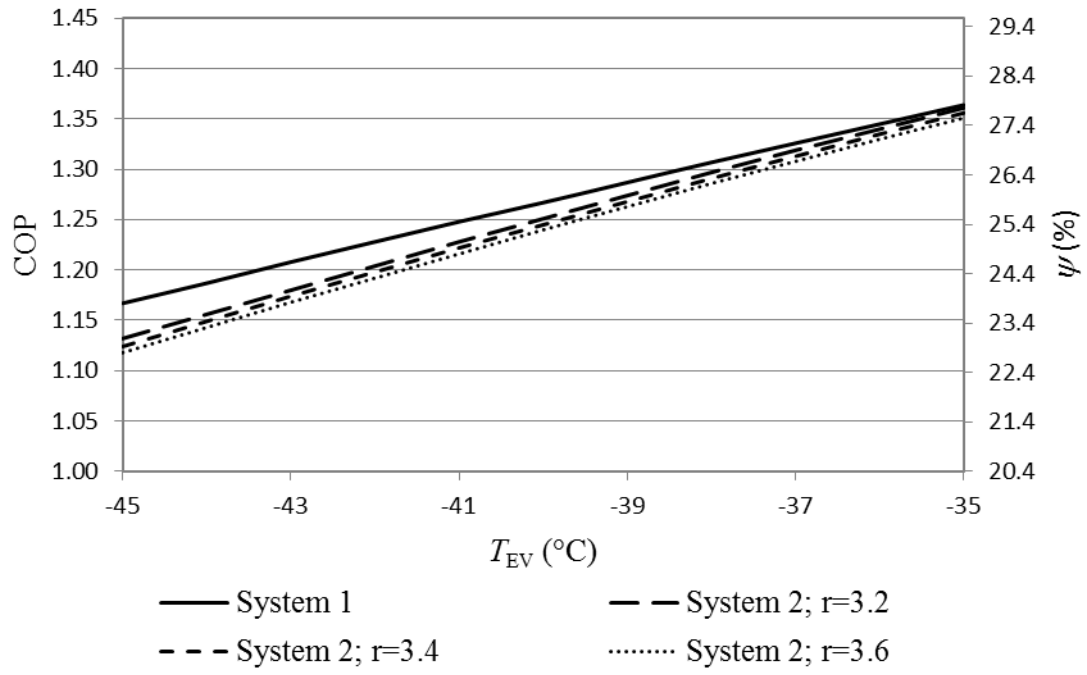


Fig. 3. Variation of system COP and exergy efficiency with CO_2 evaporating temperature.

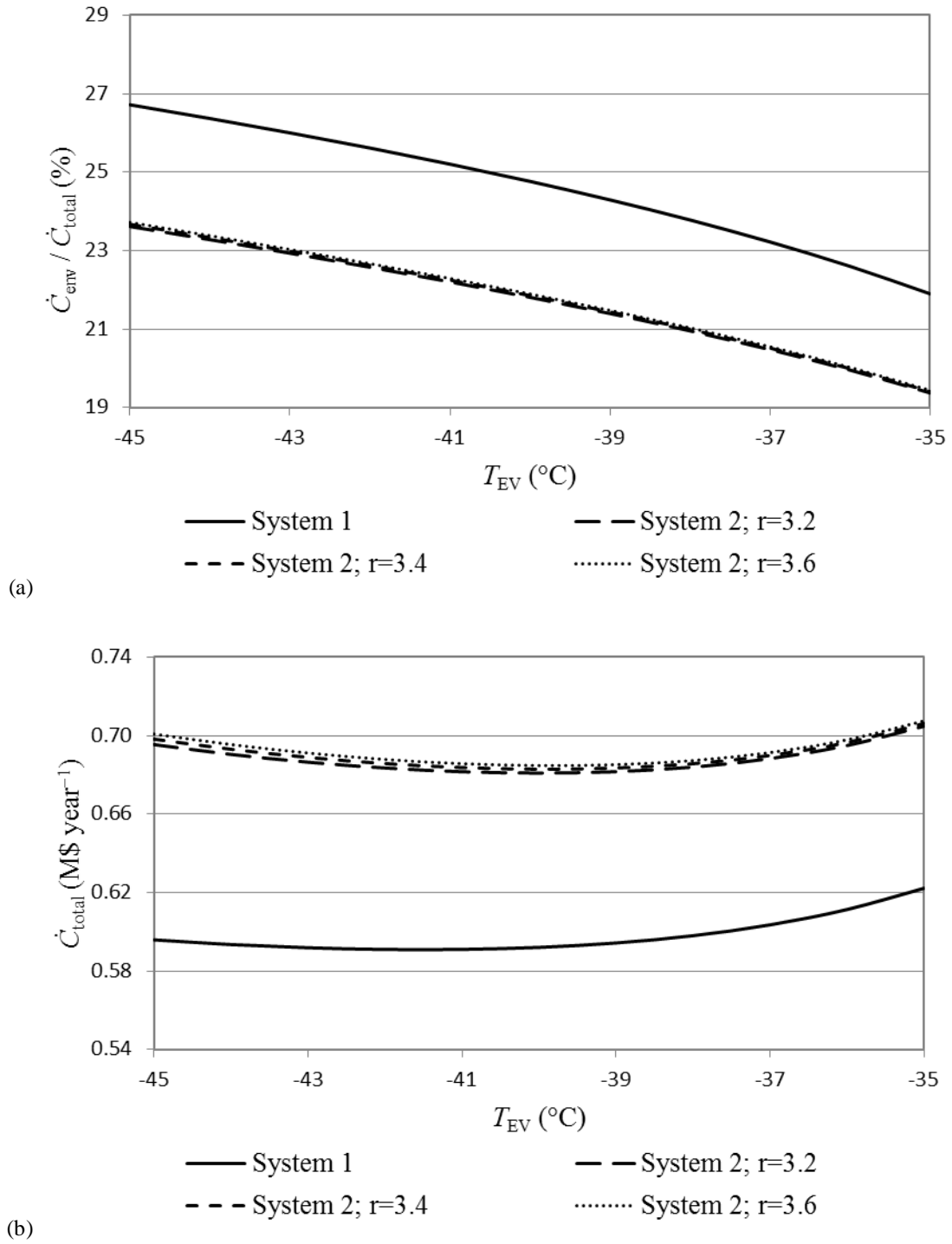


Fig. 4. Effect of CO₂ evaporating temperature on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

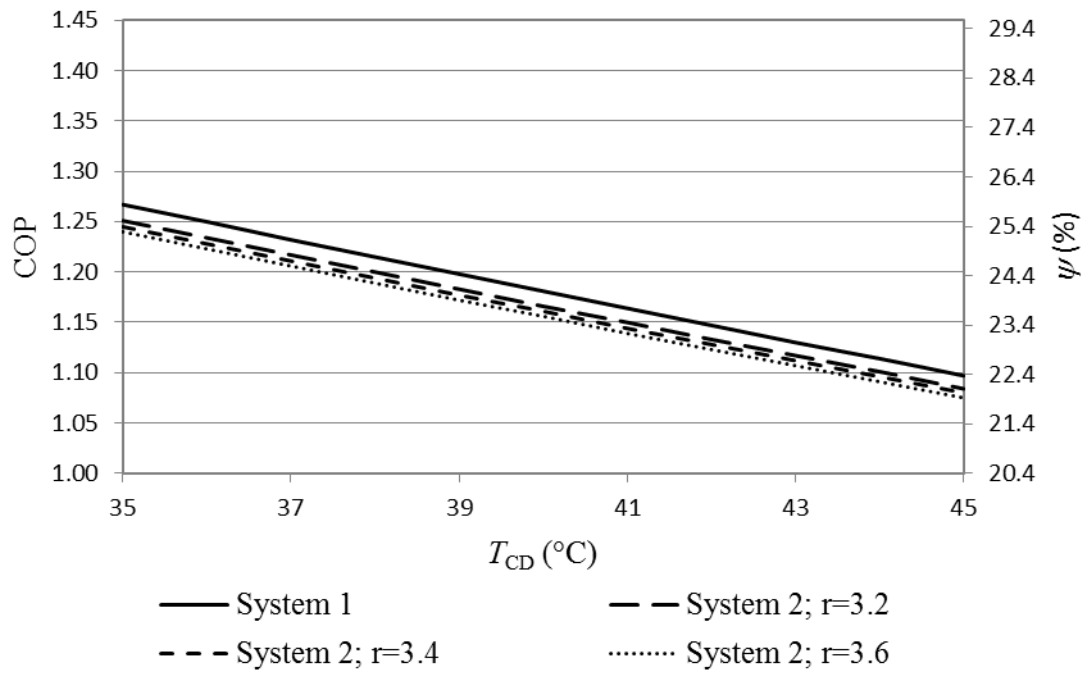


Fig. 5. Variation of system COP and exergy efficiency with NH_3 condensing temperature.

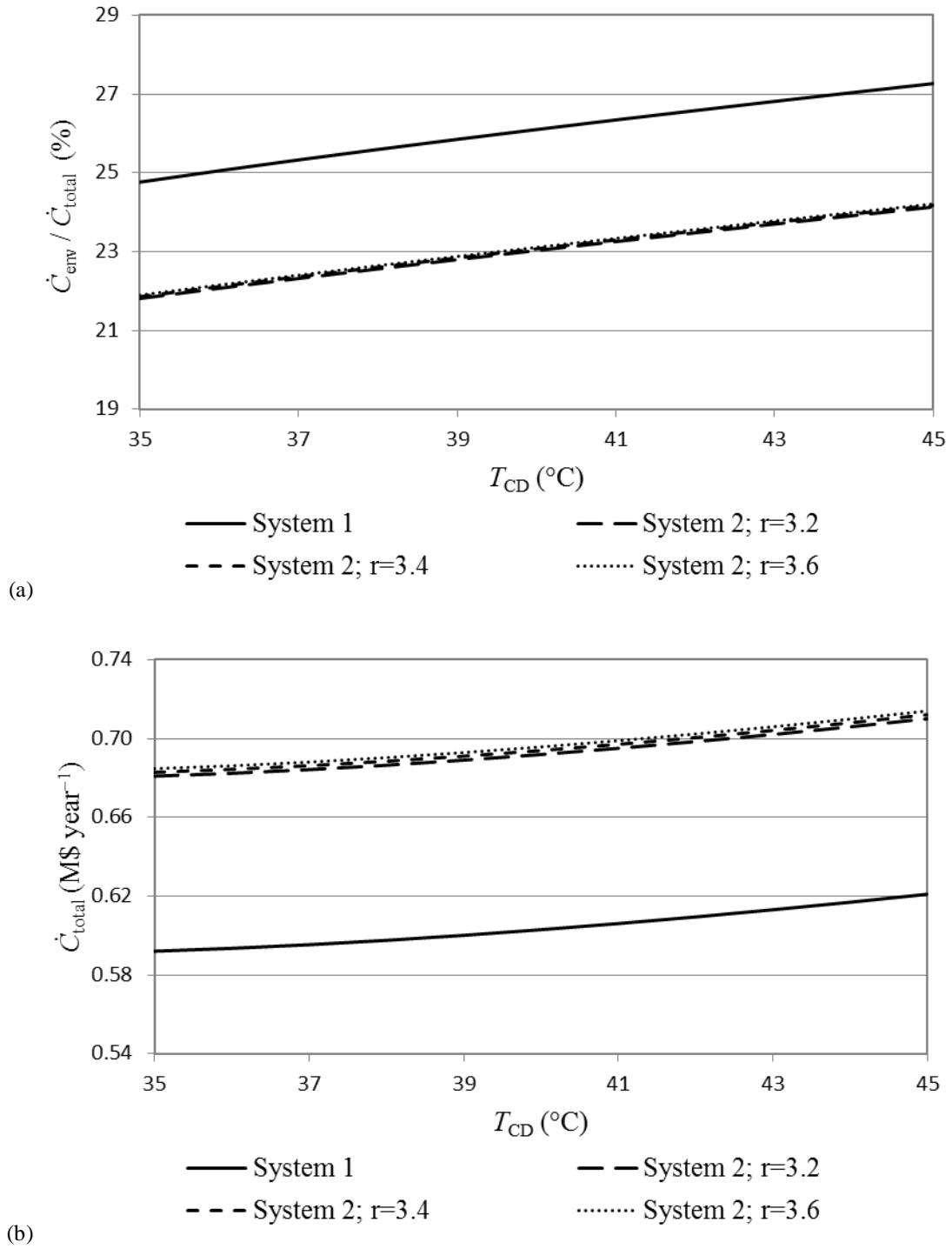


Fig. 6. Effect of varying NH_3 condensing temperature on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

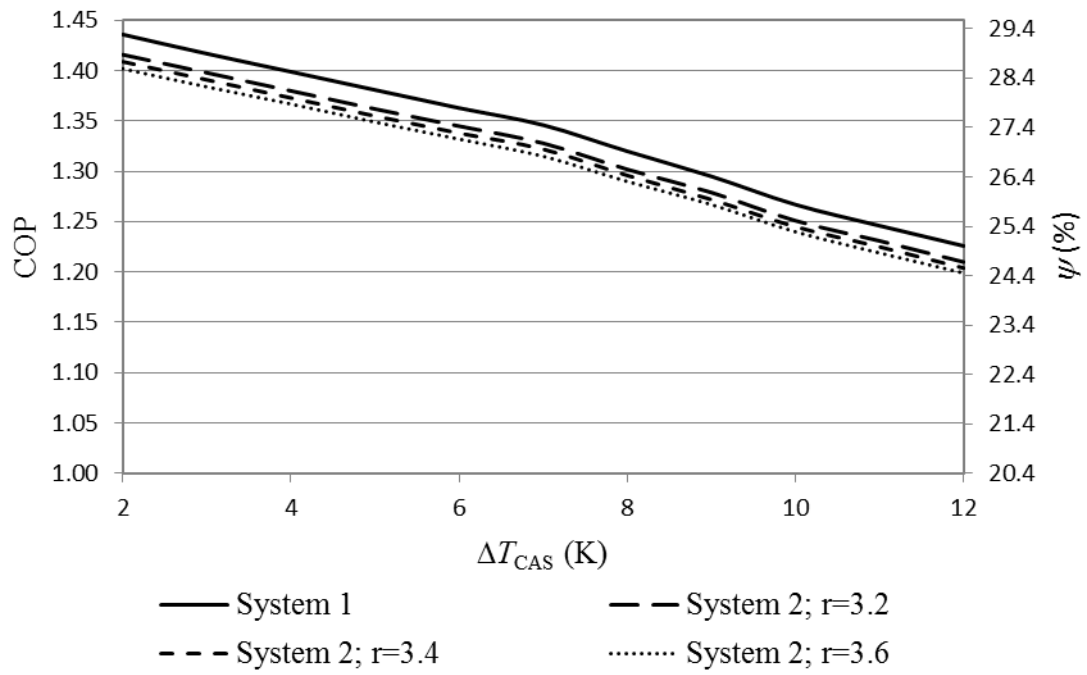


Fig. 7. Variation of system COP and exergy efficiency with cascade heat exchanger temperature difference.

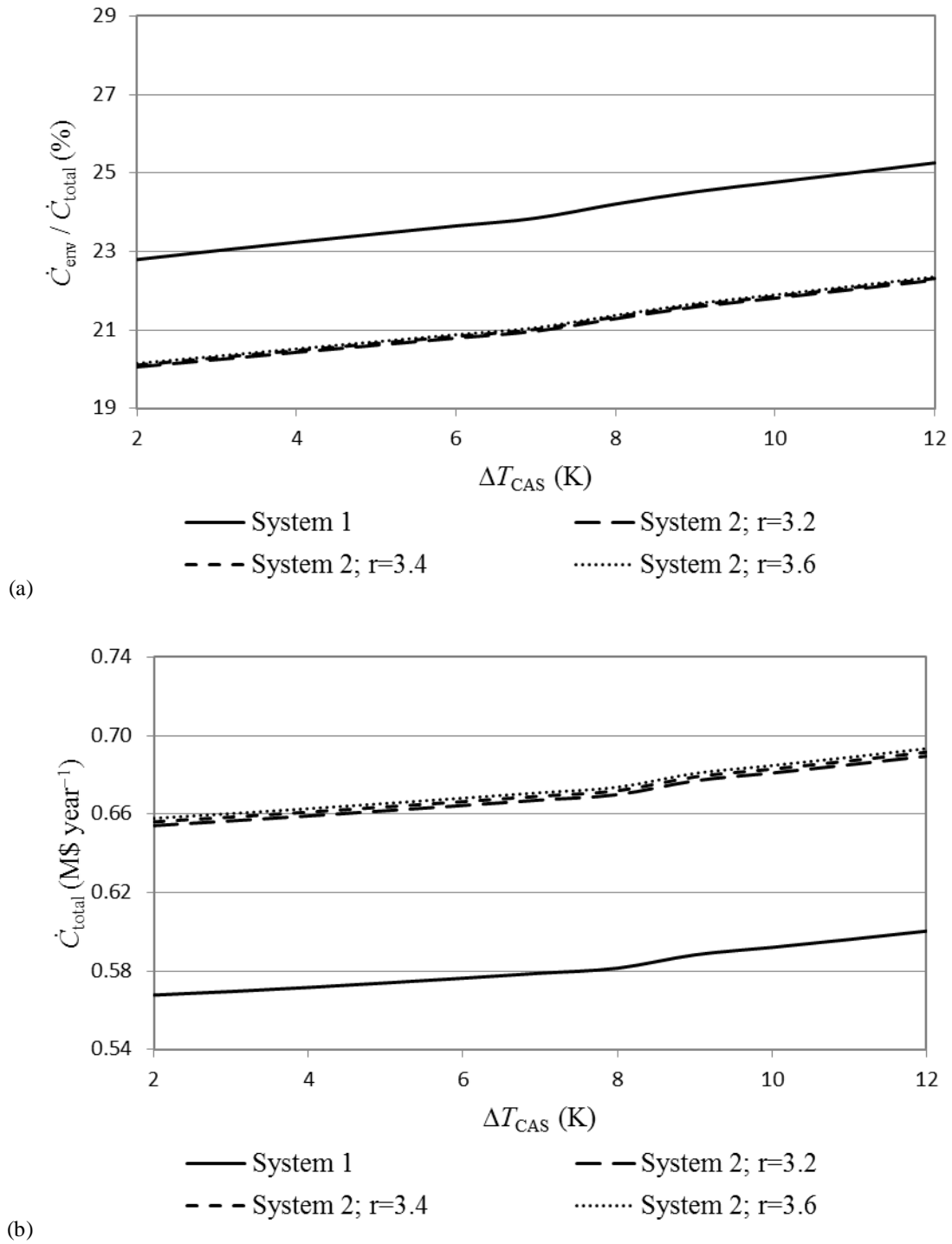


Fig. 8. Effect of varying cascade heat exchanger temperature difference on (a) the ratio of penalty cost of GHG emission and (b) total annual cost rate.

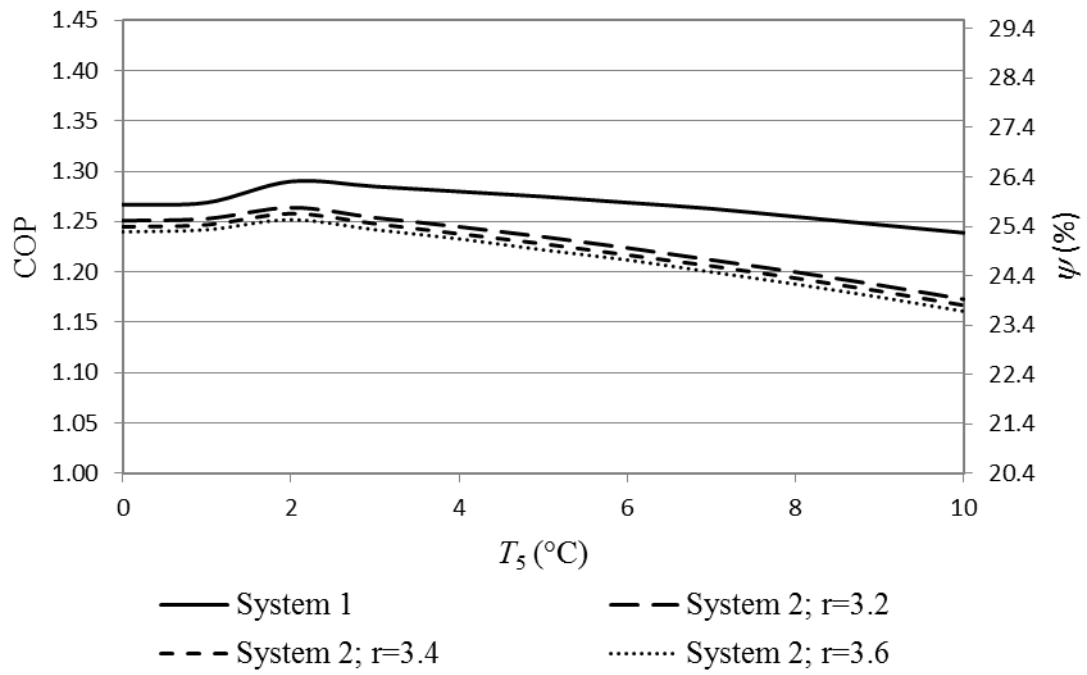


Fig. 9. Variation of system COP and exergy efficiency with CO₂ condensing temperature.

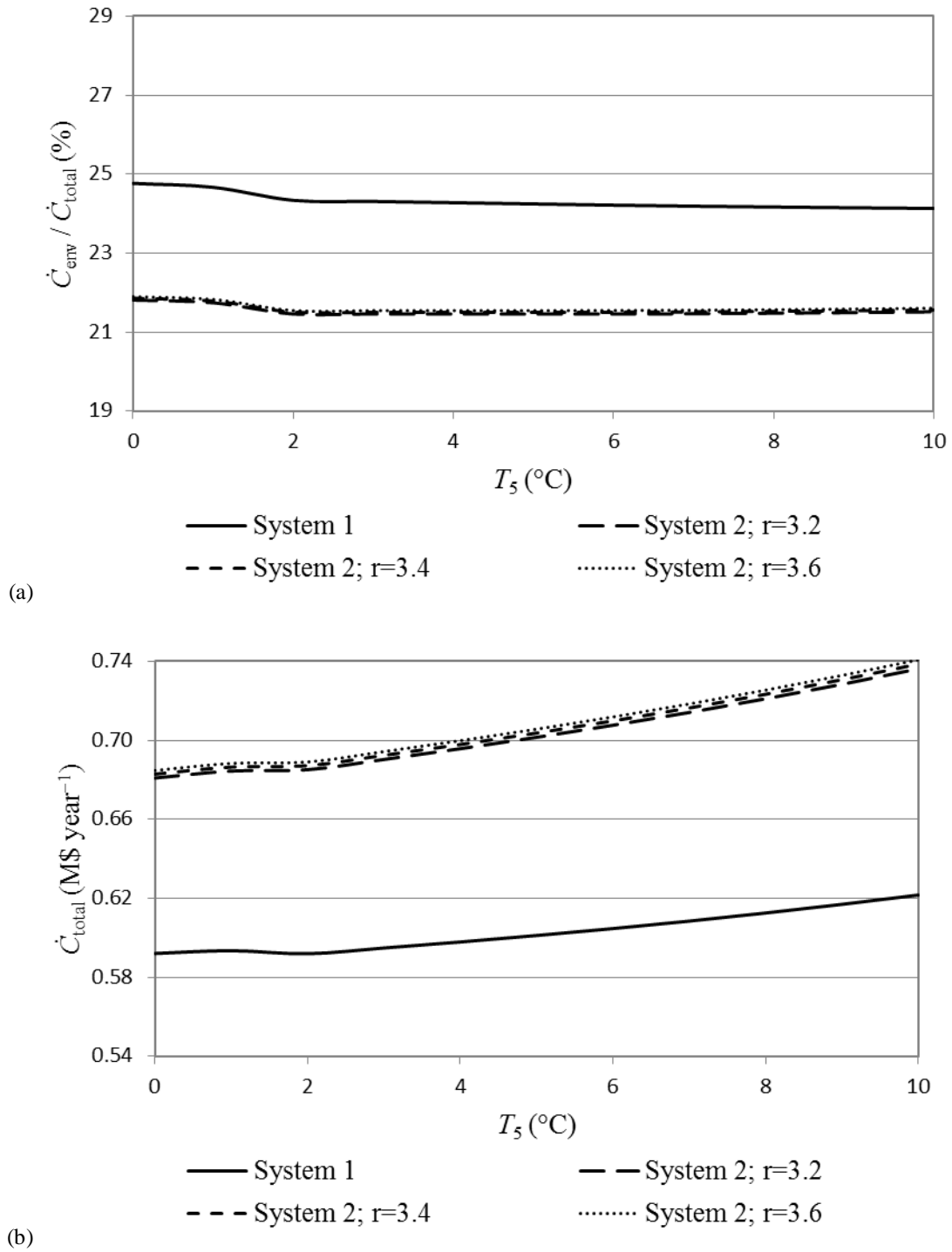


Fig. 10. Effect of varying CO₂ condensing temperature on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

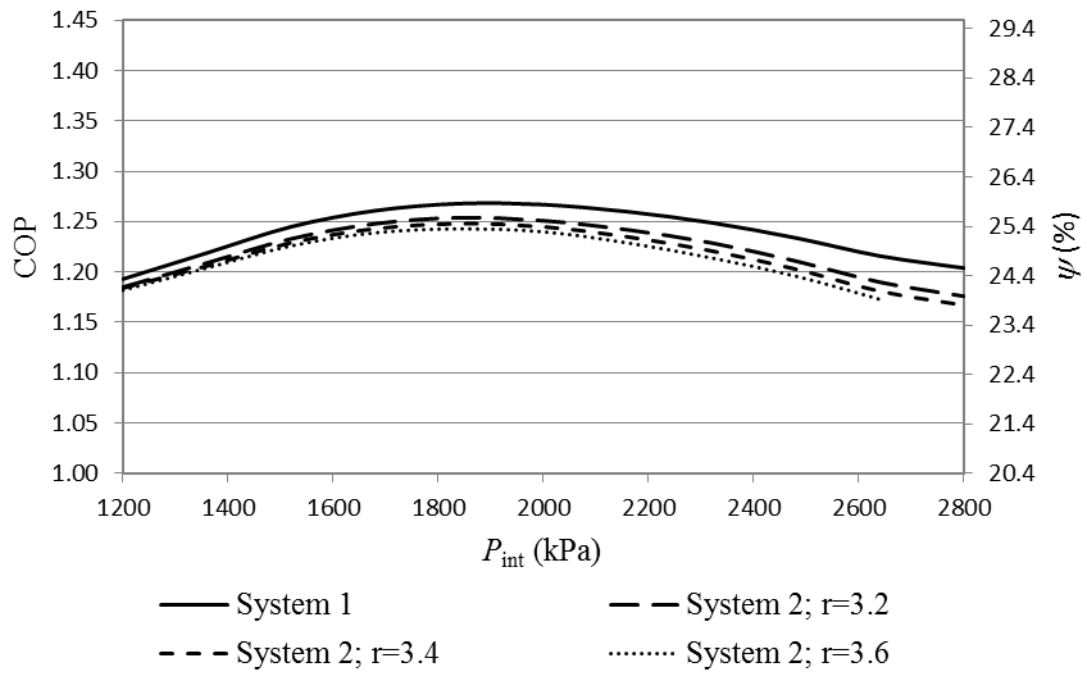


Fig. 11. Variation of system COP and exergy efficiency with intermediate pressure in the low-temperature circuit.

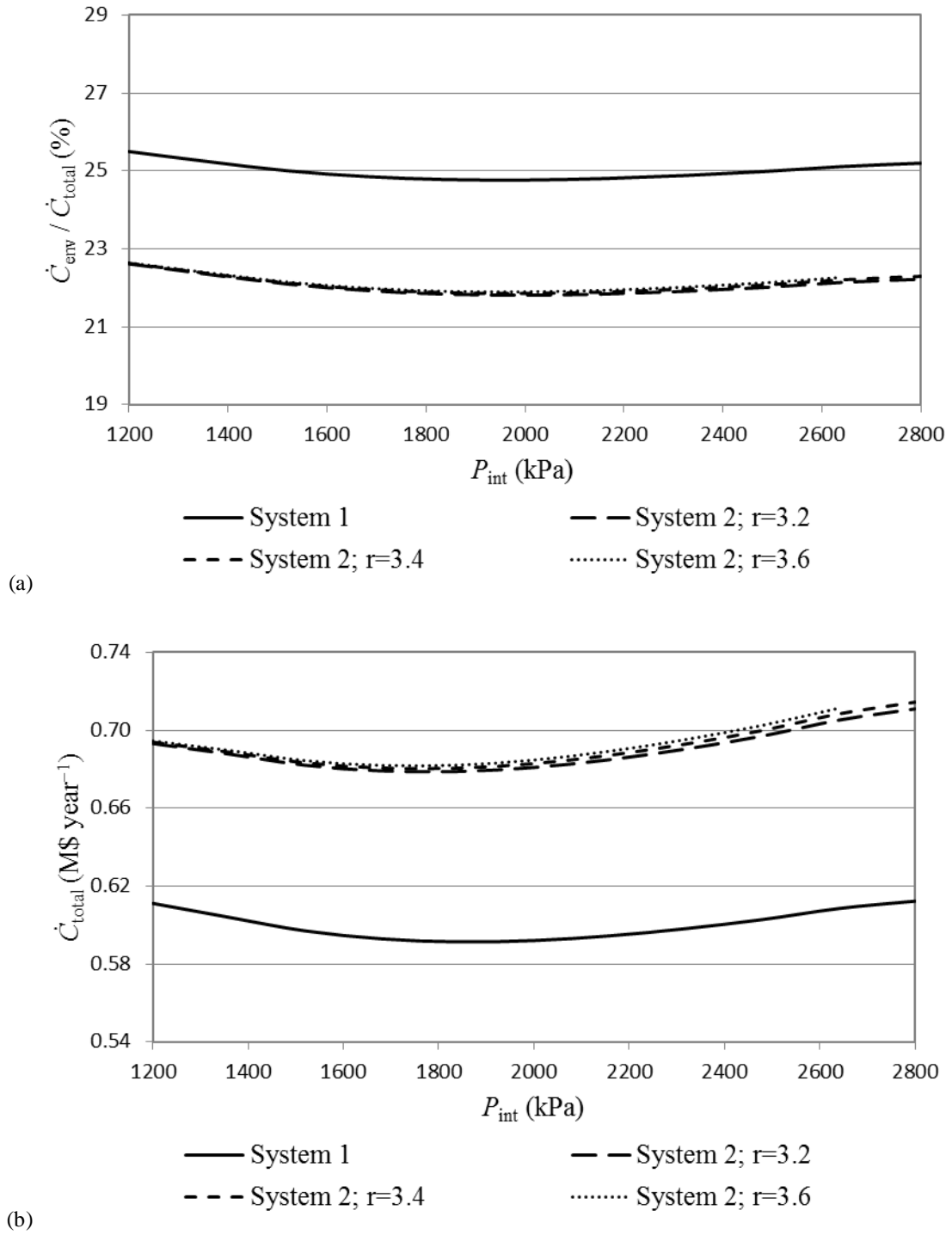


Fig. 12. Effect of varying intermediate pressure in the low-temperature circuit on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

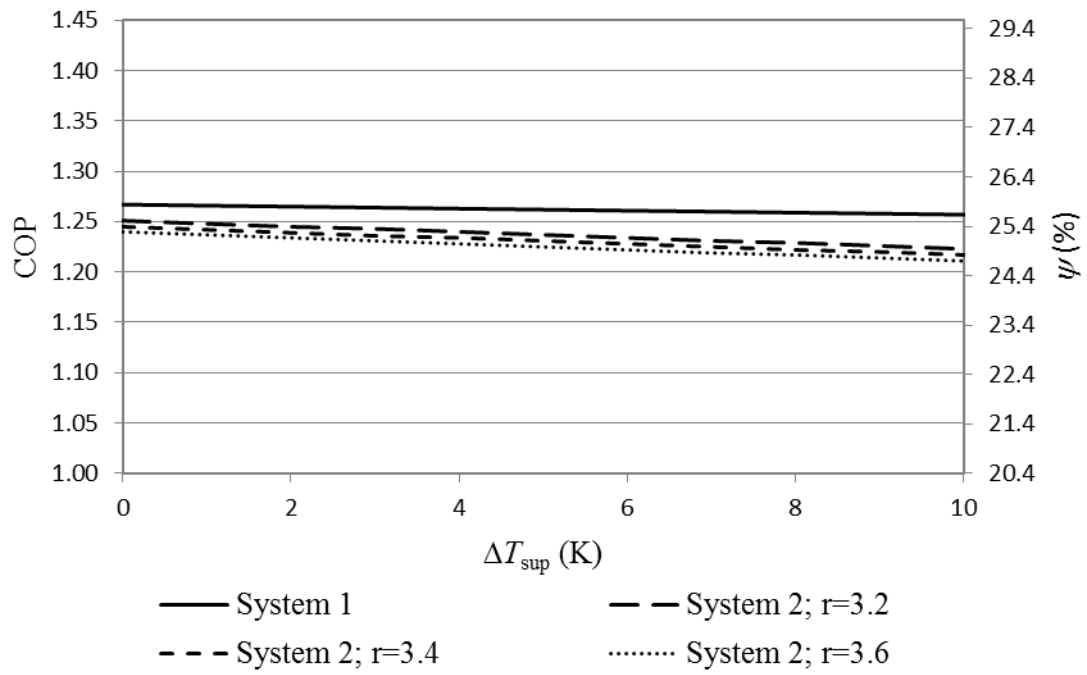


Fig. 13. Variation of system COP and exergy efficiency with degree of superheating of CO₂ at evaporator outlet.

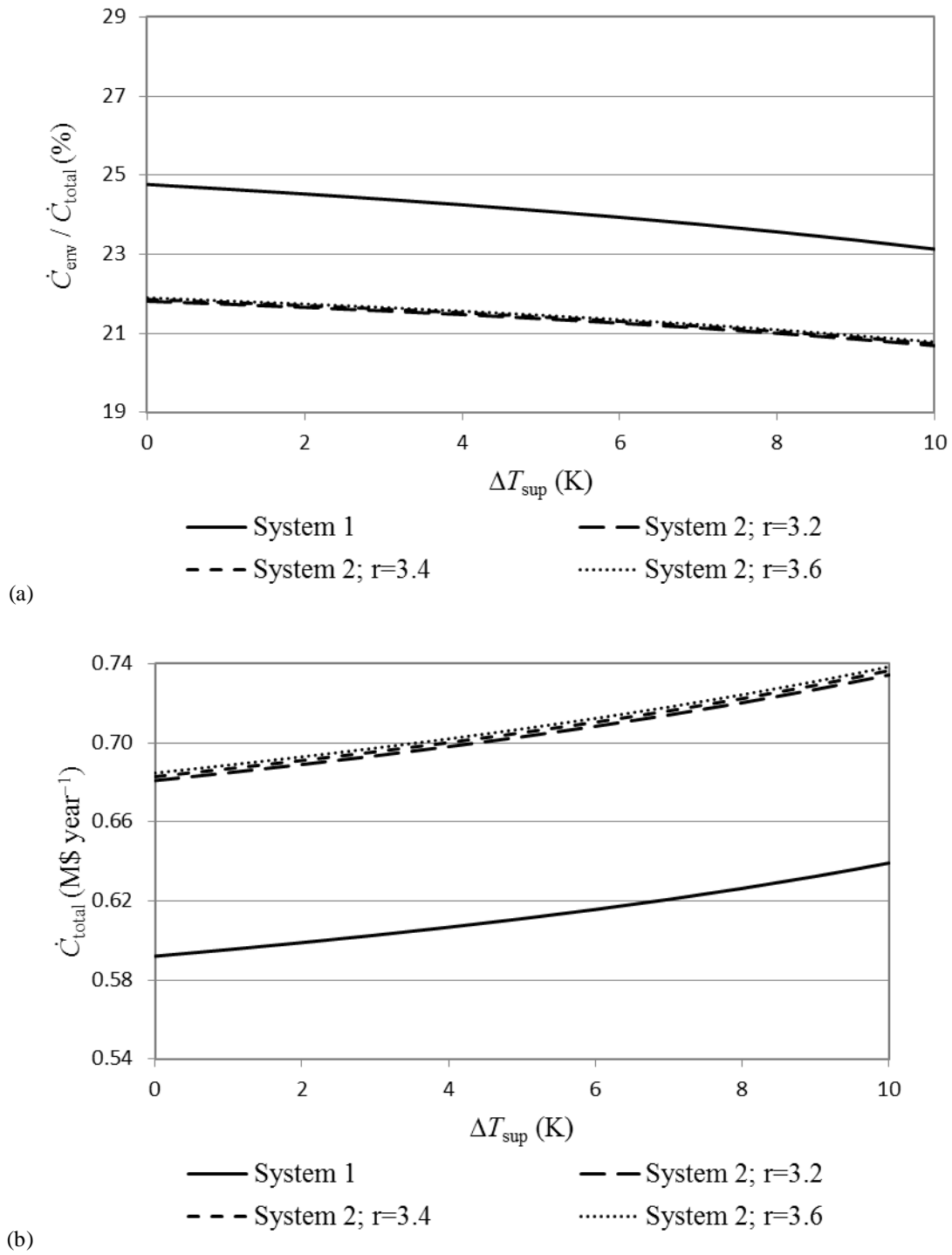


Fig. 14. Effect of varying superheating degree of CO₂ at evaporator outlet on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

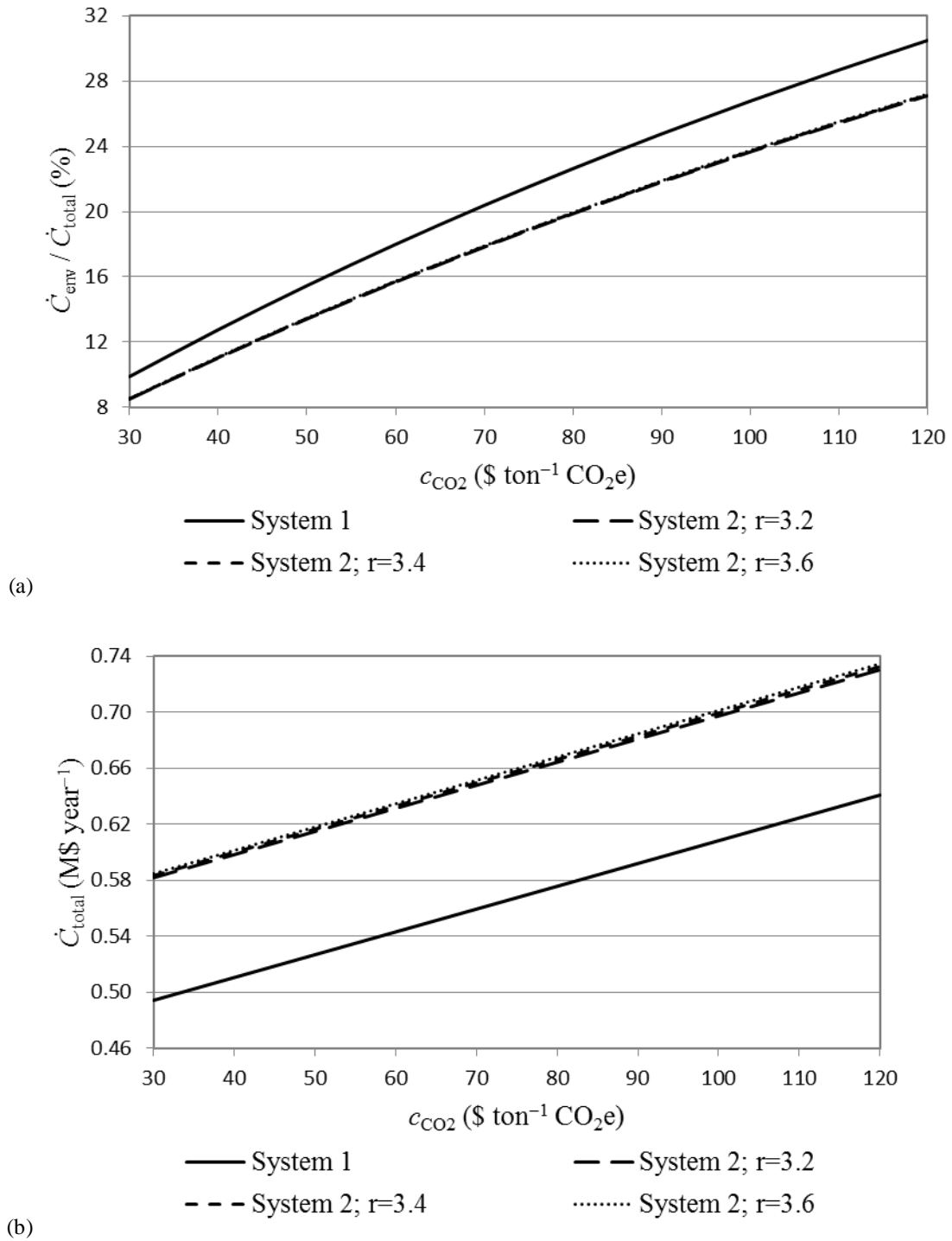


Fig. 15. Effect of varying cost of CO₂ avoided on (a) the ratio of penalty cost of GHG emission and (b) the total annual cost rate.

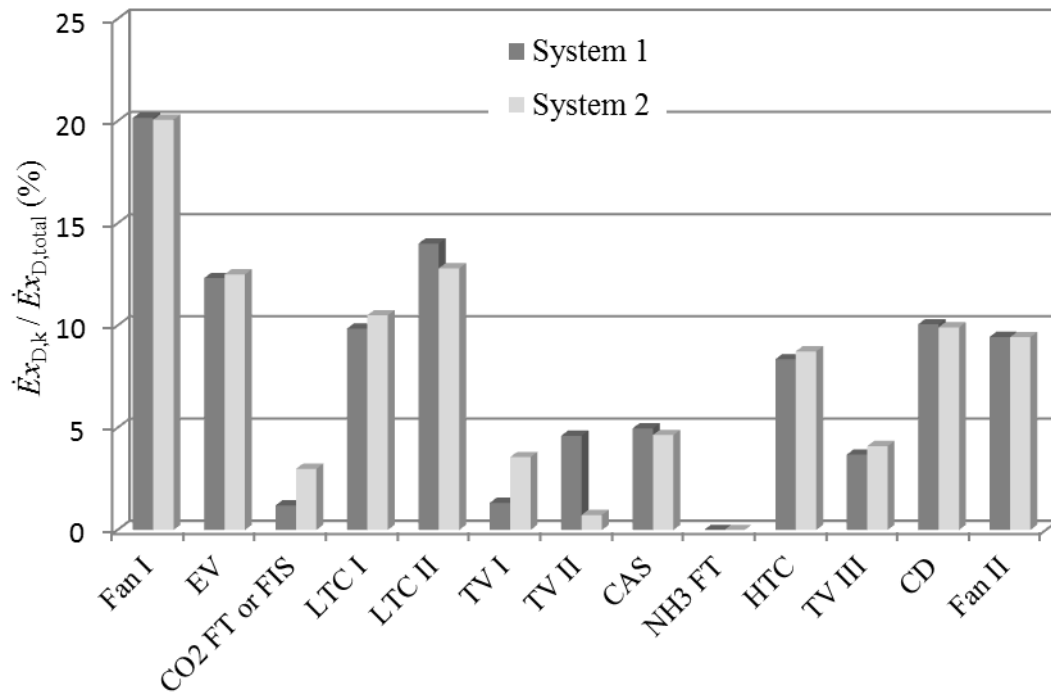


Fig. 16. Relative exergy destruction rate in components of the proposed cascade cycles operating at the thermodynamic optimal design condition.

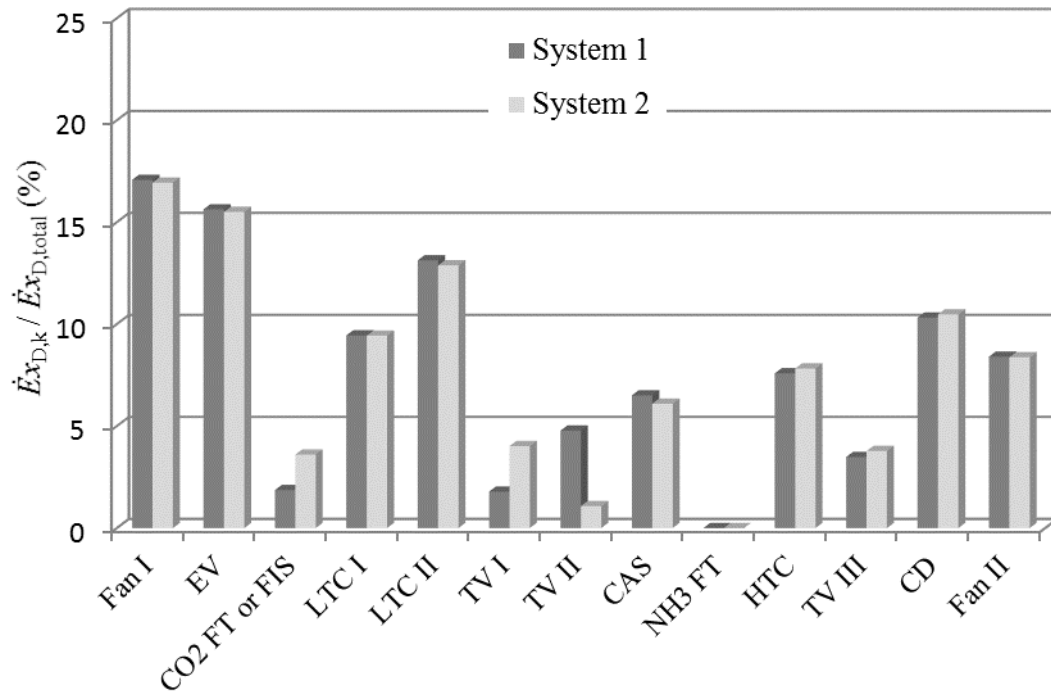


Fig. 17. Relative exergy destruction rates in components of the proposed cascade cycles operating at the cost optimal design condition.

Table 1. Fuel and product exergy rate for various components in two cycles.

Component	$\dot{E}x_F$	$\dot{E}x_P$
Fan I	$\dot{W}_{\text{Fan I}}$	$\dot{E}x_{\text{ca,i}} - \dot{E}x_{\text{ca,i}}$
Evaporator	$\dot{E}x_8 - \dot{E}x_1$	$\dot{E}x_{\text{ca,e}} - \dot{E}x_{\text{ca,i}'}$
CO ₂ flash tanks	$\dot{E}x_6 - \dot{E}x_7$	$\dot{E}x_3 - \dot{E}x_2$
Flash intercooler with indirect subcooler	$(\dot{m}_7/\dot{m}_5)\dot{E}x_5 + \dot{E}x_6 - \dot{E}x_7$	$\dot{E}x_3 - \dot{E}x_2$
Low-temperature compressor I	\dot{W}_{LTCI}	$\dot{E}x_2 - \dot{E}x_1$
Low-temperature compressor II	\dot{W}_{LTCII}	$\dot{E}x_4 - \dot{E}x_3$
Throttling valve I	$\dot{E}x_{\text{m,7}} - \dot{E}x_{\text{m,8}}$	$\dot{E}x_{\text{t,8}} - \dot{E}x_{\text{t,7}}$
Throttling valve II	$\dot{E}x_{\text{m,5}} - \dot{E}x_{\text{m,6}}$	$\dot{E}x_{\text{t,6}} - \dot{E}x_{\text{t,5}}$
Cascade heat exchanger	$\dot{E}x_{13} - (\dot{m}_{13}/\dot{m}_9)\dot{E}x_9$	$\dot{E}x_5 - \dot{E}x_4$
NH ₃ flash tank	$\dot{E}x_{12} - \dot{E}x_{13}$	$\dot{E}x_9 - (\dot{m}_{13}/\dot{m}_9)\dot{E}x_9$
High-temperature compressor	\dot{W}_{HTCI}	$\dot{E}x_{10} - \dot{E}x_9$
Throttling valve III	$\dot{E}x_{\text{m,11}} - \dot{E}x_{\text{m,12}}$	$\dot{E}x_{\text{t,12}} - \dot{E}x_{\text{t,11}}$
Condenser	$\dot{E}x_{10} - \dot{E}x_{11}$	$\dot{E}x_{\text{env,e}} - \dot{E}x_{\text{env,i}'}$
Fan II	$\dot{W}_{\text{Fan II}}$	$\dot{E}x_{\text{env,i}'}$ - $\dot{E}x_{\text{env,i}}$

Table 2. Cost functions of various components [29–32].

Component	Capital cost function (Z)
Evaporator and condenser	$1397 \times A_{\text{EV or CD}}^{0.89}$
Cascade heat exchanger	$383.5 \times A_{\text{CAS}}^{0.65}$
Low-temperature compressor	$10167.5 \times W_{\text{LTC}}^{0.46}$
High-temperature compressor	$9624.2 \times W_{\text{HTC}}^{0.46}$
Flash tank	$280.3 \times \dot{m}_i^{0.67}$
Flash intercooler with indirect subcooler	$1438.1 \times A_{\text{FI}}^{0.65}$
Throttling valve	$114.5 \times \dot{m}$
Fan	$155 \times (\dot{V} + 1.43)$
Installation of refrigeration system	$150.2 \times \dot{Q}_{\text{EV}}$

Table 3. Thermodynamic conditions considered in modelling.

Parameter	Value
Cooling capacity, \dot{Q}_{EV}	500 kW
Condensing temperature of NH ₃ , T_{CD}	35 °C
Evaporating temperature of CO ₂ , T_{EV}	-40 °C
Degree of superheating of CO ₂ at evaporator outlet, $\Delta T_{sup} (= T_1 - T_8)$	0 K
Temperature difference of air in evaporator and condenser	10 K
Condensing temperature of CO ₂ , T_5	0 °C
Cascade heat exchanger temperature difference, $\Delta T_{CAS} (= T_5 - T_{13})$	10 K
Temperature of the inlet air to the evaporator, $T_{ca,i}$	-20 °C
Ambient temperature, $T_{env,i}$	25 °C
Ambient pressure, P_0	101.3 kPa
Intermediate pressure of flash tank in CO ₂ circuit, P_{int}	2000 kPa
Overall heat transfer coefficient of evaporator, U_{EV}	30 W m ⁻² K ⁻¹
Overall heat transfer coefficient of condenser, U_{CD}	40 W m ⁻² K ⁻¹
Overall heat transfer coefficient of cascade heat exchanger, U_{CAS}	1000 W m ⁻² K ⁻¹
Overall heat transfer coefficient of flash intercooler, U_{FIS}	1000 W m ⁻² K ⁻¹

Table 4. Comparison of performance parameters obtained from present modelling for a basic CO₂/NH₃ cascade refrigeration system and the corresponding results reported elsewhere.

Parameter	Operational conditions			
	$\dot{Q}_{EV} = 40 \text{ kW}, T_{CD} = 56.3^\circ\text{C}, T_{EV} = -56^\circ\text{C},$		$\dot{Q}_{EV} = 50 \text{ kW}, T_{CD} = 40.1^\circ\text{C}, T_{EV} = -48.7^\circ\text{C},$	
	$T_s = -8.1^\circ\text{C}, \Delta T_{CAS} = 3.44^\circ\text{C}, N = 6570 \text{ h}$		$T_s = -7.1^\circ\text{C}, \Delta T_{CAS} = 2^\circ\text{C}, N = 7000 \text{ h}$	
	Present work	Ref. [18]	Present work	Ref. [29]
\dot{W}_{total} (kW)	62.96	63.01	32.57	33.44
COP	0.635	0.634	1.53	1.49
ψ (%)	19.49	19.48	47.10	45.89
\dot{C}_f (\$ year ⁻¹)	28,954	28,978	13,681	14,048
\dot{C}_{total} (\$ year ⁻¹)	110,683	109,242	-	-

Table 5. Results of thermodynamic optimization for two cycles.

Parameter	System 1	System 2
T_{EV} (°C)	-35	-35.20
T_{CD} (°C)	35	35.01
T_5 (°C)	0.01	-1.98
ΔT_{CAS} (K)	2.01	2.27
ΔT_{sup} (K)	0.10	0.45
P_{int} (kPa)	1861	1935
r	-	3.79
A_{EV} (m ²)	1686	1671
A_{CD} (m ²)	659.2	627.6
A_{CAS} (m ²)	57.32	59.86
$\sum \dot{W}_{CM}$ (kW)	265.13	267.37
$\dot{E}x_{D,total}$ (kW)	222.5	223.5
COP	1.547	1.536
ψ (%)	31.52	31.30
\dot{C}_{env} (\$ year ⁻¹)	120,150	121,007
\dot{C}_{total} (\$ year ⁻¹)	600,006	675,530

Table 6. Results of cost optimization for two cycles.

Parameter	System 1	System 2
T_{EV} (°C)	-40	-40
T_{CD} (°C)	36.2	36.67
T_5 (°C)	1.66	0.0
ΔT_{CAS} (K)	3.67	3.33
ΔT_{sup} (K)	1.67	1.67
P_{int} (kPa)	1833	1750
r	-	3.2
A_{EV} (m ²)	1148	1148
A_{CD} (m ²)	644.3	612.6
A_{CAS} (m ²)	45.61	46.81
$\sum \dot{W}_{CM}$ (kW)	304.19	307.93
$\dot{E}x_{D,total}$ (kW)	262.76	264.8
COP	1.38	1.36
ψ (%)	28.04	27.75
\dot{C}_{env} (\$ year ⁻¹)	135,082	136,494
\dot{C}_{total} (\$ year ⁻¹)	580,387	661,197

Table 7. Exergoeconomic analysis results for the thermodynamic optimal design conditions of the presented systems.

Component	System 1			System 2		
	ψ_k (%)	$c_{F,k}$ (\$ GJ ⁻¹)	f_k (%)	ψ_k (%)	$c_{F,k}$ (\$ GJ ⁻¹)	f_k (%)
Fan I	19.62	25.0	5.46	19.62	25.0	5.46
Evaporator	79.96	174.5	70.95	79.63	502.2	45.03
CO ₂ flash tank	97.48	172.6	1.73	-	-	-
Flash intercooler with liquid subcooler	-	-	-	93.29	504.2	63.70
Low-temperature compressor I	56.53	25.0	55.83	59.07	25.0	55.57
Low-temperature compressor II	65.53	25.0	53.80	63.12	25.0	54.18
Throttling valve I	92.64	172.6	30.43	91.62	495.8	5.60
Throttling valve II	85.82	168.6	14.74	87.20	495.8	7.07
Cascade heat exchanger	85.43	155.5	19.46	87.21	148.3	3.85
NH ₃ flash tank	100	155.5	100	100	148.3	100
High-temperature compressor	85.04	25.0	68.14	85.30	25.0	67.74
Throttling valve III	90.54	145.5	38.72	90.17	137.8	37.27
Condenser	3.76	114.0	66.52	3.54	107.2	67.15
Fan II	1.67	25.0	17.91	1.67	25.0	17.91

Table 8. Exergoeconomic analysis results for the cost optimal design conditions of the presented systems.

Component	System 1		System 2			
	ψ_k (%)	$c_{F,k}$ (\$ GJ ⁻¹)	f_k (%)	ψ_k (%)	$c_{F,k}$ (\$ GJ ⁻¹)	f_k (%)
Fan I	19.62	25.0	5.46	19.62	25.0	5.46
Evaporator	72.68	188.9	51.81	72.68	485.9	29.83
CO ₂ flash tank	96.22	186.1	1.89	-	-	-
Flash intercooler with liquid subcooler	-	-	-	91.50	484.3	55.82
Low-temperature compressor I	62.61	25.0	55.91	61.21	25.0	55.41
Low-temperature compressor II	66.58	25.0	52.79	66.44	25.0	52.89
Throttling valve I	91.61	186.1	20.48	90.84	472.7	4.44
Throttling valve II	84.78	181.2	12.14	85.36	472.7	5.14
Cascade heat exchanger	78.43	155.3	12.00	80.43	108.1	2.12
NH ₃ flash tank	100	155.3	100	100	150.0	100
High-temperature compressor	85.19	25.0	67.41	85.39	25.0	67.10
Throttling valve III	89.95	143.7	37.54	89.51	139.3	36.19
Condenser	3.25	110.3	61.98	2.99	108.1	61.28
Fan II	1.66	25.0	17.89	1.66	25.0	17.89