Thermoeconomic analysis of a novel zero-CO2-emission high-efficiency power cycle using LNG coldness

Meng Liu, Noam Lior, Na Zhang, Wei Han

Abstract

This paper presents a thermoeconomic analysis aimed at the optimization of a novel zero-CO2 and other emissions and high-efficiency power and refrigeration cogeneration system, COOLCEP-S (Patent pending), which uses the liquefied natural gas (LNG) coldness during its revaporization. It was predicted that at the turbine inlet temperature (TIT) of 900 °C, the energy efficiency of the COOLCEP-S system reaches 59%. The thermoeconomic analysis determines the specific cost, the cost of electricity, the system payback period and the total net revenue. The optimization started by performing a thermodynamic sensitivity analysis, which has shown that for a fixed TIT and pressure ratio, the pinch point temperature difference in the recuperator, ∆T_{psi}, and that in the condenser, ∆T_{psi}, are the most significant unconstrained variables to have a significant effect on the thermal performance of novel cycle. The payback period of this novel cycle (with fixed net power output of 20 MW and plant life of 40 years) was ~5.9 years at most, and would be reduced to ~3.1 years at most when there is a market for the refrigeration byproduct. The capital investment cost of the economically optimized plant is estimated to be about 1000 $/kWe, and the cost of electricity is estimated to be 0.34–0.37 CNY/kWh (~0.04 $/kWh). These values are much lower than those of conventional coal power plants being installed at this time in China, which, in contrast to COOLCEP-S, do produce CO2 emissions at that.

1. Introduction

Natural gas is one of the most widely used fossil energy resource with higher heat value and less pollutant production than the other fossil energy resources. Since the first liquefied natural gas (LNG) trade in 1964, the global LNG trade has seen a continuously rapid growth, mainly because the transformation from natural gas to the LNG reduces its volume by about 600-fold and thus facilitates the conveyance from the gas source to receiving terminal. Liquefaction of the gas to LNG requires, however, approximately 500 kWh electric energy per ton LNG. It is noteworthy that the LNG, at about 110 K, thus contains a considerable portion of the energy and exergy that were invested in this process. The principle of the novel COOLCEP-S system is the effective use of that stored potential during the revaporization and heating to approximately ambient temperature of the LNG for pipeline transmission to the consumers. This use of the valuable energy and exergy replaces the commonly employed revaporization methods of using ambient (ocean or air) or gas combustion heat, which simply waste it and may also cause undesirable environmental effects.

Recovery of the cryogenic exergy in the LNG evaporation process by incorporating this process into a properly designed thermal power cycle, in different ways, has been proposed in a number of past publications [1–13]. This includes methods which use the LNG as the working fluid in natural gas direct expansion cycles, or its coldness as the heat sink in closed-loop Rankine cycles [1–6], Brayton cycles [7–9], and combinations thereof [10,11]. Other methods use the LNG coldness to improve the performance of conventional thermal power cycles. For example, LNG vaporization can be integrated with gas turbine inlet air cooling [5,12] or steam turbine condenser system (by cooling the recycled water [11]), etc. Some pilot plants have been established in Japan from the 1970s, combining closed-loop Rankine cycles (with pure or mixture organic working fluids) and direct expansion cycles [1].

Increasing concern about greenhouse effects on climate change prompted a significant growth in research and practice of CO2 emission mitigation in recent years. The main technologies proposed for CO2 capture in power plants are physical and chemical absorption, cryogenic fractionation, and membrane separation. The amount of energy needed for the CO2 capture would lead to the reduction of power generation energy efficiency by up to 10 percentage points [14,15].
Beside the efforts for reduction of CO₂ emissions from existing power plants, concepts of power plants having zero-CO₂-emission were proposed and studied. Oxy-fuel combustion is one of the proposed removal strategies. It is based on the close-to-stoichiometric combustion, where the fuel is burned with enriched oxygen (produced in an air separation unit, ASU) and recycled flue gas. The combustion is accomplished in absence of the large amounts of nitrogen and produces only CO₂ and H₂O. CO₂ separation is accomplished by condensing water from the flue gas and therefore requires only a modest amount of energy. Some of the oxy-fuel cycles with ASU and recycled CO₂/H₂O from the flue gas are the Graz cycle, the Water Cycle, and the Matiant cycle [16–20]. We proposed and analyzed the semi-closed oxy-fuel cycles with integration of the LNG cold exergy utilization [21,22]. The additional power use for O₂ production amounts to 7–10% of the cycle total operation and configuration. A much more limited paper on the COOLCEP-S cycle reaches 59%.

Consider the thermoeconomic analysis, the recuperator and the CO₂ condenser are the two most important heat exchangers in the COOLCEP-S cycle in their effect on performance and cost. In this paper, we conduct a thermoeconomic optimization of the pinch point temperature differences, the ΔTₚᵢ, in the recuperator and the ΔTₚ₂, in the CO₂ condenser, based on the cycle thermal and exergy efficiencies, and the economic performance evaluation criteria that include the plant specific cost, the cost of electricity, the payback period and the total net revenue, to find the thermoeconomically optimal values of ΔTₚᵢ and ΔTₚ₂.

2. System configuration description

Fig. 1 shows the layout of the COOLCEP-S cycle, which consists of a power subcycle and an LNG vaporization process. Fig. 2 is the cycle i-s diagram. The interfaces between the power subcycle and the LNG vaporization process are the CO₂ condenser CON, the heat exchangers HEX1, and the fuel feed stream 8.

The power subcycle can be identified as 1–2–3–4–5–6–7–8–9–10–11-12/13-14-1. The low temperature (−30 °C) liquid CO₂ as the main working fluid (1) is pumped to about 30 bar (2), then goes through a heat addition process (2–3) in the evaporator EVA1 and can thereby produce refrigeration if needed. The O₂ (4) produced in an air separator unit (ASU) is compressed and mixed with the main CO₂ working fluid. The gas mixture (6) is heated (6–7) by turbine (GT) exhaust heat recuperation in REP. The working fluid temperature is further elevated in the combustor B, fueled with natural gas (8), to its maximal value (the turbine inlet temperature Tᵢ) (9). The working fluid expands to the working fluid condensation pressure (10) in the gas turbine (GT) to generate power and is then cooled (to 11) in the recuperator REP.

The gases in the mixture at the exit of REP (11) need to be separated, and the combustion-generated CO₂ component needs to be condensed for ultimate sequestration, and this is performed by further cooling: in the LNG-cooled heat exchanger HEX1, in which the H₂O vapor in the mixture is condensed and drained out (12). Afterwards, the remaining working gas (13) is condensed (14) in the condenser CON against the LNG evaporation, and recycled (1). The remaining working fluid (15) enriched with noncondensable species (mainly N₂, Ar and O₂) is further compressed in C3 at a higher pressure level under which the combustion-generated CO₂ is condensed and captured, ready for final disposal.

The LNG vaporization process is 18–19–19a/b–20a/b–20–22/23/8. LNG (18) is pumped by P2 to the highest pressure (73.5 bar), typical for receiving terminals, which supply long-distance pipeline network, and then evaporated with the heat addition from the power cycle. The evaporated NG (natural gas) may produce a small amount of cooling in HEX3 if its temperature is still low enough at the exit of HEX1, and thus contribute to the overall system useful outputs. Finally, the emerging natural gas stream is split into two parts where most of it (23) is sent to outside users and a small part (8) is used as the fuel in the combustor of this cycle.

3. Calculation assumptions and evaluation criteria

3.1. Calculation assumptions

The simulations were carried out to the COOLCEP-S cycle by using the commercial Aspen Plus software [29], in which the component models are based on the energy balance and mass balance, with the default relative convergence error tolerance of 0.0001% which is used to determine whether a tear stream is converged or not, the tear stream is one for which Aspen Plus makes an initial guess, and iteratively updates the guess until two consecutive guesses are within a specified tolerance.
The tear stream is converged when the following is true for all tear convergence variables $X$ including the total mole flow, all component mole flows, pressure, and enthalpy:

$$\frac{|X_{\text{calculated}} - X_{\text{assumed}}|}{X_{\text{assumed}}} < \text{tolerance}$$

where the default for tolerance is 0.0001, $X_{\text{assumed}}$ is the assumed value of $X$ before the calculation is conducted, $X_{\text{calculated}}$ is the calculated value of $X$.

The PSRK property method was selected for the thermal property calculations, which is based on the Predictive Soave–Redlich–Kwong equation of state model (an extension of the Redlich–Kwong–Soave equation of state). It can be used for mixtures of non-polar and polar compounds, in combination with light gases, and up to high temperatures and pressures. Some properties of feed streams are reported in Table 1, and the main assumptions for simulations are summarized in Table 2.

### Oxygen (95 mol%) from a cryogenic ASU is chosen for the combustion, since this was considered to be the optimal oxygen purity when taking into account the tradeoff between the cost of producing the higher-purity oxygen and the cost of removing noncondensable species from the CO$_2$. The O$_2$ composition and its power consumption for production follow those in [26].

For the water separation, the turbine exhaust gas is cooled in HEX1 to 0°C. Water is condensed and removed before CO$_2$ compression in C$_2$. To simplify the simulation it is assumed that water and CO$_2$ are fully separated.

### 3.2. Thermal performance evaluation criteria

The commonly used thermal power generation efficiency is defined as:

$$\eta_t = \frac{W_{\text{net}}/\text{m}_f}{LHV}$$

Since the power and refrigeration cogeneration energy efficiency definition is problematic (cf. [30], for evaluating the cogeneration we use the exergy efficiency as:

$$\theta = \frac{(W_{\text{net}} + E_c)/\text{m}_f \cdot e_f + m_{\text{LNG}} \cdot e_{\text{LNG}})}{m_f \cdot e_f + m_{\text{LNG}} \cdot e_{\text{LNG}}}$$

![Fig. 1. The process flowsheet of the COOLCEP-S system.](image)

![Fig. 2. Cycle $t$–$s$ diagram in the COOLCEP-S system.](image)

<table>
<thead>
<tr>
<th>Table 1: Molar composition and some properties for feed streams.</th>
</tr>
</thead>
<tbody>
<tr>
<td>LNG</td>
</tr>
<tr>
<td>CH$_4$ (mol%)</td>
</tr>
<tr>
<td>C$_2$H$_6$ (mol%)</td>
</tr>
<tr>
<td>C$_3$H$_8$ (mol%)</td>
</tr>
<tr>
<td>C$_4$H$_10$ (mol%)</td>
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<td>N$_2$ (mol%)</td>
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<tr>
<td>O$_2$ (mol%)</td>
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</tr>
<tr>
<td>H$_2$O (mol%)</td>
</tr>
<tr>
<td>Ar (mol%)</td>
</tr>
<tr>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>Lower heating value (kJ/kg)</td>
</tr>
<tr>
<td>Power consumption for O$_2$ production (kJ/kg)</td>
</tr>
</tbody>
</table>
To optimize system configuration and design, we adopted and used four system economic performance criteria: (I) specific cost \( C_u \), (II) cost of electricity \( COE \), (III) system payback period \( PY \), (IV) system net revenue \( \text{Net} \).

3.3.1. Specific cost \( C_u \)

The specific cost \( C_u \) is defined as the ratio between the total plant investment \( C \) and the cycle net power output \( W_{\text{net}} \).

\[
C_u = \frac{C}{W_{\text{net}}}
\]  
(4)

where the total plant investment, \( C \), is the sum of the costs of all the hardware (dynamic equipments \( C_{\text{DYN}} \), heat exchangers \( C_{\text{HEX}} \), the conventional LNG evaporators \( C_{\text{EVA}} \), and the balance of plant \( C_{\text{BOP}} \)). It should be pointed out that the conventional LNG evaporators is necessary as a backup for the LNG vaporization in case of the plant shutdown due to routine maintenance or emergency.

Balance of plant consists of the remaining systems, components, and structures that comprise a complete power plant or energy system that are not included in the prime mover [31]. As the systems are more complex than the conventional power generation system, here we assumed that the BOP accounts for 20% of the known component cost of the system.

3.3.2. Cost of electricity (COE)

The cost of electricity in the operation period is calculated as:

\[
COE = \frac{P_i + C_i + C_j - r_{CO2} - \lambda \cdot r_{ref}}{H \cdot W_{\text{net}}}
\]  
(5)

\( c_m \) is the annual cost of operating and maintenance (O&M), assumed to be 4% of the total plant investment, \( C_i \) [32]. Taxes and insurance are not considered in this preliminary evaluation. \( C_j \) is the annual fuel cost, and \( r_{CO2} \) is the annual CO2 credit defined as the product of the annual CO2 emission reduction multiplied by the CO2 tax. \( H \) is the annual operation hours. \( \beta \) is a function of interest rate and the plant operation life \( n \):

\[
\beta = \frac{i}{1 - (1 + i)^{-n}}
\]  
(6)

with \( n = 40 \) and \( i = 8\% \), \( \beta = 0.08386 \). \( \lambda \cdot r_{ref} \) is the actual annual refrigeration revenue because the refrigeration production may not be all sold out, so we adopt a refrigeration revenue factor \( \lambda \) with a range of 0–1 (0 for the case of no cooling requirement from users, 1 when all the refrigeration is sold) to indicate how much we can benefit from the refrigeration. The refrigeration price is assumed to be the same as the price of the electric power that would have been needed to supply the same cooling capacity \( Q_c \) by using state of the art vapor compression refrigeration machinery. The needed electricity, \( W_{\text{comp}} \) is thus calculated by

\[
W_{\text{comp}} = Q_c / \text{COP}
\]  
(7)

where the \( \text{COP} \) (coefficient of performance) is assumed to be 7, a normal value for the compression refrigeration which normally can provide refrigeration with the temperature of 5°C. It should be pointed out that, in practice, the refrigeration price is influenced by many other non-technical factors such as the market demands, climate change and artificial interference.

3.3.3. System payback period (PY)

The net current value, \( P \), within \( n \) years is calculated as [33]:

\[
P = B \cdot \frac{(1 + i)^n - 1}{i(1 + i)^n}
\]  
(8)

where \( n = 1, 2, \ldots , 40 \), and \( i = 8\% \) is the annual value

\[
B = r_{CO2} + r_j + \lambda \cdot r_{ref} - C_j - C_m
\]  
(9)

\( r_e \) is the annual electricity power revenue defined as the product of the annual electricity output multiplied by the electricity price.
when the cash flow $P$ is equal to the total plant investment $C_i$. The related value of $n$ is the payback period $P_Y$.

3.3.4. Total net revenue ($R_{net}$)

The system total net revenue, $R_{net}$, within the plant life $L_p$ of 40 years is the sum of the total gross revenue $R_g$ minus the total plant investment $C_i$.

$$R_{net} = R_g - C_i = L_p \cdot r_{net} - C_i$$

(10)

where $r_{net}$ is the system annual net revenue,

$$r_{net} = \frac{C_0 + \lambda \cdot r_{net} - C_f - C_m - \beta \cdot C_i}{C_0}$$

(11)

Table 3 presents the equipment and product information from the manufactures [34–37] and the product price in China, except the turbine price.

The price of the gas turbine, $Y_{GT}$, is calculated based on a costing correlation we developed (from data in the Gas Turbine World Handbook 2005–2006) for mechanical drive gas turbines,

$$Y = 76.07 - 63.735 \times 10^3 + 6.21 \times 10^6 + 2.514 \times 10^8$$

$$+ 4.669 \times 10^9 - 3.271 \times 10^{10} - 3.76 \times 10^7 + 1343.02\eta_e$$

$$- 1635.07\eta_e^2$$

(12)

where $Y$ ($$/kW$$) is the cost; $W_{GT}$ (MW) is the turbine power output; and $\eta_e$ is the thermal efficiency.

It should be pointed out that Eq. (12) is intended for the price calculation of conventional simple gas turbines. To account for the fact that the turbine in our system uses CO2 as the working fluid, which may require some modifications of conventional turbines, we multiplied by 1.5 the price calculated by Eq. (12), which is based on the price of the model UGT-15000 + 20 MW turbine\(^1\) of 289 $$$/kW$$ (excluding compressor). As a result, the turbine price, $Y_{GT}$, is the product of the price $Y$ obtained from Eq. (12) multiplied by the modification factor $\gamma$. Finally, $Y_{GT}$ is of the form,

$$Y_{GT} = \gamma \cdot Y$$

(13)

4. Sensitivity analysis of the pinch point temperature differences in the major heat exchangers

In the COOLCEP-S cycle, lowering the temperature difference $\Delta T_{p1}$ in the heat transfer processes is helpful to improve the cycle thermal performance, but at the same time requires larger and thus more expensive heat exchangers. Hence, a sensitivity analysis was carried here out of the COOLCEP-S cycle to study the effect of the pinch point temperature differences ($\Delta T_{p1}$ in the recuperator REP and $\Delta T_{p2}$ in the CO2 condenser CON) on the cycle thermal performance and the economic performance.

In the following calculation, the assumptions are kept unchanged as shown in Table 2, except the pinch point temperature differences, 45 K of $\Delta T_{p1}$ in recuperator and 8 K of $\Delta T_{p2}$ in CO2 condenser.

4.1. Effect of the pinch point temperature difference $\Delta T_{p1}$ in the recuperator REP

4.1.1. Effect of $\Delta T_{p1}$ on the cycle thermal performance

With the same net power output $W_{net}$ of 20 MW, simulation computation is made of the basic cycle for values of $\Delta T_{p1}$ from 45 K to 90 K. The heat exchanger transfer area estimation and the cycle thermal performance are shown in Tables 4 and 5. It should be pointed out that the heat transfer area estimation here is rough and based on the assumption that the heat exchangers are of the shell-and-tube type, and using average typical overall heat transfer coefficient values for these heat exchangers and fluids as found in the process heat transfer literature [38]. The recuperator REP is a conventional gas-to-gas heat exchanger; the heat exchanger HEX1 is also a gas-to-gas exchanger, HEX2 is a heat exchanger with phase change of gas-to-liquid in the hot side (16–17), and of LNG (containing some noncondensable gas) vaporizing in the cold side (19b–20b). The condenser CON consists of two parts, in the first part cooling the CO2 gas by the colder natural gas, followed by the second part in which CO2 is then condensed due to cooling by liquid, boiling and gaseous natural gas, with an overall heat transfer coefficient of 600 W/m2 K; the hot stream in EVA1 and HEX3 is assumed to be water with the inlet and outlet temperatures of 25 and 20 °C, respectively.

As shown in Table 4, as $\Delta T_{p1}$ is increased from 45 K to 90 K, the heat duty $Q$ of the REP keeps decreasing and thus the heat transfer area $A$ of the REP decrease too although they are always higher than those in the other heat exchangers. Fig. 3 explains the reason of the heat duty change in the REP: the hot side inlet temperature $t_{10}$ is maintained fixed because of the fixed turbine inlet temperature $t_0$ and pressure ratio $p_{10}/p_{10}$, and the cold side inlet temperature $t_6$. The $\Delta T_{p1}$ always appears on the hot end of REP irrespective of its changes in this analysis. Hence, looking at Fig. 3, the $t_7$ decreases and the $t_6$ increases. As a result, the $\Delta T_{p1}$ temperature changes lead to the heat duty changes in the REP despite the mass flow rate increase of the working fluid in it.

Table 4 indicates the heat transfer area $A$ in REP is reduced by 52% (or, by 8047 m\(^2\)) as the $\Delta T_{p1}$ is increased from 45 K to 90 K; while the total heat transfer area of all heat exchangers in the system, $\Sigma A$, is reduced by 30% (7483 m\(^2\)), which is less than the decrease for REP alone because the increase of $\Delta T_{p1}$ causes some increase in the LNG evaporation unit area.

Table 5 indicates that the increase of $\Delta T_{p1}$ causes the increase of work input/output of the dynamic equipment (pumps P1 and P2, compressors C1 and C3, gas turbine GT). This is because the increase of $\Delta T_{p1}$ leads to the decrease of the inlet temperature of combustor $B (T_i)$ as shown in Fig. 3, causing more fuel input to the combustor, and thus the working fluid flow rate going through

\footnote{Zarya–Mashproekt State Enterprise Gas Turbine Research & Production Complex, Ukraine.}

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Price</th>
<th>Equipment</th>
<th>Price</th>
<th>Price ($10^3 $/m(^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>ASU</td>
<td>1376 $/kg O_2$</td>
<td>Recuperator</td>
<td>0.244</td>
<td></td>
</tr>
<tr>
<td>O₂ compressor C1</td>
<td>164.5 $/kg$</td>
<td>Condenser</td>
<td>0.097</td>
<td></td>
</tr>
<tr>
<td>Compressor C3</td>
<td>164.5 $/kg$</td>
<td>Heat exchangers</td>
<td>0.097</td>
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</tr>
<tr>
<td>LNG pump P2</td>
<td>3.44 $/kg$</td>
<td>Evaporator EVA1</td>
<td>0.097</td>
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<tr>
<td>CO₂ pump P1</td>
<td>3.2 $/kg$</td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td>Product</td>
<td>Price ($/kW h$)</td>
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<td>0.033 $/ton CO₂$</td>
<td>Refrigeration</td>
<td>0.059</td>
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</tr>
</tbody>
</table>

Table 3

Equipment and product cost information.
the dynamic equipments increases (while the pressure changes across them remain the same).

We then examine the refrigeration output from the evaporators EVA1 where the low temperature liquid CO2 is evaporated, and from HEX3 where the low temperature NG is heated to the near environment temperature. As the ΔT_p1 is increased, Table 5 shows that the cooling capacity and refrigeration exergy in the evaporator EVA1 increase, and that is entirely because of the associated increase in the liquid CO2 mass flow rate, since the refrigeration temperature range is maintained fixed. Things are different in the HEX3: as the ΔT_p1 increases, both the HEX3 inlet temperature t21 and the LNG flow rate increase, and these two factors have opposite effects on the refrigeration output. The calculation results indicate that the negative effect of the former one dominates so overall that the refrigeration production in HEX3 decreases. It can be seen from Table 5 that the reduction of refrigeration output in HEX3 surpasses the increment in EVA1, so there are reductions of 10.7% (6.1 MW) in the total cooling

<table>
<thead>
<tr>
<th>ΔT_p1 (K)</th>
<th>Recuperator</th>
<th>LNG evaporation unit</th>
<th>Evaporation unit</th>
<th>Evaporation unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>45 (Basic case)</td>
<td>REP</td>
<td>CON</td>
<td>EVA1</td>
<td>EVA1</td>
</tr>
<tr>
<td>45</td>
<td>74.17</td>
<td>44.81</td>
<td>42.42</td>
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<td>60</td>
<td>72.77</td>
<td>45.14</td>
<td>42.68</td>
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<td>75</td>
<td>71.37</td>
<td>45.49</td>
<td>42.95</td>
<td>42.95</td>
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<tr>
<td>90</td>
<td>69.95</td>
<td>45.83</td>
<td>43.23</td>
<td>43.23</td>
</tr>
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</table>

Table 4

Estimation of heat transfer areas, A, for different ΔT_p1.

<table>
<thead>
<tr>
<th>ΔT_p1 (K)</th>
<th>Unit</th>
<th>Q (MW)</th>
<th>LMTD (K)</th>
<th>U (W/m² K)</th>
<th>A (m²)</th>
<th>A (%)</th>
<th>ΣA (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>45 (Basic case)</td>
<td>REP</td>
<td>74.17</td>
<td>51.5</td>
<td>93</td>
<td>15,487</td>
<td>62.2</td>
<td>24,892</td>
</tr>
<tr>
<td>LNG evaporation unit</td>
<td>CON</td>
<td>44.81</td>
<td>29.6</td>
<td>99/600</td>
<td>4159</td>
<td>16.7</td>
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<tr>
<td>HEX1</td>
<td>9.75</td>
<td>72.2</td>
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<td>HEX2</td>
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<td>55.9</td>
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<tr>
<td>HEX3</td>
<td>14.51</td>
<td>33.8</td>
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<td>60</td>
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<td>19.9</td>
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<tr>
<td>HEX1</td>
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<td>HEX2</td>
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<td>Evaporation unit</td>
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<td>429</td>
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<tr>
<td>HEX3</td>
<td>12.27</td>
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<td>Republic</td>
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Table 5

Cycle thermal performance for different ΔT_p1.

<table>
<thead>
<tr>
<th>ΔT_p1 (K)</th>
<th>Net power output, W_{net} (MW), kept constant</th>
<th>Heat duty (MW)</th>
<th>Work (MW)</th>
<th>Refrigeration</th>
<th>Mass flow rate (kg/s)</th>
<th>Total cooling capacity, Q_C (MW)</th>
<th>Total exergy, E_C (MW)</th>
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<td>45 (Basic cycle)</td>
<td>20</td>
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<td>20</td>
<td>20</td>
<td>20</td>
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<tr>
<td>60</td>
<td>74.17</td>
<td>72.77</td>
<td>71.37</td>
<td>69.95</td>
<td>20</td>
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<tr>
<td>75</td>
<td>45.49</td>
<td>45.14</td>
<td>45.49</td>
<td>45.83</td>
<td>20</td>
<td>20</td>
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<tr>
<td>90</td>
<td>45.83</td>
<td>45.14</td>
<td>45.49</td>
<td>45.83</td>
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<tr>
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<td>16.9</td>
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<td>600</td>
<td>25</td>
<td>0.1</td>
<td>1.08</td>
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<td>1.93</td>
<td>1.942</td>
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<td>1.918</td>
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<tr>
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<td>1.906</td>
<td>1.918</td>
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<td>75</td>
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<td>1.942</td>
<td>1.906</td>
<td>1.918</td>
<td>1.93</td>
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<tr>
<td>90</td>
<td>1.906</td>
<td>1.918</td>
<td>1.93</td>
<td>1.942</td>
<td>1.906</td>
<td>1.918</td>
<td>1.93</td>
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</table>

a Work loss associated with the mechanical efficiency and generator electrical efficiency.

capacity \(Q_c\) and of 14.6% (1.3 MW) in the total refrigeration exergy output \(E_C\).

Fig. 4 shows that the increase of \(\Delta T_{p1}\) is unfavorable for the cycle efficiencies. As \(\Delta T_{p1}\) is increased from 45 K to 90 K, the thermal efficiency \(\eta_e\) declines from 59.1% to 48.9%, a reduction of 17.3%; and the exergy efficiency \(\theta\) declines from 39.8% to 34.2%, a reduction of 14.1%.

4.1.2. Effect of \(\Delta T_{p1}\) on the cycle economic performance

Based on the cycle thermal performance results shown in Table 5, an economic analysis of the effect of \(\Delta T_{p1}\) on cycle economic performance, including the system specific cost \(C_w\), cost of electricity \(COE\), payback period \(PY\), total net revenue \(R_{net}\), etc., is performed and the results are summarized in Table 6 with the assumption of 40 years of plant life \(T_p\), 7000 of annual operation hours \(H\), and 20 MW of net cycle power output \(W_{net}\).

The results of the economic analysis are shown in Table 6 and Figs. 5–9, and the following conclusions are drawn:

Increasing \(\Delta T_{p1}\) from 45 K to 90 K indeed results in a reduction of 40.7% (1,907,000 $) in the cost of heat exchangers \(C_{HEX}\) mainly due to the decrease in the heat transfer areas, but also results in a counterproductive increase of 22.3% (2,865,000 $) in the cost \(C_{DYN}\) of the dynamic equipment among which the increase of gas turbine cost is caused by the increase of \(W_{ET}\) and the decrease of \(\eta_e\) (see Eqs. (12) and (13)) and the increase of the other equipment’s cost is caused by the increase of working fluid flow rate. Overall, that increase of \(\Delta T_{p1}\) causes a 5.4% increase (1,159,000 $) in the total plant investment \(C_i\), and thus the related O&M cost increases by 47,000 $/year. Increasing \(\Delta T_{p1}\) also increases the annual fuel cost by 21% (888,000 $/year).

The system revenue is composed of three parts: (i) the CO\(_2\) credit \(r_{CO2}\), that is the revenue due to the reduction of CO\(_2\) emission; one of the most important characteristics of this cycle is zero-CO\(_2\)-emission which enables the power plant to benefit from CO\(_2\) emission allowance trading. Since more fuel is consumed as \(\Delta T_{p1}\) increases from 45 K to 90 K, more CO\(_2\) is produced and retrieved, therefore, the related revenue also increases by 20.7%, 318,000 $/year; (ii) the electricity revenue \(r_e\) remains unchanged because the net power output is assumed to be fixed as 20 MW for all values of \(\Delta T_{p1}\), but here we prefer to use the net electricity revenue \(r_{net}\), which is defined as the electricity revenue \(r_e\) reduced by the fuel cost \(c_f\), which in total shows a reduction of 22%, 888,000 $/year totally due to the increment of the fuel cost; (iii) the actual refrigeration revenue \(r_{ref}\) depends on the refrigeration market availability extent expressed by \(\lambda\) that can assume any value between 0 and 1. Table 6 shows that the upper limit \((\lambda = 1)\) of refrigeration revenue \(r_{ref}\) is reduced by 10.7%, 358,000 $/year.

Based on the above analysis of cycle cost and revenue, the increase of \(\Delta T_{p1}\) from 45 K to 90 K affects the specific cost \(C_w\), total net revenue \(R_{net}\), cost of electricity \(COE\) and payback period \(PY\) as follows:

1. Specific cost \(C_w\) increases. According to Eq. (4) and Table 6, the 5.4% increase (1,159,000 $) in the total plant investment \(C_i\) leads to a 5.4% increase of \(C_w\) from 1075 $/kW to 1133 $/kW.

Since the actual refrigeration revenue \(\lambda \cdot r_{ref}\) varies with the value of the refrigeration revenue factor \(\lambda\), we consider the total net revenue \(R_{net}\), the cost of electricity \(COE\) and the payback period \(PY\), respectively, as a function of the refrigeration revenue factor \(\lambda\) as well as of the pinch point temperature difference \(\Delta T_{p1}\). So the following analysis will discuss not only the effects of \(\Delta T_{p1}\), but also the effects in two extreme cases of \(\lambda = 0\) and \(\lambda = 1\).

2. Cost of electricity \(COE\) increases. According to Eqs. (5)–(7) and Table 6, as \(\Delta T_{p1}\) increases from 45 K to 90 K: (i) for \(\lambda = 0\), the resulting increase of the sum of all the cost \((\beta \cdot C_i + c_f + c_m + c_j)\) surpasses the increase of the CO\(_2\) credit \(r_{CO2}\), and the cost of electricity \(COE\) thus increases by 13.4%; (ii) for \(\lambda = 1\), the refrigeration revenue \(r_{ref}\) decreases with the increase of \(\Delta T_{p1}\), causing that the reduction of \((r_{CO2} + r_{ref})\) surpasses the increase of \((\beta \cdot C_i + c_f + c_m + c_j)\), and the cost of electricity \(COE\) thus increases by 53.1%.

3. System payback period \(PY\) is prolonged. Table 6 indicates that the annual value \(B\) decreases and the total plant investment \(C_i\) increases, and therefore (Eq. (8)): (i) for \(\lambda = 0\), the system payback period \(PY\) increases from 5.91 years to 7.61 years and (ii) for \(\lambda = 1\), the \(PY\) increases from 3.12 years to 3.84 years.

4. Total net revenue \(R_{net}\) decreases. According to Eq. (10) for \(R_{net}\), the reduction of the net annual revenue \(r_{net}\) and increase of the total plant investment \(C_i\) in a reduction of 31.4% (29,719,000 $/40 years) for \(\lambda = 0\) and 19.2% (44,039,000 $/40 years) for \(\lambda = 1\).
It is thus concluded that the increase of the $D_Tp_1$ from 45 K to 90 K has negative effects on the cycle economic performance and makes it obvious that the optimal design in the considered range of parameters is at the lowest practical $D_Tp_1 = 45$ K originally assumed in the system development.

It was also found that, for the same $D_Tp_1$, the system has a much better economic performance for $k = 1$ than for $k = 0$: the total net revenue $R_{net}$ is 142% higher, COE is 50% lower, and $PY$ is shortened by at least 2.8 years.

### 4.2. Effect of the pinch point temperature difference $D_Tp_2$ in the CO2 condenser CON

Among the needed heat exchangers, second to the size of the recuperator REP is the CO2 condenser CON. In the following section, a sensitivity analysis is made of the thermal and economic

---

**Table 6**

<table>
<thead>
<tr>
<th>$D_Tp_1$ (K)</th>
<th>45 (Base case)</th>
<th>60</th>
<th>75</th>
<th>90</th>
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<tr>
<td>Operation time of plant (h year$^{-1}$)</td>
<td>7000</td>
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<td>Cost of dynamic equipments $C_{dyn}$ (10$^3$ $$$)</td>
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<td>4226 4497 4775</td>
<td>1317.6 1376</td>
<td>14337 15703</td>
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<tr>
<td>ASU</td>
<td>3956 4226 4497 4775</td>
<td>150 169 189</td>
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<td></td>
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<tr>
<td>C1, C3, P1, P2</td>
<td>1201.8 1259.2 1317.6 1376</td>
<td>367 358</td>
<td></td>
<td></td>
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<tr>
<td>Total ($10^3$ $$$)</td>
<td>12,838 14,002 14,937 15,703</td>
<td>3163 2783</td>
<td></td>
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<tr>
<td>Cost of heat exchangers, $C_{heex}$ (10$^3$ $$$)</td>
<td>3778 2799 2211 1815</td>
<td>376 367 358</td>
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<tr>
<td>CON</td>
<td>403 408 416 421</td>
<td>110 111 113</td>
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<td></td>
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<tr>
<td>HEX1,2</td>
<td>126 150 169 189</td>
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<td>HEX</td>
<td>383 376 367 358</td>
<td>3163 2783</td>
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<td>Total ($10^3$ $$$)</td>
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<td>Cost of conventional LNG evaporators, $C_{eva}$ (10$^3$ $$$)</td>
<td>395 398 400 403</td>
<td>407 388 3907</td>
<td>907</td>
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</tr>
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<td>BOP, $CO2$ (20%) (10$^3$ $$$)</td>
<td>3585 3627 3700 3778</td>
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<tr>
<td>Total plant investment, $C_i$ (10$^3$ $$$)</td>
<td>21,508 21,760 22,200 22,667</td>
<td>22,200 22,667</td>
<td>22,667</td>
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<tr>
<td>Cost of O&amp;M, $C_m$ (10$^3$ $$$/year)</td>
<td>1075 1088 1110 1133</td>
<td>1110 1133</td>
<td></td>
<td></td>
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<tr>
<td>Fuel cost, $C_f$ (10$^3$ $$$/year)</td>
<td>860 870 888 907</td>
<td>870 8907</td>
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**Table 6**: Economic estimation for different $D_Tp_1$.

* The net electricity revenue, $r_{net}$, is defined as the sum of the electricity revenue $r_e$ minus the fuel cost $c_f$. 

---

**Fig. 5.** Effect of $D_Tp_1$ on cycle costs.

**Fig. 6.** Effect of $D_Tp_1$ on cycle revenues.
the pinch point temperature difference $\Delta T_{p2}$ in CON. $\Delta T_{p1}$ in REP is fixed at its near-optimal value of 45 K, and other main assumptions, including the turbine inlet/outlet parameters and the CO$_2$ condensation pressure/temperature are maintained unchanged as in the basic cycle where the values were fixed at $\Delta T_{p1} = 45$ K, $\Delta T_{p2} = 8$ K.

4.2.1. Effect of $\Delta T_{p3}$ on the cycle thermal performance

With the same net power output $W_{net}$ of 20 MW, simulation calculations are made to the basic cycle as the $\Delta T_{p2}$ is varied from 8 K (the practical minimum, used in the basic cycle) to 17 K. Tables 7 and 8 show the heat exchanger transfer area estimation and the cycle thermal performance under different $\Delta T_{p2}$, respectively.

Fig. 10 is the $t$–$Q$ diagram of CON, it can be seen that the heat duty of CON rises by 1.8% (0.82 MW) as the $\Delta T_{p2}$ is increased, because their LMTD-s remain unchanged while their heat duties increase. For the heat exchangers CON, HEX1,2 and HEX3, their heat transfer areas decrease as $\Delta T_{p2}$ increases because their LMTD-s and heat duties increase but the LMTD-s increase more than the heat duties. The heat transfer area decrease in the LNG evaporation unit dominates over the area increase in the power subcycle, with a subsequent overall reduction of 2% (~500 m$^2$) in the total area $\Sigma A$ as $\Delta T_{p2}$ is increased from 8 K to 17 K.

Fig. 11 illustrates the effect of $\Delta T_{p3}$ on the thermal efficiency $\eta_e$ and exergy efficiency $\theta$. As the $\Delta T_{p3}$ increases from 8 K to 17 K, the working fluid mass flow rate increases and the cycle specific power decreases. As a result, the net power output remains the same (20 MW), at the same time 1.5% (0.6 MW) more fuel energy input is required in the combustor $B$, and therefore the $\eta_e$ drops by 1.8%. Also, more LNG flows through the cycle and thus 22% (8.2 MW) more LNG exergy is consumed and 43% (3.9 MW) more refrigeration exergy is produced, so the sum of exergy outputs is increased by 13.4% and the sum of exergy inputs is increased by 12%. As a result, the exergy efficiency $\theta$ has a 1.2% increase.

4.2.2. Effect of $\Delta T_{p3}$ on the cycle economic performance

Based on the simulation results shown in Tables 7 and 8, an analysis of the economic effect of $\Delta T_{p3}$ was performed and the cycle economic performance for different $\Delta T_{p3}$ is summarized in Table 9. The main assumptions are plant life $L_p = 40$ years, annual operation $H = 7000$ h, and net cycle power output $W_{net} = 20$ MW.

Table 9 and Figs. 12–16 indicate the economic effects of increasing $\Delta T_{p3}$ from 8 K to 17 K, as follows:

As shown in Fig. 12 and Table 9, there is little effect on the cost of heat exchangers $C_{HEX}$, but it results in an increase of 3.7% (476,000 $) in the cost $C_{DYN}$ of the dynamic equipment because of the increase of the working fluid flow rate. Overall, that increase of $\Delta T_{p3}$ causes a 3.1% increase (667,000 $) in the total plant investment $C_i$ and the related O&M cost increases by 27,000 $/year. Increasing $\Delta T_{p3}$ also causes the annual fuel cost to increase by 1.9% (79,000 $/year).
Fig. 13 and Table 9 show the cycle revenue, which is composed of three parts: (i) the CO2 credit \( r_{CO_2} \); since more fuel is consumed as \( \Delta \tau_{tp2} \) increases, more CO2 is produced and retrieved, therefore, the related revenue also increases by 1.8%, or 28,000 $/year; (ii) the electricity revenue \( r_e \) remains unchanged because of the fixed net power output, but the net electricity revenue \( r_e \) drops by 2%, or 79,000 $/year, entirely due to the increase of the fuel cost. Apparently, the reduction in the electric power revenue is much higher than the revenue increase due to zero-CO2 emission; and (iii) the refrigeration revenue \( r_{ref} \). The upper limit (\( \lambda = 1 \)) of refrigeration revenue \( r_{ref} \) increases by 26.3%, 883,000 $/year, mainly because an increase of 98%, 14.2 MW, in the refrigeration cooling capacity in the HEX3 caused by the increase of the LNG mass flow rate from 95.5 kg/s to 116.2 kg/s and the drop of the inlet temperature \( t_{21} \) from –35 °C to –50 °C.

Consequently, the effects of increasing \( \Delta \tau_{tp2} \) from 8 K to 17 K on the specific cost \( c_o \), the total net revenue \( R_{net} \), the cost of electricity COE and the payback period \( PY \) is as follows:

---

**Table 7**

<table>
<thead>
<tr>
<th>( \Delta \tau_{tp2} ) (K)</th>
<th>Unit</th>
<th>( Q ) (MW)</th>
<th>LMTD (K)</th>
<th>( U ) (W/m² K)</th>
<th>( A ) (m²)</th>
<th>( A ) (%)</th>
<th>( R_{net} ) (m²)</th>
<th>( Q ) (MW)</th>
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<tr>
<td>8 (Base case)</td>
<td>Power subcycle</td>
<td>REP</td>
<td>74.17</td>
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<tr>
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<td>51.5</td>
<td>93</td>
<td>15,572</td>
<td>63.4</td>
<td>24,555</td>
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<td>36.9</td>
<td>429</td>
<td>1197</td>
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<tr>
<td>14</td>
<td>Power subcycle</td>
<td>REP</td>
<td>75.04</td>
<td>51.5</td>
<td>93</td>
<td>15,668</td>
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<td>LNG evaporation unit</td>
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<td>23.76</td>
<td>39.6</td>
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<td>17</td>
<td>Power subcycle</td>
<td>REP</td>
<td>75.51</td>
<td>51.5</td>
<td>93</td>
<td>15,765</td>
<td>64.6</td>
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<td></td>
<td>LNG evaporation unit</td>
<td>CON</td>
<td>45.63</td>
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<td>99/600</td>
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<td>5636</td>
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**Table 8**

<table>
<thead>
<tr>
<th>( \Delta \tau_{tp2} ) (K)</th>
<th>8 (Base case)</th>
<th>11</th>
<th>14</th>
<th>17</th>
</tr>
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<tbody>
<tr>
<td>Net power output, ( W_{net} ) (MW)</td>
<td>20</td>
<td>20</td>
<td>20</td>
<td>20</td>
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<tr>
<td>Heat duty (MW)</td>
<td></td>
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<tr>
<td>REP</td>
<td>74.17</td>
<td>74.58</td>
<td>75.04</td>
<td>75.51</td>
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<td>CON</td>
<td>44.81</td>
<td>45.06</td>
<td>45.34</td>
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<td>HEX1,2</td>
<td>10.65</td>
<td>10.7</td>
<td>10.77</td>
<td>10.83</td>
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<tr>
<td>Work (MW)</td>
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<tr>
<td>( W_{ex} )</td>
<td>0.831</td>
<td>0.833</td>
<td>0.834</td>
<td>0.835</td>
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<tr>
<td>( W_{ASU} )</td>
<td>2.338</td>
<td>2.347</td>
<td>2.362</td>
<td>2.376</td>
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<tr>
<td>( P_1 )</td>
<td>0.269</td>
<td>0.27</td>
<td>0.272</td>
<td>0.273</td>
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<tr>
<td>( P_2 )</td>
<td>1.906</td>
<td>2.035</td>
<td>2.176</td>
<td>2.318</td>
</tr>
<tr>
<td>( C_1 )</td>
<td>0.924</td>
<td>0.928</td>
<td>0.933</td>
<td>0.939</td>
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<tr>
<td>( C_3 )</td>
<td>0.264</td>
<td>0.265</td>
<td>0.267</td>
<td>0.269</td>
</tr>
<tr>
<td>( C_7 )</td>
<td>26.533</td>
<td>26.678</td>
<td>26.843</td>
<td>27.01</td>
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<tr>
<td>Refrigeration</td>
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<tr>
<td>EVA1</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Temperature range (°C)</td>
<td>–49.4 to +8</td>
<td>–49.4 to +8</td>
<td>–49.4 to +8</td>
<td>–49.4 to +8</td>
</tr>
<tr>
<td>Cooling capacity (MW)</td>
<td>42.42</td>
<td>42.66</td>
<td>42.92</td>
<td>43.19</td>
</tr>
<tr>
<td>Exergy (MW)</td>
<td>6.57</td>
<td>6.61</td>
<td>6.65</td>
<td>6.69</td>
</tr>
<tr>
<td>HEX3</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature range (°C)</td>
<td>–34.7 to +8</td>
<td>–40.8 to +8</td>
<td>–45.8 to +8</td>
<td>–49.8 to +8</td>
</tr>
<tr>
<td>Cooling capacity (MW)</td>
<td>14.51</td>
<td>18.94</td>
<td>23.76</td>
<td>28.7</td>
</tr>
<tr>
<td>Exergy (MW)</td>
<td>2.39</td>
<td>3.47</td>
<td>4.76</td>
<td>6.15</td>
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<td>Total cooling capacity ( Q_c ) (MW)</td>
<td>56.94</td>
<td>61.6</td>
<td>66.68</td>
<td>71.89</td>
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<td>Total exergy ( E_c ) (MW)</td>
<td>8.96</td>
<td>10.08</td>
<td>11.41</td>
<td>12.84</td>
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<tr>
<td>Mass flow rate (kg/s)</td>
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<td></td>
<td></td>
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<tr>
<td>Main working fluid, ( m_{wf} )</td>
<td>101.61</td>
<td>102.17</td>
<td>103.43</td>
<td>103.43</td>
</tr>
<tr>
<td>Retrieved liquid CO2, ( m_{ret,rec} )</td>
<td>1.846</td>
<td>1.855</td>
<td>1.867</td>
<td>1.879</td>
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<tr>
<td>NG fuel, ( m_{fuel} )</td>
<td>0.688</td>
<td>0.692</td>
<td>0.696</td>
<td>0.701</td>
</tr>
<tr>
<td>LNG, ( m_{LNG} )</td>
<td>95.54</td>
<td>102</td>
<td>109.07</td>
<td>116.2</td>
</tr>
<tr>
<td>Thermal efficiency, ( \eta_{t} ) (%)</td>
<td>59.06</td>
<td>58.73</td>
<td>58.37</td>
<td>58.01</td>
</tr>
<tr>
<td>Exergy efficiency, ( \eta_{e} ) (%)</td>
<td>39.79</td>
<td>39.82</td>
<td>39.99</td>
<td>40.25</td>
</tr>
</tbody>
</table>
(1) The specific cost $C_w$ increases: According to Eq. (4) and Table 9, the 3.1% increase (667,000 $) in the total plant investment $C_i$ results in a 3.2% increase of $C_w$, from 1075 $/kW$ to 1109 $/kW$.

Again, the effects are considered for the limiting cases $k=0$ and $k=1$ in the following analysis.

(2) Cost of electricity COE: According to the Eqs. (5)–(7) and Table 9, as $\Delta T_{p2}$ is increased from 8 K to 17 K: (i) for $k=0$, the resulting increase of the sum of all the cost ($\beta \cdot C_i + C_m + C_f$) surpasses the increase of the CO2 credit $r_{CO2}$, and the cost of electricity COE thus increases by 2.6%

---

Table 9

<table>
<thead>
<tr>
<th>$\Delta T_{p2}$ (K)</th>
<th>8 (Base case)</th>
<th>11</th>
<th>14</th>
<th>17</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operation time of plant (h/year)</td>
<td>7000</td>
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<tr>
<td>Cost of dynamic equipments, $C_{dyn}$ (10^3 $)</td>
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<td>7777</td>
<td>7883</td>
<td>7998</td>
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<tr>
<td>ASU</td>
<td>3956</td>
<td>3978</td>
<td>4003</td>
<td>4028</td>
</tr>
<tr>
<td>C1, C3, P1, P2</td>
<td>1201.8</td>
<td>1229</td>
<td>1258.5</td>
<td>1287.8</td>
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<tr>
<td>Total (10^3 $)</td>
<td>12,838</td>
<td>12,984</td>
<td>13,145</td>
<td>13,314</td>
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<tr>
<td>Cost of heat exchangers, $C_{HEX}$ (10^3 $)</td>
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<td>3800</td>
<td>3823</td>
<td>3847</td>
</tr>
<tr>
<td>REP</td>
<td>285</td>
<td>287</td>
<td>289</td>
<td>291</td>
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<tr>
<td>EVA1</td>
<td>403</td>
<td>348</td>
<td>308</td>
<td>279</td>
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<td>CON</td>
<td>126</td>
<td>120</td>
<td>116</td>
<td>113</td>
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<td>HEX1,2</td>
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<td>HEX3</td>
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<td>4672</td>
<td>4685</td>
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<td>Total (10^3 $)</td>
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<td>21,692</td>
<td>21,922</td>
<td>22,175</td>
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<td>Cost of the conventional LNG evaporators $C_{eva}$ (10^3 $)</td>
<td>395</td>
<td>422</td>
<td>451</td>
<td>480</td>
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<td>BOP, $C_{BOP}$ (10^3 $)</td>
<td>3858</td>
<td>3615</td>
<td>3654</td>
<td>3696</td>
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<tr>
<td>Total plant investment, $C_i$ (10^3 $)</td>
<td>21,508</td>
<td>21,692</td>
<td>21,922</td>
<td>22,175</td>
</tr>
<tr>
<td>Specific cost, $C_o$ ($/kW)</td>
<td>1075</td>
<td>1085</td>
<td>1096</td>
<td>1109</td>
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<tr>
<td>Cost of O&amp;M, $C_{oM}$ (10^3 $/year)</td>
<td>860</td>
<td>868</td>
<td>877</td>
<td>887</td>
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<td>Fuel cost, $C_f$ (10^3 $/year)</td>
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<td>4250</td>
<td>4274</td>
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<td>Revenue</td>
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<tr>
<td>Electricity revenue, $r_e$ (10^3 $/year)</td>
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<td>8260</td>
<td>8260</td>
<td>8260</td>
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<tr>
<td>Net electricity revenue, $r_{n,e}$ (10^3 $/year)</td>
<td>4035</td>
<td>4010</td>
<td>3986</td>
<td>3956</td>
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<td>CO2 credit, $r_{CO2}$ (10^3 $/year)</td>
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<td>1544</td>
<td>1553</td>
<td>1563</td>
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<td>Refrigeration revenue, $r_{ref}$ (10^3 $/year)</td>
<td>3359</td>
<td>3634</td>
<td>3932</td>
<td>4242</td>
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<tr>
<td>Net annual revenue, $r_{ann}$ (10^3 $/year)</td>
<td>2906</td>
<td>2867</td>
<td>2824</td>
<td>2772</td>
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<tr>
<td>($k=0$)</td>
<td>6263</td>
<td>6501</td>
<td>6756</td>
<td>7014</td>
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<tr>
<td>Total net revenue, $R_{ann}$ (10^3 $/40 yrs)</td>
<td>94,732</td>
<td>92,984</td>
<td>91,023</td>
<td>88,721</td>
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<tr>
<td>($k=0$)</td>
<td>229,092</td>
<td>238,344</td>
<td>248,302</td>
<td>258,401</td>
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<td>COE ($/kWh)</td>
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<tr>
<td>($k=0$)</td>
<td>0.0382</td>
<td>0.0385</td>
<td>0.0388</td>
<td>0.0392</td>
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<tr>
<td>($k=1$)</td>
<td>0.0143</td>
<td>0.0126</td>
<td>0.0108</td>
<td>0.0089</td>
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<tr>
<td>Payback years, PY (with plant life of 40 years)</td>
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<tr>
<td>($k=0$)</td>
<td>5.91</td>
<td>6.01</td>
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<td>6.28</td>
</tr>
<tr>
<td>($k=1$)</td>
<td>3.12</td>
<td>3.06</td>
<td>2.97</td>
<td>2.9</td>
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</table>

* The net electricity revenue, $r_{ann}$, is defined as the sum of the electricity revenue $r_e$ minus the fuel cost $C_f$. 

---

Fig. 10. t–Q diagram of the condenser CON under different $\Delta T_{p2}$.

Fig. 11. Effect of $\Delta T_{p2}$ on the thermal efficiency $\eta_e$ and exergy efficiency $\eta_h$. 

Fig. 11. Effect of $\Delta T_{p2}$ on the thermal efficiency $\eta_e$ and exergy efficiency $\eta_h$. 

---

(i) for $k = 0$, the net annual revenue $r_{net}$ decreases, thus prolonging the system payback period $PY$ from 5.91 years to 6.28 years according to Eq. (8); (ii) for $k = 1$, $PY$ is shortened from 3.12 years to 2.9 years, mainly because of the increase in the annual value $B$ caused by the increase in the refrigeration revenue $r_{ref}$.

(4) Total net revenue $R_{net}$: As $\Delta T_{p2}$ is increased from 8 K to 17 K; (i) for $\lambda = 0$, the reduction of the net annual revenue $r_{net}$ and increase of the total plant investment $C_i$ cause a 6.3% (6,011,000 $/40$ yrs) reduction in the total net revenue $R_{net}$ according to Eq. (10) and (ii) for $\lambda = 1$, $R_{net}$ increases by 12.8% (29,309,000 $/40$ yrs).

It is interesting to note from the above that increasing the $\Delta T_{p2}$ from 8 K to 17 K is unfavorable as evaluated by $COE$, $PY$ and $R_{net}$ for $\lambda = 0$, while it is favorable for $\lambda = 1$.

It was predicted, as shown in Figs. 14–16 that, at the same $\Delta T_{p2}$ the system economic performance is much better for $\lambda = 1$ than for
5. Conclusions

A thermoeconomic analysis was performed aimed at optimization of a novel power and refrigeration cogeneration system, COOLSEP-S, which produces near-zero-CO₂ and other emissions and has high efficiency. To achieve these desirable attributes, it uses the liquefied natural gas (LNG) coldness during its revaporization. In that, we focus on the study of the thermodynamic and economic effect of the pinch point temperature differences of the two most important heat exchangers, ΔTₚ₁ of the recuperator REP, and ΔTₚ₂ of the CO₂ condenser CON in the COOLSEP-S system.

For the turbine inlet temperature of 900 °C and pressure ratio of 4, cycle net power output of 20 MW, plant life of 40 years and 7000 annual operation hours, and two extreme cases of refrigeration changes: Δ = 0 when this system has no financial benefit from the available refrigeration capacity and and Δ = 1 when all the refrigeration produced in this plant can be sold for revenue. The increase of ΔTₚ₁ from 45 K to 90 K causes the following changes:

1. The cycle thermal performance is worsened by a reduction of 17% in the thermal efficiency ηₜ, and 14% in the exergy efficiency θ.

2. The cycle economic performance is worsened too: the specific cost Cw increases by 5.4%, the cost of electricity COE increases by 13.4% (Δ = 0) and by 53.1% (Δ = 1), the system payback period PPy is prolonged by ~1.7 years (Δ = 0) and ~0.7 year (Δ = 1), the total net revenue Rnet is reduced by 31.4% (Δ = 0) and by 19.2% (Δ = 1).

The increase of ΔTₚ₂ from 8 K to 17 K causes the following changes:

1. The thermal efficiency ηₜ is reduced by a 1.8% and the exergy efficiency θ is increased by 1.2%.

2. The specific cost Cw increases by 3.2%.

3. For Δ = 0, the cycle economic performance is worsened: the cost of electricity COE increases by 2.6%, the system payback period PPy is prolonged by 0.37 year, and the total net revenue Rnet is reduced by 6.3%.

4. For Δ = 1, the cycle economic performance is improved: the cost of electricity COE decreases by 37.8%, the system payback period PPy is shortened by 0.22 years, and the total net revenue Rnet increases by 12.8%.

The resulting main recommendations are: (1) the optimal design in the considered range of parameters is at the lowest practical ΔTₚ₁ = 45 K, (2) increasing ΔTₚ₂ is unfavorable for COE, PY and Rnet for Δ = 0, but favorable for Δ = 1, (3) for the same ΔTₚ₁ or ΔTₚ₂, the system has a much better economic performance for Δ = 1 than for Δ = 0, (4) the cost of electricity in the base case (ΔTₚ₁ = 45 K, ΔTₚ₂ = 8 K) of this system is 0.0382 $/kWh (~0.3 CNV/kWh) and the payback period is 5.9 years, much lower than those of conventional coal power plants being installed at this time in China, and yet COOLSEP-S has the additional major advantage in that it produces no CO₂ emissions.

Acknowledgements

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