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A novel near-zero CO_2 emission thermal cycle with LNG cryogenic exergy utilization

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Abstract

A novel liquefied natural gas (LNG) fueled power plant is proposed, which has virtually zero CO_2 and other emissions and a high efficiency. Natural gas is fired in highly enriched oxygen and recycled CO_2 flue gas. The plant operates in a quasi-combined cycle mode with a supercritical CO_2 Rankine-like cycle and a CO_2 Brayton cycle, interconnected by the heat transfer process in the recuperation system. By coupling with the LNG evaporation system as the cycle cold sink, the cycle condensation process can be achieved at a temperature much lower than ambient and high-pressure liquid CO_2 ready for disposal can be withdrawn from the cycle without consuming additional power. The net energy and exergy efficiencies are found to be over 65 and 50%, respectively. In the case computed (but not optimized), the required total heat exchanger area is estimated to be about 460 m^2/MW electricity produced. Besides electricity and condensed CO_2, the byproducts of the plant are H_2O, liquid N_2 and A_r.

1. Introduction

Liquefied natural gas (LNG) is regarded as a relatively clean energy resource. During the process of its preparation, approximately 500 kWh energy/t LNG is consumed for compression and refrigeration and a considerable portion of this invested exergy is preserved in the LNG [1], which has a final temperature of about 110 K, much lower than that of the ambient or of seawater. The liquefaction reduces its volume 600-fold and thus makes long distance transportation convenient.

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Nomenclature

A	heat exchanger surface area (m^2)
e	specific exergy (kJ/kg)
G	mass flow rate (kg/s)
H_u	fuel LHV value (kJ/kg)
h	specific enthalpy (kJ/kg)
P	pressure (bar)
R_g	mass flow rate ratio of Brayton cycle (%), Eq. (4)
T	temperature (K)
t	temperature (°C)
s	specific entropy (kJ/kg K)
Q	heat duty (MW)
U	overall heat transfer coefficient (W/m^2 K)
W	power output (MW)
w	specific power output (kJ/kg)
ΔT_P	pinch point temperature difference (K)
η_1	energy efficiency
η_2	exergy efficiency

Subscripts

f	fuel
h	high pressure
m	intermediary pressure
L	liquefied natural gas
l	low pressure
1,2,…,26	states on the cycle flow sheet

LNG is loaded into insulated tankers and transported to receiving terminals, where it is off-loaded and first pumped to certain pressure and then revaporized and heated, by contact with seawater or with ambient air, to approximately ambient temperature for pipeline transmission to the consumers. It is thus possible to withdraw cryogenic exergy from the LNG evaporation process which otherwise will be wasted by seawater heating. This can be achieved with a properly designed thermal power cycle using the LNG evaporator as the cold sink [1–13].

Use of the cryogenic exergy of LNG for power generation 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 [1–5], Brayton cycles [6–9] 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 climatic change prompted a significant growth in research and practice of CO_2 emission mitigation in recent years. The technologies available for CO_2 capture in power plants are mainly physical and chemical absorption, cryogenic fractionation, membrane separation. The amount of energy needed for CO_2 capture could lead to the reduction of power generation efficiency by up to 10% points [14,15].

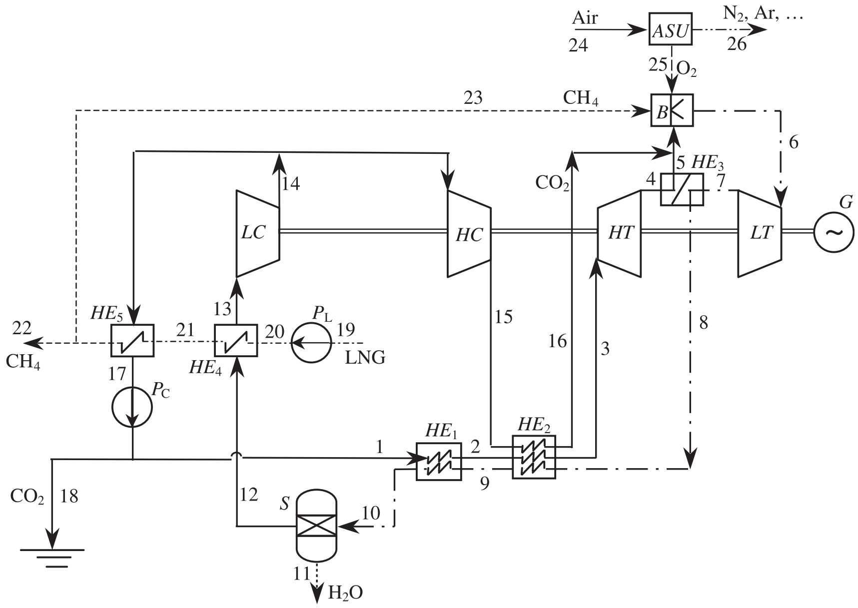
Besides the efforts for reduction of CO_2 emissions from existing power plants, concepts of power plants with zero CO_2 emission were proposed and studied. Particular attention has been paid to the research of trans-critical CO_2 cycles with fuel burning in highly enriched oxygen (99.5%+) and recycled CO_2 from the flue gas [16–24]. The common features of these cycles are the use of CO_2 as the working fluid and O_2 as the fuel oxidizer, produced by an air separation unit. With CO_2 condensation at a pressure of 60–70 bar (temperature 20–30 °C), efficiencies of 0.35–0.49 were reported for plants based on such cycles, despite the additional power use for O_2 production and CO_2 condensation. Staicovici [25] proposed an improvement to these cycles by coupling with a thermal absorption technology to lower the CO_2 condensation below ambient temperature (30 bar, −5.5 °C) and estimated a net power efficiency of 54%.

In a proposal by Velautham et al. [13], an LNG evaporation system is included in a gas–steam combined power plant just for captured CO_2 liquefaction and for air separation to provide oxygen for gas combustion. Deng et al. [9] proposed a gas turbine cycle with nitrogen as its main working fluid. The stoichiometric amount of air needed for the combustion is introduced at the compressor inlet and mixed with the nitrogen. The turbine exhaust contains mainly nitrogen, combustion generated CO_2 and H_2O. With the cycle exothermic process being integrated with the LNG evaporation process, CO_2 and H_2O are separated from the main stream by change of their phase, from gas to solid and liquid states, respectively, and the extra nitrogen is discharged. The main merit of this cycle is the absence of the air separation unit, but the combustion product may contain NO_x as well and the collection and removal of solidified CO_2 may be difficult.

In this paper, a novel zero emission CO_2 capture system is proposed and thermodynamically modeled. The plant is operated by a CO_2 quasi-combined two-stage turbine cycle with methane burning in an oxygen and recycled-CO_2 mixture. Compared to the previous works, two new features are developed in this study. The first is the integration with the LNG evaporation process. As a result, the CO_2 condensation and cycle heat sink are at temperatures much lower than ambient. The second one is the thermal cross-integration of the CO_2 Rankine-like cycle and Brayton cycle inside the recuperation system, so that the heat transfer-related irreversibility could be reduced to improve the global plant efficiency. Our cycle has both high power generation efficiency and extremely low environmental impact.

2. The cycle configuration

The cycle layout and the corresponding *t–s* diagram are shown in Figs. 1 and 2, respectively. It follows the well-established general principle of a topping Brayton cycle (working fluid here is CO_2/H_2O; TIT = 1300 °C), with heat recovery in a bottoming supercritical CO_2 Rankine cycle (TIT = 641 °C; a similar idea was first proposed by Angelino [2] in an organic Rankine cycle with CF_4 as its working fluid), but here with some sharing of the working fluids, to take best advantage of the properties of available hardware for these cycles and of good exergy management in the cycles and heat exchangers.



ASU − Air separation unit *B* − Combustor *G* − Generator *HE* − heat exchanger
LC/*HC* − Low / High pressure compressors *LT*/*HT* – Low / High pressure turbines
P_C – Liquid CO_2 pump P_L – LNG pump *S* – water separator
LNG is assumed to be CH_4 only
CO_2 H_2O LNG CO_2 / H_2O
CH_4 Air O_2 N_2, Ar, …

Fig. 1. CO_2 cycle flow sheet.

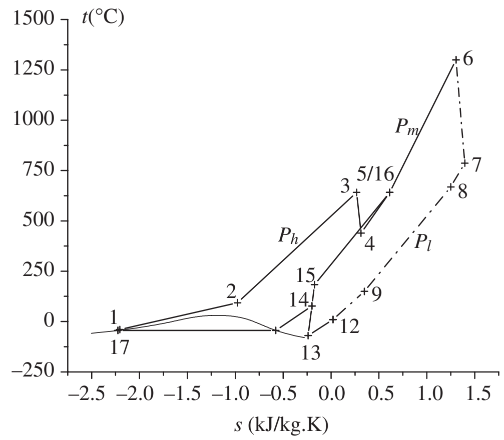


Fig. 2. *t*–*s* diagram for CO_2 cycle.

The fuel is a small fraction of the evaporated LNG and the combustion oxidizer is pure oxygen produced in a conventional cryogenic vapor compression air separation plant. The system produces power, evaporates the LNG for further use while preventing 52.2% of the LNG exergy from going to waste during its evaporation and produces liquefied CO_2 and water as the combustion products and liquid nitrogen and argon as the air separation products.

The topping Brayton cycle can be identified as $12\rightarrow13\rightarrow14\rightarrow15\rightarrow16\rightarrow6\rightarrow7\rightarrow8\rightarrow9\rightarrow10\rightarrow12$. The bottoming Rankine cycle is $17\rightarrow1\rightarrow2\rightarrow3\rightarrow4\rightarrow5\rightarrow\cdots\rightarrow14\rightarrow17$. The LNG evaporation process is $19\rightarrow20\rightarrow21\rightarrow22$ and 23. The air separation process is $24\rightarrow25$ and 26. The process material products are liquid CO_2 (18), water (11), nitrogen and argon (26), and gaseous methane (22).

The *Brayton cycle* uses its exhaust gas heat to preheat its working fluid (CO_2) before entrance to the combustor (*B*), by HE_2 and then to evaporate the working fluid (CO_2) for the Rankine cycle by HE_1, the three-pass HE_2 and HE_3. The working fluid is then cooled further, by heating the LNG in HE_4, before compression by compressors *LC* and *HC* (this cooling reduces the compression work). The first compressor, *LC*, is used then to compress the working fluid to a pressure that would allow its condensation (in HE_5, the triple point of CO_2 is 5.718 bar and −56.6 °C) and some of the working fluid is withdrawn and condensed in HE_5. The remainder of the working fluid is compressed further in *HC* to the top pressure of the Brayton cycle and then passed through the preheater HE_2 and combustor *(B)* before passing into the Brayton cycle turbine (*LT*). Assuming stoichiometric combustion, the exhaust gas of the Brayton cycle contains the combustion products CO_2 and H_2O only through the path $6\rightarrow7\rightarrow8\rightarrow9\rightarrow10$ and the H_2O is separated from the CO_2 by condensation and withdrawal in *S*. A minute amount of CO_2 may be released along with water; but it is assumed here that the water and carbon dioxide are fully separated to simplify the calculation.

In the *Rankine cycle*, the Brayton cycle recuperators HE_1 and HE_2 serve as the two-stage boiler of the working fluid (CO_2), HE_5 is the condenser using the LNG as coolant and P_C is the pump to raise the liquid CO_2 pressure to the top value of the Rankine cycle and for the withdrawal of the excess liquid CO_2 for sequestration (at 18). The Rankine cycle turbine (*HT*) exhaust is preheated by the Brayton cycle exhaust recuperator HE_3 before being brought as additional working fluid into the combustor (*B*).

The air separator (*ASU*) is assumed here to produce oxygen to the combustor (*B*) at the combustor pressure. Liquid O_2 is pumped within the *ASU* to the combustor pressure by a cryogenic pump and its cryogenic exergy is regenerated within the *ASU* (as in [25]). Further analysis is under way to integrate the air separation process into the cycle, thus taking advantage of the coldness of its products.

LNG off-loaded from its storage (19) is first pumped to its evaporation pressure (20) and then heated in the evaporation system (HE_4 (21) and HE_5 (22)) to near-ambient temperature. A small portion (~4%) of the natural gas (23) is sent to the combustor as fuel (a fuel pump may be needed when the combustion pressure in *B* is higher than the natural gas delivery pressure) and the remainder is sent out to customers via pipeline. It is assumed in this paper that LNG is pure methane. It is noteworthy that both the thermal energy required for evaporation and the power that can be produced with the cryogenic cycle depend strongly on the LNG evaporation pressure. There are different levels of delivery pressure available in the receiving terminals: supercritical pressure (typically 70 bar) for long distance pipeline network supply; subcritical pressure (typically 30 bar) for local distribution and power stations based on heavy-duty combined cycles [10]. In this paper, the subcritical natural gas evaporation process (30 bar) is considered and the influence of different evaporation parameters will be investigated in forthcoming papers.

The placement of the heat exchangers in the cycle and the choice of temperatures were made to reduce heat transfer irreversibilities. Furthermore, a combination of the high-pressure (higher heat capacity) but lower mass flow rate fluid on the Rankine cycle side of the recuperators with the low-pressure (lower heat capacity) but higher mass flow rate fluid on the Brayton cycle side is also intended for reduction of irreversibilities.

3. The cycle performance

The simulations are carried out using the commercial Aspen Plus [26] code. To simplify computation, it was assumed that the system operates at steady state, the natural gas is pure methane, the combustion is stoichiometric with CO_2 and H_2O the only combustion products, no turbine blade cooling, the work for pumping the liquid oxygen to the combustor is negligible and the stoichiometric amount of the water evacuated from the cycle does not contain dissolved CO_2. Besides, the outlet temperatures of the cold streams from HE_2 and HE_3 are set to be the same, i.e. $t_3 = t_{16} = t_5$, since the calculation results suggest a worse efficiency for $t_3 < t_{16} = t_5$. The most relevant assumptions for the calculations in this paper are summarized in Table 1.

The cycle minimal temperature is chosen as -70 °C to avoid gas condensation, since the saturated temperature of CO_2 under ambient pressure (1 bar) is -78.4 °C.

Energy efficiency is calculated as the ratio between overall power output and heat input in the topping cycle [11]

$$\eta_1 = W/(G_f H_u) \tag{1}$$

Table 1
Main assumptions for the calculation

Cycle parameter	High pressure, P_h[a] (bar)	150
	Intermediary pressure, P_m (bar)	25
	Low pressure, P_l (bar)	1
	CO_2 condensation pressure (bar)	8
	CO_2 condensation temperature (°C)	−44.1
	Lowest temperature, t_{13} (°C)	−70
	Mass flow rate ratio of Brayton cycle, R_g (%)	30
	Methane LHV, H_u (kJ/kg)	50,010
Turbine	*LT* Inlet temperature, t_6 (°C)	1300
	Isentropic efficiency (%)	88
Compressor	Pressure ratio (%)	25
	Isentropic efficiency (%)	88
Combustor	Efficiency (%)	100
	Pressure loss (%)	3
Recuperation system	Water separator working temperature (°C)	10
	Heat exchangers Pressure loss (%)	2
ASU	Specific work for O_2 separation (kJ/kg O_2)	720 [25]
Fuel pump	Efficiency (%)	77
LNG vaporization System	LNG pump efficiency (%)	77
	Pressure loss (%)	3
	Evaporation pressure (bar)	30
	Delivery temperature (°C)	15

[a] The highest pressure of the cycle is $P_1 = 156$ bar, 6 bar is for pressure losses in the heat exchangers.

where W is the overall power output from the turbines reduced by the power input to the compressors (LC and HC) and pumps (P_C,P_L), G_f is the fuel mass flow rate input. Since this cycle employs both fuel and LNG coldness (via its evaporation) as its input resources, the exergy efficiency is a more suitable criterion for performance evaluation than the fuel energy alone. It is defined here as the ratio between the obtained and consumed exergy:

$$\eta_2 = W/(G_f H_u + G_L e_L) \tag{2}$$

assuming that the fuel exergy is approximately equal to its lower heating value H_u, G_L is the treated LNG mass flow rate and e_L the exergy difference between the initial and the final states of the LNG evaporation process

$$e_L = (h_{19} - h_{22}) - T_0(s_{19} - s_{22}) \tag{3}$$

and in the subcritical evaporation case (30 bar), which is 566.8 kJ/kg $_{\mathrm{LNG}}$.

For a given mass flow rate of the cycle working medium, the mass flow rates of needed fuel, water and carbon dioxide recovered, and LNG regasified can all be determined.

With 100 kg/s mass flow rate of CO_2 at the combustor inlet taken as reference, Table 2 summarizes the parameters, including temperature, pressure, flow rate and composition, of each stream for the subcritical pressure (30 bar) and temperature of 15 °C natural gas delivery. The mass flow rate of LNG regasified is found to be 51.51 kg/s, of which about 4.17% (2.15 kg/s) should be sent to the combustor as fuel for the cycle; and the amount of water and CO_2 recovered are found to be 4.84 and 5.91 kg/s, respectively.

The computed performance of the cycle is summarized in Table 3. The total power produced is found to be 76.8 MW. Reduced by the power consumed for O_2 separation, which is roughly 6.2 MW (8.1%), the net power output is 70.6 MW, resulting in a energy efficiency (η_1) of 65.5% and exergy efficiency

Table 2
The stream parameters of CO_2 cycle

No.	t (°C)	P (bar)	G (kg/s)	Mol composition		No.	t (°C)	P (bar)	G (kg/s)	Mol composition		
				CO_2	H_2O					CO_2	CH_4	O_2
1	−39.7	156	70	1	0	14	77.4	8.12	105.91	1	0	0
2	93	153	70	1	0	15	182.9	25.5	30	1	0	0
3	641.1	150	70	1	0	16	641.1	25	30	1	0	0
4	439.1	25.5	70	1	0	17	−44.1	8	75.91	1	0	0
5	641.1	25	70	1	0	18	−39.7	156	5.91	1	0	0
6	1300	24.25	110.75	0.9	0.1	19	−162	1	51.51	0	1	0
7	786.9	1.075	110.75	0.9	0.1	20	−160.6	30.9	51.51	0	1	0
8	669.1	1.055	110.75	0.9	0.1	21	−124.8	30.45	51.51	0	1	0
9	150	1.035	110.75	0.9	0.1	22	15	30	49.36	0	1	0
10	10	1.015	110.75	0.9	0.1	23	15	30	2.15	0	1	0
11	10	1.015	4.84	0	1	24	25	1	37.07	air		
12	10	1.015	105.91	1	0	25	15	25	8.60	0	0	1
13	−70	1	105.91	1	0	26	/	/	28.47	N_2, Ar		

Combustor inlet CO_2 mass flow rate of 100 kg/s assumed as references.

Table 3
Cycle performance summary

LT turbine work (MW)	77.1
HT turbine work (MW)	16.8
LC compressor work (MW)	12.4
HC compressor work (MW)	2.9
LNG pump work (MW)	0.5
CO_2 pump work (MW)	1.4
O_2 separation work (MW)	6.2
Net power output (MW)	70.6
LNG mass flow rate (kg/s)	51.5
Fuel ratio (%)	4.17
Energy efficiency (%)	65.5
Exergy efficiency (%)	51.6

(η_2) of 51.6%. Consequently, such a plant would produce about 130 MWe if installed with the first Chinese LNG receiving terminal that has an import capacity of 3,000,000 t/yr (95 kg/s).

Figs. 3 and 4 are the *t–Q* diagrams for the recuperation system and the LNG evaporation system, respectively, where *Q* is the heat duty of a heat exchanger. Heat load distribution is not even among the different heat exchangers. The minimal temperature differences are present in HE_1 and HE_5. The pinch point in HE_1 appears at the point where the H_2O vapor contained in the hot *LT* exhaust stream begins to condense. The minimal temperature difference, ΔT_{p1}, is 4.8 K in this case and one way to raise it is to increase the flue gas temperature out of HE_1 (t_{10}), which will lead to more flue gas exhaust heat for LNG evaporation. The pinch point in HE_5 appears at the point where CO_2 begins to condense.

Table 4 shows the heat duties of the heat exchangers and the estimated required heat exchanger surface areas. There are totally five heat exchangers in the system and they can be divided into two groups: recuperators (HE_1, HE_2, HE_3) and LNG evaporators (HE_4, HE_5). The recuperators are conventional heat

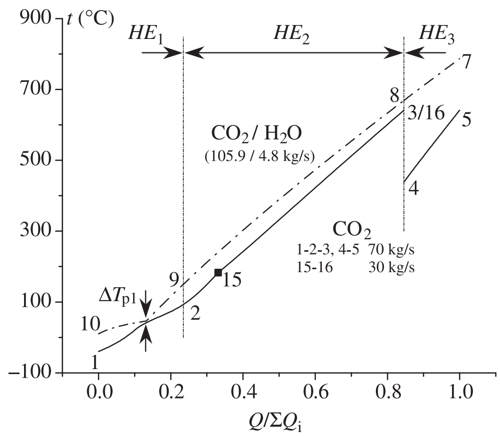


Fig. 3. *t–Q* diagram in the CO_2 recuperation system.

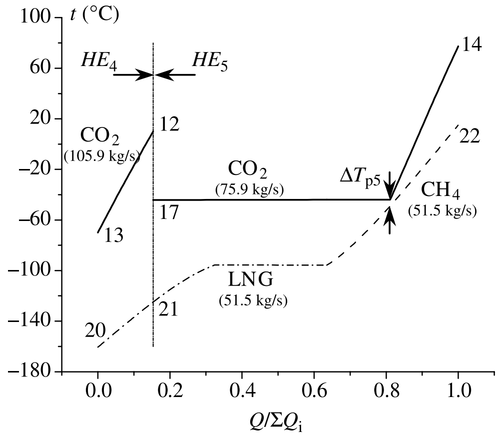


Fig. 4. t–Q diagram in the LNG evaporation system.

exchangers with gas streams flow through both sides (ignoring the small amount water condensation in HE_1). HE_4 is a CO_2 gas-to CH_4 liquid heat exchanger. As shown in Fig. 4, HE_5 consists of two parts, in the first part heat is exchanged between CO_2 gas and natural gas, in the second part CO_2 is condensed due to cooling by liquid, boiling, and gaseous CH_4 with an overall heat transfer coefficient estimated as 600 W/m^2 K [27][1]. The total heat transfer area for the cycle is estimated to be 32,651 m^2, of which the recuperators are nearly 83% and the LNG evaporators 17%, the latter accommodating about 29% of the total heat duty.

Sine the power system eliminates the need for the conventional LNG evaporator heated by ambient seawater, its heat transfer area is also estimated and reported in Table 4. To treat the same amount of LNG mass flow (51.51 kg/s), the total area is estimated to be 1196 m^2, which is about 21% of that of the LNG evaporation system and 3.7% of the total heat transfer area in the CO_2 cycle. The extra surface area and related investment is the payment for generating 70.6 MW electricity.

4. Parameter sensitivity analysis and discussion

The Brayton cycle mass flow rate ratio R_g is defined as the ratio of the mass flow rate of stream 16 (Fig. 1) over that of the total CO_2 recycled in the system

$$R_g = G_{16}/(G_5 + G_{16}) \tag{4}$$

If $R_g = 1$, the plant will turn into a pure Brayton cycle and less flue gas exhaust heat will be recovered in the recuperation system due to the sizable increase of the flue gas temperature at the inlet of the LNG evaporation system, which should equal the sum of t_{15} and a heat transfer temperature difference. At the

[1] Precise determination of heat exchanger areas requires their detailed design specification. The estimates here are very rough, 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 exchanger processes and fluids as found in the process heat transfer literature [27]. Use of better heat exchangers, such as plate type, may reduce the required heat transfer area by as much as an order of magnitude.

Table 4
Heat exchanger surface area estimations

	Heat exchanger	Q (MW)	UA (kW/K)	U [27] (W/m^{2}K)	A (m^2)	A (%)	$\sum A$ (m^2)
Recuperations	HE_1	25.39	1121.73	99	11,330.6	34.7	26,992.6
	HE_2	66.02	1366.09	93	14,689.1	45.0	
	HE_3	16.66	90.48	93	972.9	3.0	
LNG evaporators	HE_4	6.70	60.01	99	606.2	1.8	5658.5
	HE_5	36.93	1091.15	93/600	5052.3	15.5	
LNG evaporation by sea water	Water–liquid	14.21	108.94	600	181.6	15.2	1196.4
	Water-boiling	13.50	122.69	600	204.5	17.1	
	Water–steam	15.91	347.62	429	810.3	67.7	

other extreme, if $R_g = 0$, it is still a kind of quasi-combined cycle of a Brayton and a supercritical Rankine-like one, similar to the 'MATIANT' cycle [24]. While the higher heat capacity of the compressed liquid CO_2 will lead to the bigger temperature difference between *LT* outlet flue gas and CO_2 entering the combustor. Therefore, the variation of R_g will result not only in the change of the flue gas heat distribution between the recuperation system and LNG evaporation system but also in the heat balance inside the recuperation system itself.

A relatively high level for P_h and P_m was employed in past studies of power cycles with CO_2 separation, for example, they are 240 and 60 bar, respectively, in the 'COOPERATE' [19,21] and 'COOLENERG' cycles [25] and 300 and 40 bar in the 'MATIANT' cycle [24]. To relieve the technical problems incurred by these high pressure levels, the pressure P_m is chosen in our cycle to be 25 bar for the design point and its influence is examined within the range of 15–55 bar in this section. Compared with the above-mentioned cycles, our cycle has two new features: first, while $R_g = 0$ (no *HC* compressor) in those cycles, $R_g > 0$ in our cycle, which allows a much better turbine exhaust heat recovery in the recuperation system; second, integration with the LNG evaporation process accomplishes CO_2 condensation at a much lower pressure. As a result, the computed energy efficiency is as high as 65% with the enabling technologies (TIT = 1300 °C, $P_h = 150$ bar and $P_m = 25$ bar), which is about 10–15% points of increment compared with the other above-mentioned cycles. Furthermore, Brayton cycle reheat and multi-stage compression (with intercooling), common practices in industry, can be employed in our cycle to further improve performance. The analysis of such improvements is under way.

Besides R_g and P_m, the influences of some other cycle parameters are investigated as well, including low-pressure turbine (*LT*) inlet temperature t_6 and cycle high pressure P_h. Fig. 5 shows the performance under different t_6 and R_g. Fig. 5a shows that both energy efficiency and exergy efficiency increase by about 3–4% points for every 100 °C increase of t_6 or 20% increase of R_g. Increasing R_g means that more flue gas waste heat is recovered in the recuperation system and less is therefore left for LNG evaporation. Since the flue gas temperature at the inlet of the LNG evaporation system is set to be constant (10 °C), the increase of t_6 has the same effect; both will lead to the increase of the outlet temperature of cold stream from the recuperation system and hence the pinch point temperature difference in HE_1 will drop, leading to the possibility of negative ΔT_{p1} calculated for higher t_6 and R_g.

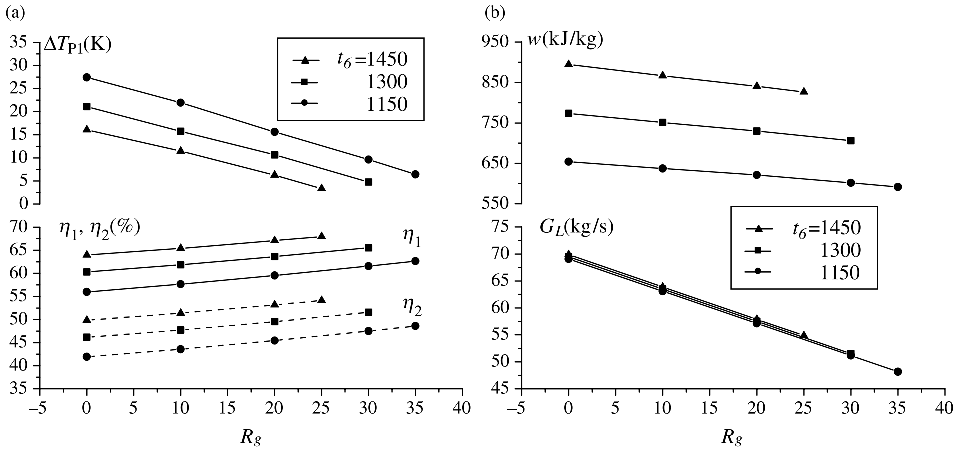


Fig. 5. The influence of t_6 and R_g ($P_h = 150$ bar, $P_m = 25$ bar).

The specific power output w increases with the increase of t_6 and with the decrease of R_g (Fig. 5b). t_6 has more significant influence on w than R_g, but its influence on the amount of LNG processed is almost negligible. The latter will decrease when R_g increases.

The influences of cycle high pressure P_h and intermediate pressure P_m are shown in Fig. 6. The increase of P_h and P_m has positive impact on the efficiencies and specific power output. When P_h increases from 150 to 200 bar for $P_m = 25$ bar, the efficiencies increase by about 0.6% point; they increase by 1.7% points when P_m increases from 15 to 25 bar for $P_h = 150$ bar. It is concluded that P_m has a more notable influence, clearly because the power output of the *LT* turbine is near 4–5 times of that of *HT* turbine. Increasing P_h and P_m result in the lowering of the *HT* and *LT* turbine flue gas temperature, respectively. Especially increasing P_h requires more heat supply in HE_3 to raise the CO_2 temperature to the desired value, which results in the lowering of t_8. This explains the reason for drop in the pinch point temperature difference under higher pressures. The influences of P_h and P_m on the amount of LNG

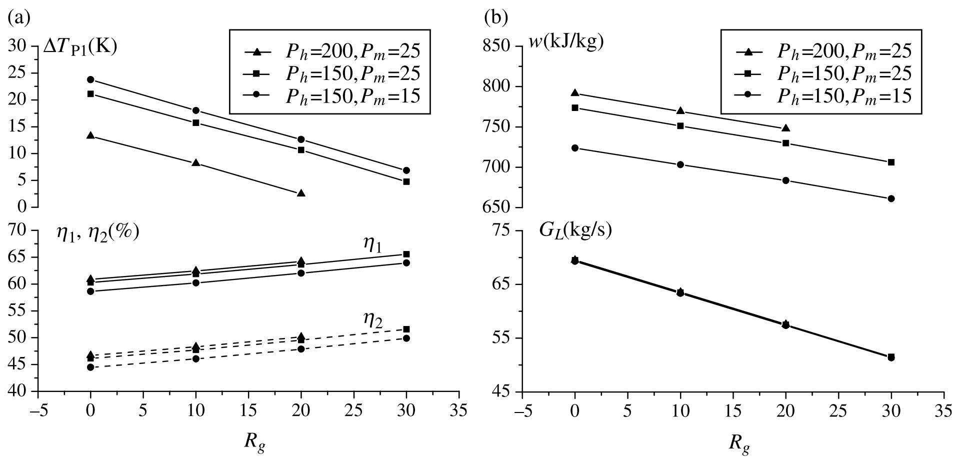


Fig. 6. The influence of P_h, P_m and R_g ($t_6 = 1300$ °C).

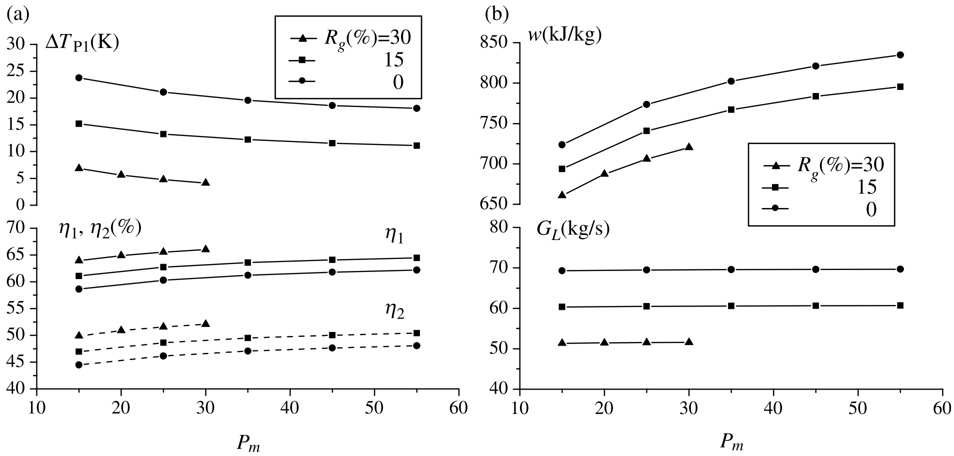


Fig. 7. The influence of P_m and R_g (t_6 = 1300 °C, P_h = 150 bar).

regasified are nearly negligible. Considering its effects on cycle efficiencies and ΔT_{p1}, it is not necessary to have very high values of P_h, since the high pressure turbine contributes less to the cycle power output.

In Figs. 7 and 8, P_m was varied from 15 to 55 bar to investigate its influence on the cycle performance. It is common sense that for a certain turbine inlet temperature, there exists a pressure ratio that produces the highest efficiencies for the Brayton cycle, but here the efficiencies are found to increase successively within the whole calculation range of P_m. The calculation is stopped at P_m = 30 and 25 bar, respectively, for R_g = 30% and P_h = 200 bar because of the constraint imposed by the pinch point ΔT_{p1} which tends to zero with increasing P_m.

The results point out that ΔT_{p1} is also a key parameter for the cycle performance. It turns up in the middle part of HE_1 due to the phase change of H_2O contained in the flue gas. Analysis of the system at constant ΔT_{p1} would be useful and will be carried out in the next step of this study.

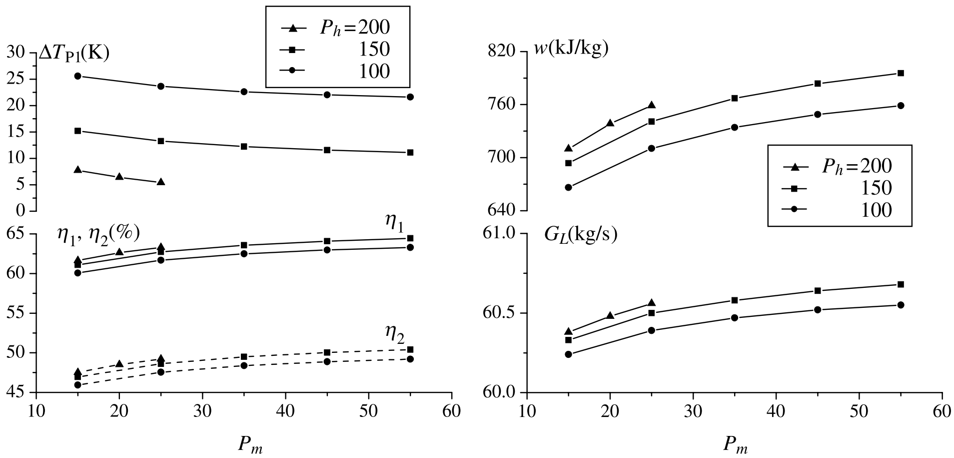


Fig. 8. The influence of P_h and P_m (t_6 = 1300 °C, R_g = 15%).

5. Conclusions

A novel power cycle producing zero CO_2 emission by integration of LNG cryogenic exergy utilization is proposed and thermodynamically modeled. The main merits of the system include:

(1) good thermodynamic performance, with the energy and exergy efficiencies reaching 66 and 52%, respectively, using conventional technologies, despite the power consumed for air separation;
(2) negligible release of pollutants to the environment;
(3) removal of high pressure liquid CO_2 ready for sale or disposal;
(4) valuable byproducts: condensed water, liquid N_2 and Ar;
(5) full exploitation of the LNG evaporation process.

The influence of some key parameters on the cycle performance, including the Brayton cycle mass flow rate ratio, the low-pressure turbine inlet temperature and pressure ratio, were investigated. It was found that the pinch point temperature difference in the recuperation system is one of the main constraints to performance improvement, its influence and parameter optimization calls for further study.

The total needed heat exchanger area is about 460 m^2/MWe, $\sim$80% of which are the recuperators HE_1 and HE_2. Employing larger heat transfer temperature differences can effectively reduce the heat transfer surface area, but will lead to a reduction of thermal efficiencies. A formal thermoeconomic optimization is obviously called for.

Based on this analysis, the proposed plant (which was not optimized yet) would produce 130 MWe if installed with the first LNG terminal in China that has an import capacity of 3,000,000 t/yr.

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