Design and Exergy Analysis for a Combined Cycle of Liquid/Solid CO_2 Production and Gas Turbine using LNG Cold/Hot Energy

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Key words: Liquefied natural gas (LNG), Cold and hot energy, Closed cycle gas turbine, Liquid/solid CO_2 production cycle, Compression power, Exergy, Irreversibility

ABSTRACT: In order to reduce the compression power and to use the overall energy contained in LNG effectively, a combined cycle is devised and simulated. The combined cycle is composed of two cycles; one is an open cycle of liquid/solid carbon dioxide production cycle utilizing LNG cold energy in CO_2 condenser and the other is a closed cycle gas turbine which supplies power to the CO_2 cycle, utilizes LNG cold energy for lowering the compressor inlet temperature, and uses the heating value of LNG at the burner. The power consumed for the CO_2 cycle is investigated in terms of a solid CO_2 production ratio. The present study shows that much reduction in both CO_2 compression power (only 35% of the power used in conventional dry ice production cycle) and CO_2 condenser pressure could be achieved by utilizing LNG cold energy and that high cycle efficiency (55.3% at maximum power condition) in the gas turbine could be accomplished with the adoption of compressor inlet cooling and regenerator. Exergy analysis shows that irreversibility in the combined cycle increases linearly as a solid CO_2 production ratio increases and most of the irreversibility occurs in the condenser and the heat exchanger for compressor inlet cooling. Hence, incoming LNG cold energy to the above components should be used more effectively.

Nomenclature

a : solid CO_2 production ratio or rate [ton/hr]
Ex_{in} : exergy input [kW]
Ex_{out} : exergy output [kW]
I : irreversibility [kW]
m : mass flow rate [kg/s]
r_P : pressure ratio
W : power [kW]

Greek symbols

ϵ : exergy efficiency
ψ : flow exergy [kJ/kg]

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1. Introduction

Recently, domestic annual energy import cost amounts to above 30 billion dollars and takes up large portion of the total import cost. Moreover, carbon dioxide from combustion of fuel and chemical process is a major concern for global warming[1-3] due to its huge amount.

Natural gas (NG) draws attention as a clean fuel and its use as an energy source increases for transportational convenience using pipeline or ship to load Liquefied Natural Gas (LNG). There are several recent research results[4-8] to use LNG cold energy (about 800 kJ/kg from −162℃ to 0℃) and part of them are of practical stage. Particularly, use of the cold energy is more effective to an energy system which requires very low temperature and constant load.

Therefore, it is a chronical need to design more effective energy system and to find a method reducing carbon dioxide emission directly or indirectly, when we consider the domestic energy situation of deficient energy resources and the worldwide movement to lay a tax on carbon dioxide emission. It can be a method to reduce indirectly CO_2 portion in the air that CO_2 emitted from industrial process is separated, recovered, liquefied/solidified, and reused to its usage including carbonated drink, cooling, fire-extinguishing, etc.[9] In addition, it is desirable to design a liquefied/solidified system in view of effective energy use because higher power is consumed for the lower liquefied/solidified temperature of gas.

In the present study, it is intended to design a combined cycle that a gas turbine cycle supplies power to a liquid/solid CO_2 production cycle using CO_2 emitted from industrial process and the cycles utilize LNG cold/hot energy. Gas turbine adopts a closed Brayton cycle to use inert gas nitrogen as a working fluid, to use high temperature source from heat exchange with LNG combustion gas, to reduce compressor power by lowering the compressor inlet temperature using LNG cold energy, and to reduce fuel consumption by exchanging heat in the regenerator between the turbine outlet gas and the compressor outlet gas. On the other hand, the liquid/solid CO_2 production cycle reduces compression power by utilizing LNG cold energy, instead of adopting cascade ammonia cooling system in the condenser.

The objective of present study is to provide a design-map. Major design parameters including compression power, mass flow rate, and temperature in the CO_2 cycle are simulated in terms of the solid CO_2 production ratio and the operating characteristics of gas turbine is also examined in terms of pressure ratio. LNG mass flow rates used for both fuel and cooling are also provided. It is difficult to evaluate the total performance of this kind of combined cycle with simple energy analysis because heat and work (LNG cold/hot energy and compression power) are invested and cold-productions and work (liquid/solid CO_2 and net work from gas turbine) are produced. Therefore, exergy analysis is performed and some suggestions for further improvement from magnitude of irreversibility for each component are shown.

2. Analysis

2.1 Combined cycle

A devised combined cycle and the corresponding T-s and P-h diagram are shown in Fig. 1 and Fig. 2, respectively.

Raw CO_2 gas is cooled by LNG after experiencing the two-stage compressions and intercoolings. Part of the cooled liquid, $(1-a)$ ton/hr, becomes the liquefied CO_2. The rest of the cooled liquid, a ton/hr, becomes the solidified CO_2 after two-stage throttling processes. The saturated vapor from the separator comes into the low-pressure compressor together with the make-up CO_2 gas, and the solidified CO_2 from the separator becomes dry-ice after pressing and forming. By utilizing LNG cold energy, the CO_2 condenser pressure is lowered up to 9 bar (the corresponding saturated temperature -42.4 ℃). In order to reduce the high-pressure compression power, the exit vapor from the low-pressure compressor is cooled by cooling-water and then cooled by the flash drum.

Gas turbine adopts a closed Brayton cycle to use inert gas nitrogen as a working fluid, to use high temperature source from heat exchange with LNG combustion gas, to reduce compressor power by lowering the compressor inlet temperature using LNG cold energy, and to reduce fuel consumption by exchanging heat in the regenerator between the turbine outlet

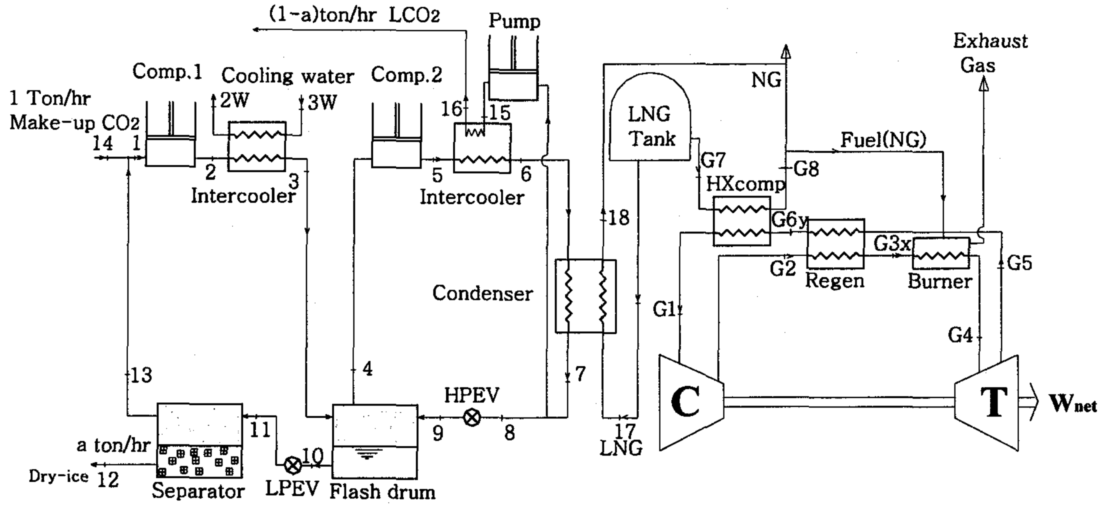

Fig. 1 Schematic diagram of a liquid/solid CO_2 production cycle with a closed cycle gas turbine.

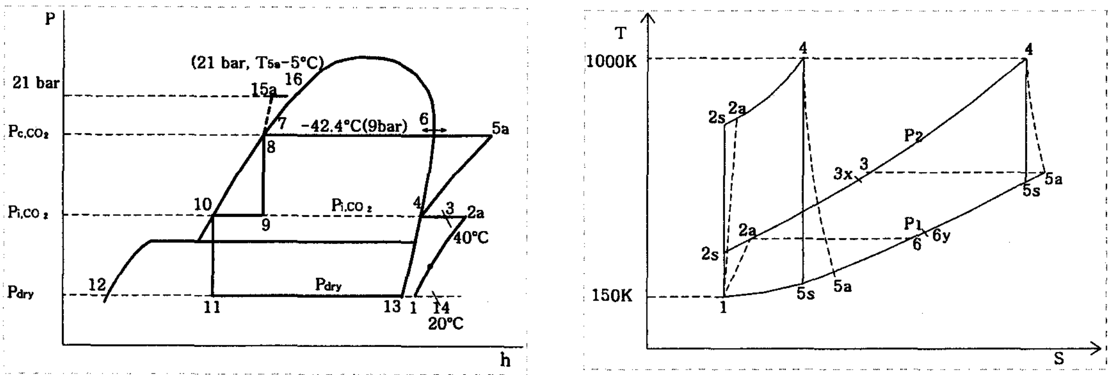

Fig. 2 P-h diagram for the liquid/solid CO_2 production cycle and T-s diagram for the gas turbine.

Table 1 Simulation conditions for a liquid/solid CO_2 production cycle with LNG cooling

Liquid/solid CO_2 cycle		Gas turbine cycle	
Variables	Values	Variables	Values
Compressor/pump efficiency, η_C, η_P	85%	Compressor efficiency, η_C	85%
Condenser pressure of CO_2, P_{c,CO_2}	9 bar	Turbine efficiency, η_t	90%
Intermediate pressure of CO_2, P_{i,CO_2}	5.5 bar	Regenerator efficiency, η_{reg}	95%
Make-up CO_2 temperature, T_{14}	20℃	Compressor inlet temperature, T_{G1}	150 K
Separator pressure, P_{dry}	1.03 bar	Compressor inlet pressure, P_{G1}	1.013 bar
Cooling water inlet temperature, T_{3w}	15℃	Turbine inlet temperature, T_{G4}	1,000 K
Cooling water outlet temperature, T_{2w}	25℃	Turbine inlet pressure, P_{G2}	5~45 bar
LNG inlet temperature in condenser, T_{17}	−152.7℃	LNG outlet temperature in HXcomp, T_{G8}	273.15 K
LNG outlet temperature in condenser, T_{18}	−50℃	Lower heating value of LNG, Q_{LHV}	50,000
LCO_2 outlet temperature in precooler, T_{16}	$(T_{5a}-5)$℃		kJ/kg
CO_2 outlet temperature in intercooler, T_3	40℃		

gas and the compressor outlet gas.

Cycle analysis through computer simulation is performed using a thermodynamic property program.[10] The simulation is based on the following assumptions. Total production of the solid and liquid CO_2 is 1 ton/hr. LNG cold energy is enough to use. In the CO_2 cycle, the inlet state of the low-pressure compressor is the superheated vapor and that of the high-pressure compressor is the saturated vapor. Pressure loss in the pipes and heat loss in the flash drum and the separator are neglected. Since the condenser pressure in the CO_2 cycle is recommended to be 9 bar,[9] simulation conditions are shown in Table 1.

2.2 Analysis for components in the CO_2 cycle

Since total production of the solid and liquid CO_2 is 1 ton/hr, the liquid CO_2 production is $(1-a)$ ton/hr for the solid CO_2 production of a ton/hr. Hence, mass flow rates of the make-up gas m_{14} kg/s, the solid CO_2 m_{12}, and the liquid CO_2 m_{15} are

$$\begin{aligned} m_{14} &= \frac{1000}{3600} \\ m_{12} &= m_{14}a \\ m_{15} &= m_{14}(1-a) \end{aligned} \tag{1}$$

Separator

From the energy equation in the separator, quality x_{11} is shown in terms of enthalpies (h_{11}, h_{12}, h_{13}).

$$x_{11} = \frac{h_{11}-h_{12}}{h_{13}-h_{12}} \tag{2}$$

The mass flow rate from the separator m_{13} is represented by

$$m_{13} = \frac{m_{13}}{m_{12}}m_{12} = \frac{x_{11}}{1-x_{11}}\frac{1000}{3600}a \tag{3}$$

The mass and energy equation for the nodal point just before the inlet of the low-pressure compressor, considering 3 states (1, 13, 14), are given by

$$m_1h_1 = m_{14}h_{14} + m_{13}h_{13} \tag{4}$$

$$m_1 = m_{13} + m_{14} = \left(\frac{a.x_{11}}{1-x_{11}} + 1\right)\frac{1000}{3600} \tag{5}$$

Flash drum

The mass flow rate m_4 of the high-pressure compressor is calculated from the mass and energy equation for the flash drum.

$$m_{10} = m_{13} + m_{12} = \frac{1000}{3600}\frac{a}{1-x_{11}} \tag{6}$$

$$m_3h_3 + m_9h_9 = m_4h_4 + m_{10}h_{10} \tag{7}$$

$$m_1 = m_2 = m_3 \tag{8}$$

$$\begin{aligned} &m_4 = m_5 = m_6 = m_7, \quad m_{15} = m_{16}, \\ &m_8 = m_9 = m_4 - m_{15} \end{aligned} \tag{9}$$

$$\begin{aligned} m_4 = &\frac{m_1}{h_4 - h_9} \\ &\times\left[h_3 - \frac{(1-x_{11})(1-a)h_9 + ah_{10}}{1-x_{11}(1-a)}\right] \end{aligned} \tag{10}$$

The validity of the above m_4 is easily found for the case of a=1.

Compressor

The exit enthalpy of compressor is calculated from the isentropic efficiency of compressor η_C.

$$h_{ea} = h_i + \frac{h_{es}-h_i}{\eta_C} \tag{11}$$

Here, subscript e represents the exit states 2 and 5, and subscript i represents the inlet states 1 and 4. Subscript a and s represent the actual and isentropic process, respectively. From these definitions, each compression power and total compression power are given by

$$\begin{aligned} W_{C1} &= m_1(h_{2a} - h_1) \\ W_{C2} &= m_4(h_{5a} - h_4) \\ W_{P\,CO2} &= m_{15}(h_{15a} - h_7) \\ W_{CO2} &= W_{C1} + W_{C2} + W_{P\,CO2} \end{aligned} \tag{12}$$

Condenser

From the energy balance for the condenser, the mass flow rate of LNG consumed for condensing CO_2 per ton/hr of CO_2, m_{17LNG} is

$$m_{17LNG} = \frac{m_4(h_6 - h_7)}{h_{18} - h_{17}} \tag{13}$$

2.3 Analysis of gas turbine component

Gas turbine is designed to operate in maximum power condition from the given conditions such as compressor inlet temperature and pressure and turbine inlet temperature. Using input values from Table 1 and thermodynamic property program,[10] power and efficiency with respect to pressure ratio are calculated. In order to verify the data validation and to examine trends of power and efficiency, the following analysis was introduced for the case of the constant specific heat of nitrogen. For convenience, subscript G representing gas turbine side was omitted.

$$\begin{aligned} w_{net} &= w_t - w_c = (h_4 - h_{5a}) - (h_{2a} - h_1) \\ &= C_P\left[T_4\eta_t\left(1 - \frac{T_{5s}}{T_4}\right) - \frac{T_1}{\eta_c}\left(\frac{T_{2s}}{T_1} - 1\right)\right] \\ &= f(r_P) \end{aligned} \tag{14}$$

Here,

$$\frac{T_{2s}}{T_1} = \frac{T_4}{T_{5s}} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = r_P^{\frac{k-1}{k}} \tag{15}$$

The optimum pressure ratio to satisfy maximum power, $r_{P,opt}$ is found by

$$r_{P,opt} = \left(\frac{P_2}{P_1}\right)_{opt} = \left(\frac{T_4}{T_1}\eta_t\eta_c\right)^{\frac{k}{2(k-1)}} \tag{16}$$

If we substitute the related values (T_1=150 K, T_4=1,000 K, η_t=0.9, η_c=0.85, and k=1.4), the optimum pressure ratio r_P=17.3 is obtained.

Assuming that the total power consumed in the solid/liquid CO_2 production is provided by the gas turbine net power, the nitrogen mass flow rate per ton/hr of CO_2 becomes

$$m_{N2} = \frac{W_{CO2}}{w_{net}} \tag{17}$$

Burner

From the energy equation for burner, the fuel consumption of LNG per ton/hr of CO_2, m_f is given by

$$m_f = \frac{m_{N2}(h_4 - h_{3x})}{Q_{LHV}} \tag{18}$$

HXcomp

From the energy equation for the compressor inlet heat exchanger, the mass flow rate of LNG needed for cooling of compressor inlet gas per 1 ton/hr of CO_2, m_{7LNG} is given by

$$m_{7LNG} = \frac{m_{N2}(h_{6y} - h_1)}{h_8 - h_7} \tag{19}$$

3. Exergy analysis

Since this combined cycle is composed of inputs with heat and work (LNG cold energy, heating value of fuel LNG, and compression power for CO_2 cycle) and outputs of cold-productions and work (liquid/solid CO_2 and net power from gas turbine), it is reasonable to evaluate the total performance of this combined cycle with exergy analysis rather than with energy analysis. Furthermore, it is necessary to perform exergy analysis to find what components use inefficiently the supplied exergy.

The exergy analysis is based on the following assumptions. All the cycle processes are in a steady-state. Kinetic energy and potential energy are neglected. Irreversibility by pressure loss in the pipes are neglected. Combustion process and heat transfer process by finite temperatures in the burner are substituted into the exergy supply process by reversible heat transfer from high temperature source. Therefore, exergy equation for a control volume in steady-state is given by

$$\sum_i m_i \psi_i + \sum_j Q_j \left(1 - \frac{T_0}{T_j}\right) = \sum_e m_e \psi_e + (W_{act,u} + I) \tag{20}$$

Here, $\psi = (h - h_0) - T_0(s - s_0)$ denotes flow exergy neglecting kinetic energy and potential energy, h and s represent enthalpy and entropy, and T_0, h_0, s_0 denote temperature, enthalpy, and entropy in the reference state, respectively. The first term of left hand side is the inflow exergy, and second term is the exergy by heat transfer. The first term of right hand side is the outflow exergy, and second term is the sum of the actual useful work transfer and irreversibility.

Applying Equation (20) to each component with the similar process described in the references,[12,13] the following irreversilities are derived. In order to avoid mistake for calculating these irreversilities, irreversilities calculated from entropy production σ are also compared and verified.

3.1 Solid/liquid CO_2 production cycle

Compressor and pump

Irreversibilities of the low-presure compressor and the high-presure compressor I_{comp1}, I_{comp2} and irreversibilities of the pump I_{pump} are given as the followings.

$$\begin{aligned} I_{comp1} &= m_1(\psi_1 - \psi_{2a}) + W_{comp1} \\ &= T_0 m_1 (s_{2a} - s_1) = T_0 \sigma_{comp1} \end{aligned} \tag{21a}$$

$$I_{comp2} = m_4(\psi_4 - \psi_{5a}) + W_{comp2} \tag{21b}$$

$$I_{pump} = m_{15}(\psi_7 - \psi_{15a}) + W_{pump} \tag{21c}$$

Intercooler and condenser

$$I_{int1} = m_1(\psi_{2a} - \psi_3) + m_{3w}(\psi_{3w} - \psi_{2w}) \tag{22a}$$

$$I_{int2} = m_4(\psi_{5a} - \psi_6) + m_{15}(\psi_{15a} - \psi_{16}) \tag{22b}$$

$$\begin{aligned} I_{cond} &= m_6(\psi_6 - \psi_7) + m_{17\mathrm{LNG}}(\psi_{17} - \psi_{18}) \\ &= T_0[m_6(s_7 - s_6) + m_{17\mathrm{LNG}}(s_{18} - s_{17})] \\ &= T_0 \sigma_{cond} \end{aligned} \tag{22c}$$

Flash drum and separator

$$I_{FD} = m_3\psi_3 - m_4\psi_4 + m_9\psi_9 - m_{10}\psi_{10} \tag{23a}$$

$$I_{sep} = m_{11}\psi_{11} - m_{12}\psi_{12} - m_{13}\psi_{13} \tag{23b}$$

High/low pressure expansion valve

$$\begin{aligned} I_{HPEV} &= m_8(\psi_8 - \psi_9) \\ &= T_0 m_8 (s_9 - s_8) = T_0 \sigma_{HPEV} \end{aligned} \tag{24a}$$

$$\begin{aligned} I_{LPEV} &= m_{10}(\psi_{10} - \psi_{11}) \\ &= T_0 m_{10}(s_{11} - s_{10}) = T_0 \sigma_{LPEV} \end{aligned} \tag{24b}$$

Total exergy balance for the CO_2 cycle

Let us take the control volume as the entire CO_2 cycle. Exergy input Ex_{in} is composed of the supplied power to the two compressors and pump, the net LNG exergy supplied to the condenser, and the net exergy of the cooling water supplied to the intercooler. Exergy output Ex_{out} is defined as the net exergy obtained from the value that the incoming exergy of raw CO_2 is subtracted from the outflow exergy of the final products, the solid and liquid CO_2. The total irreversibility I_{tot} in the control volume is the sum of each irreversibility. Hence, the total exergy balance is given as the followings.

$$Ex_{in} = Ex_{out} + I_{tot} \tag{25a}$$

$$\begin{aligned} Ex_{in} &= W_{CO2} + m_{17\mathrm{LNG}}(\psi_{17} - \psi_{18}) \\ Ex_{out} &= m_{12}\psi_{12} + m_{15}\psi_{15} - m_{14}\psi_{14} \end{aligned} \tag{25b}$$

$$I_{tot} = \sum_j I_j \tag{25c}$$

(j=comp, pump, intercooler, cond, FD, sep, HPEV, LPEV)

3.2 Gas turbine cycle

Compressor and turbine

Irreversibilities of compressor and turbine I_{Gcomp}, I_{Gturb} are given by

$$\begin{aligned} I_{Gcomp} &= m_{N2}(\psi_{G1} - \psi_{G2a}) + W_{Gcomp} \\ &= T_0 m_{N2}(s_{G2a} - s_{G1}) = T_0 \sigma_{Gcomp} \end{aligned} \quad (26)$$

$$\begin{aligned} I_{Gturb} &= m_{N2}(\psi_{G4} - \psi_{G5a}) - W_{Gturb} \\ &= T_0 m_{N2}(s_{G5a} - s_{G4}) = T_0 \sigma_{Gturb} \end{aligned} \quad (27)$$

Burner

$$I_{burn} - \sum_j Q_j \left(1 - \frac{T_0}{T_j}\right) = m_{N2}(\psi_{3x} - \psi_4) \quad (28)$$

Here, the first term of left hand side represents the irreversibility in burner and is neglected by the introduced assumptions. Therefore, the above equation represents that the exergy input by the reversible heat transfer increases the exergy of the working fluid nitrogen.

Regenerator

$$I_{reg} = m_{N2}(\psi_{G2a} - \psi_{G3x} + \psi_{G5a} - \psi_{G6y}) \quad (29)$$

HXcomp

$$\begin{aligned} I_{HXcomp} &= T_0 \sigma_{HXcomp} \\ &= m_{N2}(\psi_{G6y} - \psi_{G1}) + m_{7LNG}(\psi_{G7} - \psi_{G8}) \\ &= T_0[m_{N2}(s_{G1} - s_{G6y}) + m_{7LNG}(s_{G8} - s_{G7})] \end{aligned} \quad (30)$$

Total exergy balance for the gas turbine cycle

Let us take the control volume as the gas turbine cycle. Exergy input Ex_{Gin} is composed of the supplied exergy by the reversible heat transfer to the burner and the net LNG cold exergy supplied to the compressor inlet heat exchanger. Exergy output Ex_{Gout} is defined as the net power of gas turbine. The total irreversibility I_{Gtot} in the control volume is the sum of each irreversibility. Hence, the total exergy balance is given by the followings.

$$Ex_{Gin} = Ex_{Gout} + I_{Gtot} \quad (31a)$$

$$\begin{aligned} Ex_{Gin} &= m_{N2}(\psi_{G4} - \psi_{G3x}) \\ &\quad + m_{7LNG}(\psi_{G7} - \psi_{G8}) \end{aligned} \quad (31b)$$

$$Ex_{Gout} = W_{net}$$

$$I_{Gtot} = \sum_j I_j \quad (j = comp,\ turb,\ reg,\ HXcomp) \quad (31c)$$

3.3 Combined cycle

Let us take the control volume as the entire combined cycle. Exergy input Ex_{comIN} is composed of the supplied power to the two compressors and pump, the net LNG cold exergy supplied to both the condenser and the compressor inlet heat exchanger, the net exergy of the cooling water supplied to the intercooler, the supplied exergy by the reversible heat transfer to the burner. Exergy output Ex_{comOUT} is defined as the sum of the net power of gas turbine and the net exergy obtained from the raw CO_2 to the final products solid/liquid CO_2. The total irreversibility of the combined cycle I_{comtot} is obtained from the value that the total exergy output is subtracted from the total exergy input. Hence, the total exergy balance is given by the followings.

$$Ex_{ComIN} = Ex_{ComOUT} + I_{Comtot} \quad (32a)$$

$$\begin{aligned} Ex_{ComIN} &= Ex_{in} + Ex_{Gin} \\ Ex_{ComOUT} &= Ex_{out} + Ex_{Gout} \\ I_{Comtot} &= I_{tot} + I_{Gtot} \end{aligned} \quad (32b)$$

Defining exergy efficiency as the ratio of exergy output to exergy input, exergy efficiency for the combined cycle and each cycle is given by the followings.

$$\epsilon_{com} = \frac{Ex_{ComOUT}}{Ex_{ComIN}}, \quad \epsilon_{CO2} = \frac{Ex_{out}}{Ex_{in}}, \quad \epsilon_{GT} = \frac{Ex_{Gout}}{Ex_{Gin}} \quad (33)$$

4. Results and discussion

The solid CO_2 production ratio a ranges from 0.1 to 1.0 ton/hr. In the solid/liquid CO_2 cycle, various design parameters per ton/hr of CO_2 are presented as a function of the ratio a. In the gas turbine cycle, design parameters per kg of nitrogen are shown as a function of pressure ratio. LNG flow rates per ton/hr of CO_2 for both fuel and cooling are shown in terms of the solid CO_2 production ratio.

4.1 Operating characteristics of the CO_2 cycle

Compression power is shown in Fig. 3. Total compression power increases from 48.3 kW to 69.3 kW as a increases. Pumping power of the liquid CO_2 can be neglected. The low-pressure compression power takes up above 70% of the total power. This is due to the fact that compression power is a function of mass flow rate and pressure ratio, and that the mass flow rate of the low-pressure side is less than that of the high-pressure side, but the pressure ratio of the low-pressure side (=5.5) is much higher than that of the high-pressure side (=1.64). Since there is no production of liquid CO_2 at $a=1$, we can verify the relations in Fig. 4, $m_1=m_{10}$ and $m_4=m_8$.

In Fig. 3, we can examine the increasing trends of the discharging heat in the condenser Q_{cond}. This phenomena can be explained as follows: As the solid CO_2 production ratio of a increases, the liquid CO_2 production ratio of $(1-a)$ decreases as shown in m_{15} of Fig. 5. Among the temperatures in the intercooler, T_{5a} and T_{16}

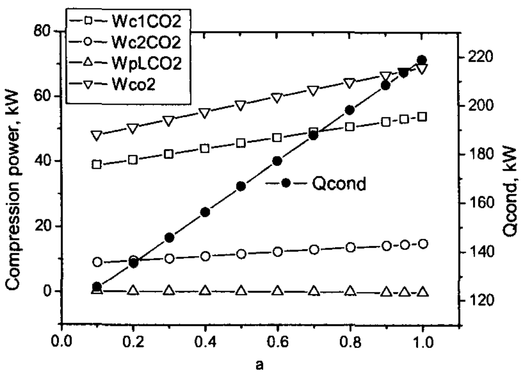

Fig. 3 Compression power and heat transfer from condenser as a function of a.

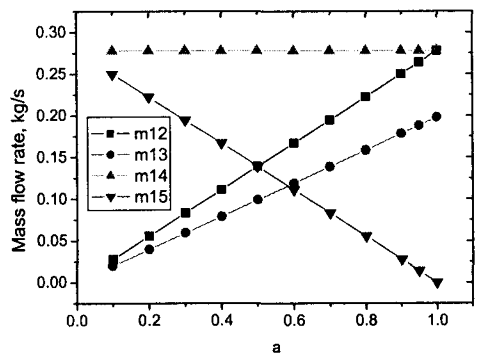

Fig. 5 Mass flow rate as a function of a.

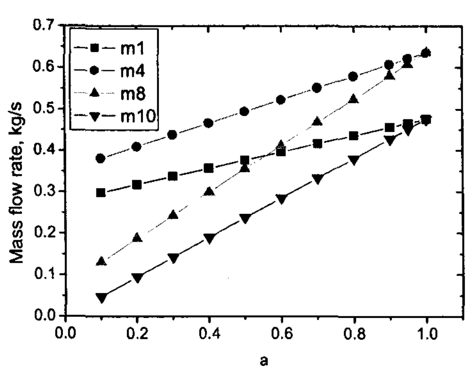

Fig. 4 Mass flow rate as a function of a.

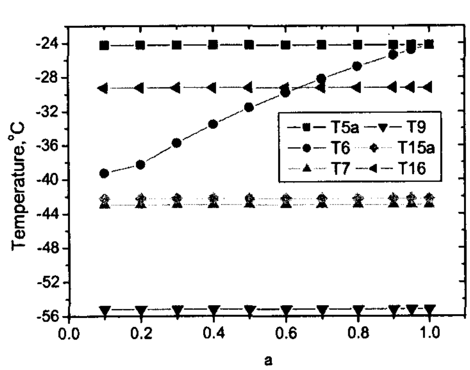

Fig. 6 Temperatures as a function of a.

are fixed and T_{15a} is nearly constant, but only T_6 increases when the production ratio of a increases, as shown in Fig. 6. As a result, the cooling in the intercooler reduces and thus the cooling load in the condenser increases as the production ratio increases.

4.2 Operating characteristics of the gas turbine cycle

Maximum net work in the gas turbine is obtained at the pressure ratio of r_P=19 as shown in Fig. 7. The efficiency of gas turbine η_{GT} decreases up to the pressure ratio of 33 and then increases. As the turbine exit temperature is lower than the compressor exit temperature above the pressure ratio of 33, we do not need regenerator.

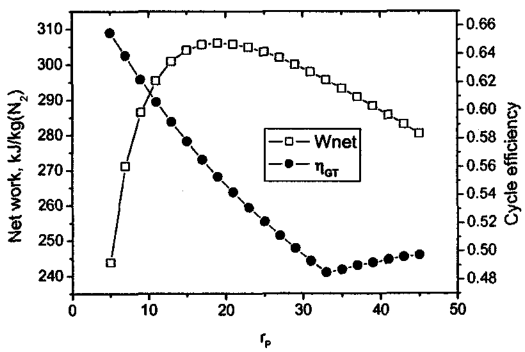

Fig. 7 Net work and cycle efficiency as a function of r_P.

In Fig. 8, LNG consumed per kg of nitrogen for both fuel and cooling of compressor inlet gas are shown. Maximum LNG consumptions for both cases are occurred at the pressure ratio of 33. This can be explained as follows: LNG consumption for fuel has a maximum from Eq. (18). And maximum LNG consumption for cooling of compressor inlet gas is caused by the maximum temperature of the regenerator exit temperature T_{G6y}. These LNG maximum consumptions correspond to the pressure ratio of 33 each other.

4.3 Mass flow rates of nitrogen and LNG

When the gas turbine is designed at maximum power condition of r_P=19, the mass flow rate of LNG consumed per ton of CO_2 for both fuel m_{fuel} and cooling of the condenser m_{17LNG} and of compressor inlet gas m_{7LNG}, and mass flow rate of nitrogen per ton of CO_2 are shown in Fig. 9. In general, it is natural that all of the above mass flow rates increase, because compression power increases if the solid CO_2 production ratio increases. Particularly, the mass flow rate of LNG for cooling of the condenser m_{17LNG} increases remarkably as the solid CO_2 production ratio increases. The mass flow rate of LNG for cooling of compressor inlet gas m_{7LNG} (0.048~0.068 kg/s) is 25 to 30 times of that of LNG for fuel m_{fuel}, and that for cool-

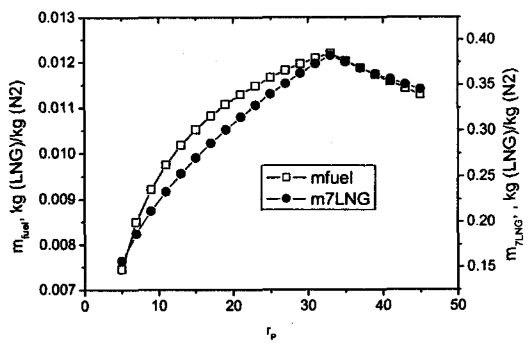

Fig. 8 Mass flow rate of LNG as a function of r_P.

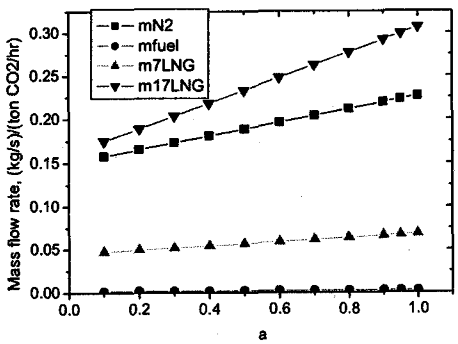

Fig. 9 Mass flow rate of LNG and N2 (per ton CO_2/hr) as a function of a.

Table 2 Results for design parameters when $a=1$

	CO_2 cycle		Gas turbine cycle						
Variable	a [ton/hr]	m_{17LNG} [kg/s]	r_P	w_{net} [kJ/kg]	$w_{net}=W_{CO2}$ [kW/(ton/hr)]	m_{N2} [kg/s]	m_f [kg/s]	m_{7LNG} [kg/s]	η_{GT}
Value	1.0	0.3064	19	306.06	69.3	0.2265	0.2507×10^{-2}	0.6815×10^{-1}	0.553

ing of the condenser m_{17LNG} is 4 to 5 times of that of LNG for cooling of compressor inlet gas.

4.4 Comparison with conventional system

Commercially, power consumed for solid CO_2 of 1 ton/hr is about 260 kW with a cascade ammonia cooling system.[11] Since this power contains additional power (associated with additional components including scrubber, desulfurizer, and dehydrator), the reference power from the previous results[12,13] neglecting these additional power is defined as about 200 kW. For comparison, present results are shown in Table 2, when the combined cycle produces only solid CO_2 and the gas turbine operates in maximum power condition. Present results shows that only about 35% of power used in the conventional ammonia cooling system is consumed by utilizing LNG cold energy.

4.5 Results of exergy analysis

From Figs. 10~11, it is observed that exergy input and output, and irreversibility for the combined cycle and for each cycle are monotonically increasing with increase of a. It is also observed that the exergy input and output of the solid/liquid CO_2 cycle take up about 70 % and 60%, respectively, out of those of the combined cycle.

Exergy efficiencies of the combined cycle and each cycle are shown in Fig. 10. With increase of a, the exergy efficiency of the combined cycle decreases from 37% to 33%, and that of the solid/liquid CO_2 cycle decreases from 31% to 26%, but that of the gas turbine cycle maintains 51%. This is due to the fact that we assumed the gas turbine operates at maximum power condition and the gas turbine is designed to change the mass flow rate of the working fluid nitrogen in spite of changing a.

In order to find the components with larger

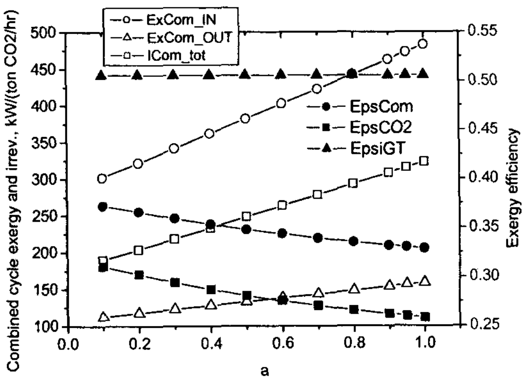

Fig. 10 Exergy and irreversibility in combined cycle, exergy efficiencies as a function of a.

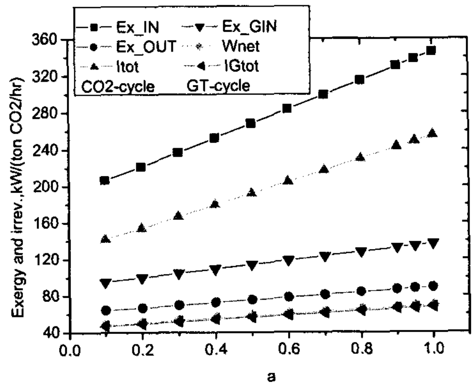

Fig. 11 Exergy and irreversibility in CO_2 cycle and gas turbine cycle as a function of a.

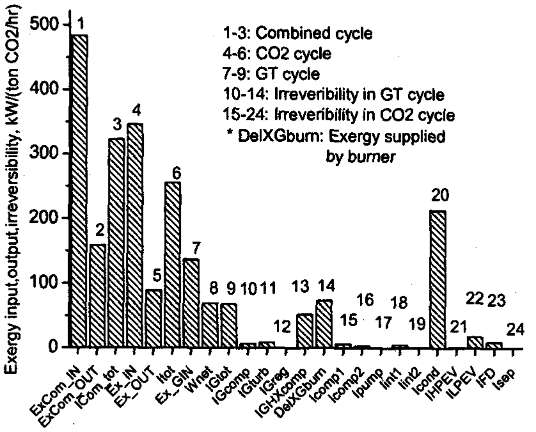

Fig. 12 Exergy input, output, and irreversibility for cycle and component when $a=1$.

irreversibility, Fig. 12 exhibits exergy input and output, and irreversibility for the combined cycle and for each cycle in the case of $a=1$. Irreversibility of the combined cycle I_{comtot} is very large, amounting to 66%. Most of this occurs in the CO_2 cycle (I_{tot}). Among the CO_2 cycle, irreversibility of the condenser (I_{cond}) has the largest portion. We can also see that in overall components of the combined cycle, the irreversibility of the condenser (I_{cond}) and that of the compressor inlet heat exchanger ($I_{GHXcomp}$) should be reduced preferentially and LNG cold energy should be used more effectively in these component.

5. Conclusions

As a mean of reducing CO_2 concentration in the air and effective use of LNG energy, a combined cycle composed of a solid/liquid CO_2 production cycle and a gas turbine as its power source was designed using LNG cold/hot energy and exergy analysis was performed. Some conclusions were drawn as follows.

(1) Compared with conventional ammonia cascade cooling system, the present combined cycle using LNG cold energy could accomplish remarkable reduction in both compression power (by 65%) and condenser pressure (from 26 bar to 9 bar), so these lead to reduction of weight and operating cost.

(2) Design parameters of the CO_2 cycle (including compression power, mass flow rates for each compressor, inlet and out temperatures of heat exchangers, LNG consumed for cooling condenser) were provided as a function of the solid CO_2 production rate.

(3) Operating characteristics of the gas turbine (including power, cycle efficiency, LNG consumed for fuel, LNG consumed for cooling compressor inlet gas) were examined as a function of pressure ratio.

(4) In designing the gas turbine with maximum power condition, several major mass flow rates (such as LNG used for fuel, LNG consumed for compressor inlet cooling, LNG for condenser cooling, and nitrogen of the gas turbine working fluid) were shown as a function of the solid CO_2 production rate.

(5) With increase of the solid CO_2 production ratio, the irreversibility of the combined cycle was increased, and the exergy efficiency was reduced from 37% to 33%.

(6) Most of irreversibility in the combined cycle occurred in the CO_2 cycle. In the CO_2 cycle, irreversibility of the condenser had the largest portion. In view of overall components of the combined cycle, the irreversibility of the condenser and that of the compressor inlet heat exchanger should be reduced preferentially and thus LNG cold energy in these component should be used more effectively.

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