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Thermodynamic analysis of an LNG fuelled combined cycle power plant with waste heat recovery and utilization system

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SUMMARY

This paper has proposed an improved liquefied natural gas (LNG) fuelled combined cycle power plant with a waste heat recovery and utilization system. The proposed combined cycle, which provides power outputs and thermal energy, consists of the gas/steam combined cycle, the subsystem utilizing the latent heat of spent steam from the steam turbine to vaporize LNG, the subsystem that recovers both the sensible heat and the latent heat of water vapour in the exhaust gas from the heat recovery steam generator (HRSG) by installing a condensing heat exchanger, and the HRSG waste heat utilization subsystem. The conventional combined cycle and the proposed combined cycle are modelled, considering mass, energy and exergy balances for every component and both energy and exergy analyses are conducted. Parametric analyses are performed for the proposed combined cycle to evaluate the effects of several factors, such as the gas turbine inlet temperature (TIT), the condenser pressure, the pinch point temperature difference of the condensing heat exchanger and the fuel gas heating temperature on the performance of the proposed combined cycle through simulation calculations. The results show that the net electrical efficiency and the exergy efficiency of the proposed combined cycle can be increased by 1.6 and 2.84% than those of the conventional combined cycle, respectively. The heat recovery per kg of flue gas is equal to $86.27\,\mathrm{kJ\,s^{-1}}$. One MW of electric power for operating sea water pumps can be saved. The net electrical efficiency and the heat recovery ratio increase as the condenser pressure decreases. The higher heat recovery from the HRSG exit flue gas is achieved at higher gas TIT and at lower pinch point temperature of the condensing heat exchanger. Copyright © 2006 John Wiley & Sons, Ltd.

KEY WORDS: combined cycle; waste heat recovery; condensing heat exchanger; LNG; thermodynamic analysis

1. INTRODUCTION

In recent years, natural gas fired gas/steam combined cycle power plant has become popular due to its high efficiency and low emissions (Andreas, 2005). However, the energy utilization is far

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from maximization in the conventional combined cycle power plant. Typically, the major sources of energy loss are the steam turbine condenser, in which the latent heat of spent steam is discarded to circulating water, and the heat recovery steam generator (HRSG) due to rejecting of the exit flue gas (Kakaras *et al*., 2004). Since the temperature of spent steam approaches the ambient temperature, recovery and utilization of the latent heat are more difficult. Although modern HRSG includes two or more pressure levels and re-heaters, which provide better recovery of the flue gas thermal energy, the temperature of the exit flue gas is still between 80 and 100°C. Generally, as high as 8% of volumetric fraction of water vapour in combustion products will be generated because hydrocarbons are the dominant components of natural gas. If the exit flue gas temperature is reduced below the dew point temperature, the water vapour in the flue gas can be condensed and both the sensible heat and latent heat released can be recovered. Previous research has concentrated on reclaiming the latent heat of the water vapour in the flue gas from heating boiler (Gordon, 1983; Shook, 1991; Che, 2002; Che *et al*., 2004). For heating boiler, the dew point of the water vapour in the flue gas is about 60°C. The efficiency of condensing boiler can be as high as, or higher than 100% if the low heating value is still taken as the calculation basis (Che *et al*., 2004). Compared to heating boiler, the dew point temperature (according to the partial pressure of water vapour in the flue gas, the dew point temperature is generally 40–50°C depending on the excess air ratio) is lower for HRSG in the combined cycle power plant. Therefore, it is difficult to recover the latent heat of water vapour in the exit flue gas. Furthermore, the waste heat recovered is hard to be utilized due to its low quality.

Natural gas fired combined cycle preferably uses the gas transported by pipeline as a fuel for the gas turbines. But the only viable way to transport the gas is to convert it into liquefied natural gas (LNG) when oceans separate the gas source and the end user. At receiving terminal, LNG, which is approximately at atmospheric pressure and at a temperature of around −160°C, has to be regasified and fed to a distribution system at the ambient temperature and at a suitably elevated pressure. Typically sea water is used as the heat source to vaporize LNG. This process not only consumes a large amount of power for driving the sea water pump but also wastes plenty of physical cold energy. With the increasing demand for cleaner fuels and the advent of larger and more reliable gas turbines for generator drivers, LNG is now playing an even significant role in power generation. Usually, LNG fuelled combined cycle power plant is located near an LNG receiving terminal. It is estimated that the amount of LNG imported to China will be 20 million tons by 2010. Therefore, the utilization of the cold energy generated during LNG vaporization becomes more and more important. Ondryas *et al*. (1991), Najjar (1996) and Kim and Ro (2000) investigated the feasibility of inlet air cooling by the cold energy of LNG to increase the power output of the conventional combined cycle power plant. Bisio and Tagliafico (2002) considered combined systems using LNG vaporization as low-temperature thermal sink. Some power generation cycles utilizing low grade heat source and the cold energy of LNG have been put forward by Hisazumi *et al*. (1998), Cheng *et al*. (2001) and Wang *et al*. (2004). The heat of spent steam from the steam turbine is used to vaporize LNG in these power generation cycles. Nevertheless, the condensation of spent steam by utilizing LNG brings about a lower condensate water temperature and causes lower feeding water temperature. Then the efficiency of HRSG is affected.

In this paper, a novel waste heat recovery and utilization system in the LNG fuelled combined cycle power plant is proposed in combination with the reclaiming of the latent heat of vaporization of water vapour contained in the flue gases. Both the heat and the cold energy that is a by-product of the combined cycle generation plants using LNG as fuel are recovered and utilized to the utmost extent in the system. The net electrical efficiency of the combined cycle power plant is improved. The proposed combined cycle provides power output as well as thermal energy with power generation as the primary goal and contributes both to saving of energy and to environmental protection.

2. PROPOSED COMBINED CYCLE

The conventional modern dual pressure combined cycle has been chosen as a reference to verify the proposed low-grade heat deep recovery and utilization system in an LNG fuelled combined cycle power plant. Figure 1 shows the process diagram of a conventional combined cycle power plant using LNG as gas turbine fuel. The air at the ambient temperature is compressed by the air compressor and directed to the combustion chamber. The compressed air mixes with the natural gas from the fuel supply system to produce hot combustion gas in the combustor. The hot combustion gas is delivered to the gas turbine where the power is generated. The exhaust gas passes through a heat recovery steam generator where water is converted to high pressure steam. The high pressure steam from the boiler drives the steam turbine. The spent steam from the turbine flows into the condenser. The fuel supply system comprises an LNG storage tank (LNG tank), an LNG pump (pump 1) and an open rack vaporizers (ORV).The ORV uses sea water as heat source to vaporize LNG.

Figure 2 shows the process diagram of an LNG fuelled combined cycle power plant with low grade waste heat recovery and utilization system. The proposed combined cycle consists of the combined cycle with gas and steam turbines, the subsystem that recovers the latent heat of spent steam from the steam turbine and vaporizes LNG, the subsystem that recovers both the sensible

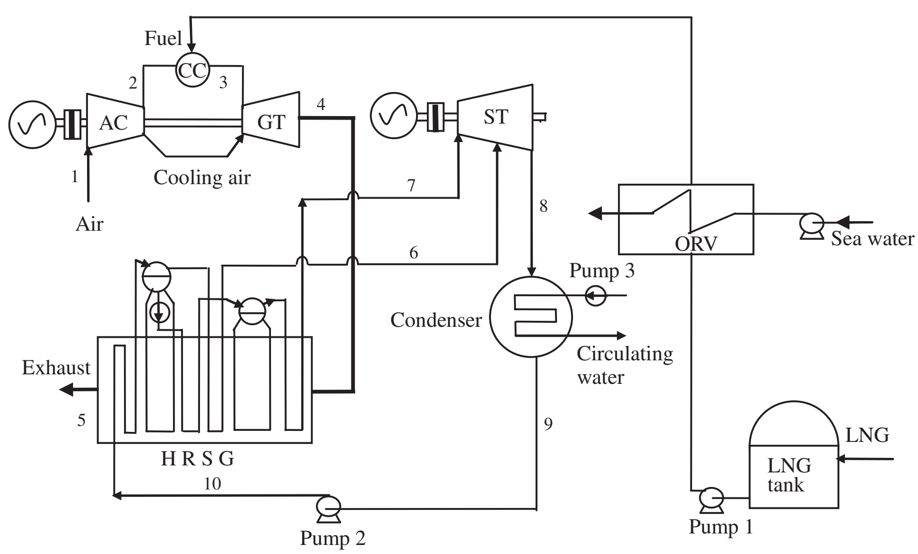

Figure 1. The conventional combined cycle power plant process flow diagram. AC, air compressor; CC, combustion chamber; GT, gas turbine; HRSG, heat recovery steam generator; ORV, open rack vaporizers; ST, steam turbine.

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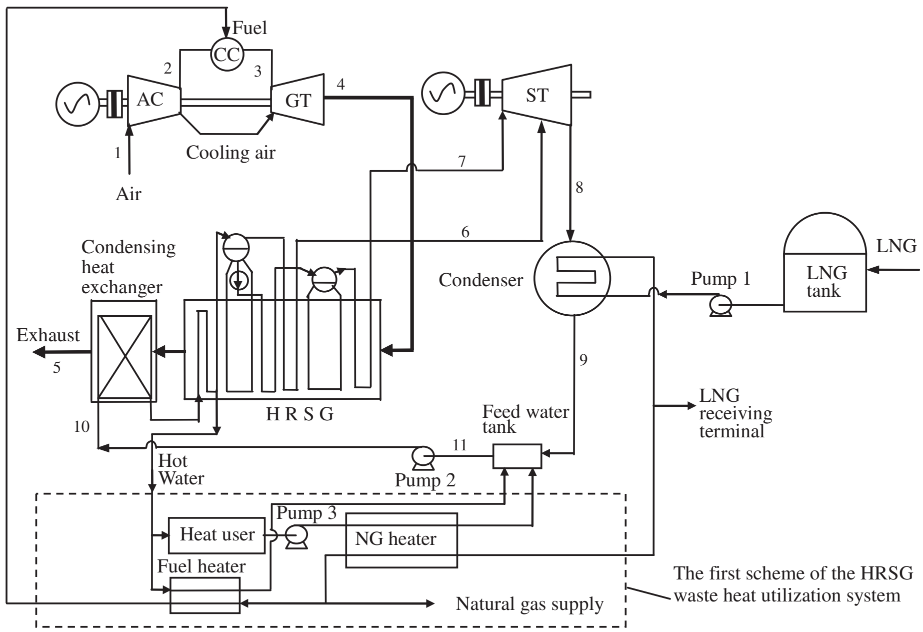

Figure 2. Process diagram of an LNG fuelled combined cycle power plant with waste heat recovery and utilization system. AC, air compressor; CC, combustion chamber; GT, gas turbine; HRSG, heat recovery steam generator; NG, natural gas; ST-steam turbine.

heat and the latent heat of exhaust gas from the HRSG by installing a condensing heat exchanger, the subsystem that utilizes hot water produced by absorbing the waste heat of the exit flue gas. The LNG at a low temperature of $-162°C$ is removed from the storage tank and pumped to the required pressure. Then the LNG enters the condenser and is vaporized through absorbing the latent heat of spent steam from the steam turbine. A fraction of vaporized LNG is delivered to the NG heater, and the remainder is distributed to the LNG receiving terminal. The spent steam is condensed by utilizing the cold energy generated during LNG vaporization. The water in the feed water tank is pumped to the condensing heat exchanger installed in the downstream flue duct of the HRSG and it absorbs both the sensible heat and latent heat of exit flue gas. The shell-and-tube heat exchanger can be selected as the condensing heat exchanger. The hot water from the condensing heat exchanger enters the low pressure economizer and is further heated to the saturation temperature corresponding to the pressure in the low pressure steam drum. The hot water is then split into two streams. One flows through the HRSG and is converted to superheated steam. The other is delivered to the HRSG waste heat utilization system, where part of it is preferably used to heat a portion of the vaporized LNG to 120°C as the fuel of the combined cycle power plant, then the remainder is firstly provided for heating system, the return water heats the vaporized LNG up to the ambient temperature, or the remaining hot water is completely utilized to heat the vaporized LNG to the ambient temperature, as shown in Figure 3. The cooled water leaving the HRSG waste heat utilization system flows into the feed water tank where it mixes with the condensate water and is recycled to recover the waste heat of exhaust gas from the HRSG.

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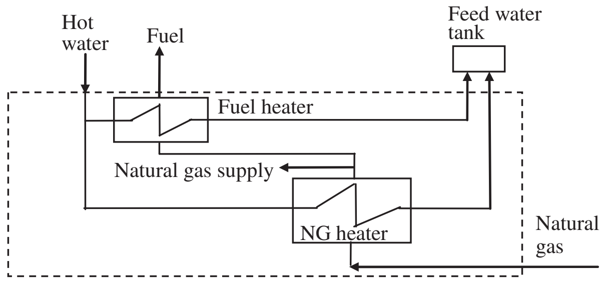

Figure 3. The second scheme of the HRSG waste heat utilization system.

The main innovations and advantages of the proposed combined cycle power plant with the low quality waste heat recovery and utilization system are as follows:

- the cold energy generated during the LNG vaporization is used to condense the spent steam from the steam turbine. Therefore, the steam condenser pressure can be reduced to a lower value for increasing the output and efficiency of the steam turbine. At the same time a great amount of electric power for driving sea water pump can be saved because sea water is no longer required as the heat source to vaporize the LNG.
- both sensible heat and latent heat of the exit flue gas from the HRSG are recovered by installing the condensing heat exchanger downstream of the HRSG. The condensate water from the condenser is mixed with the water cooled by regasified LNG as the cooling medium of the condensing heat exchanger. Compared to other power generation system utilizing LNG to condense the spent steam, the feeding water temperature at the inlet of the low pressure economizer is higher. As such, in the proposed system, the condensation of spent steam by utilizing LNG is no longer affected by the steam flow rate of the HRSG. The hot water, whose thermal conditions are the same as those of the outlet water of the low pressure economizer, which is produced by recovering the waste heat of the exit flue gas, can be sent to heating system or used for natural gas heating.
- reduction of pollutant emission since part of pollutants in the flue gas can be dissolved in the condensed water.
- a great amount of water is conserved because the condensed water from the condensing heat exchanger can be treated in a condensate polishing plant and used as makeup water for the power plant.

3. ANALYSIS

To determine the performance of the combined cycle power plant, each component must be modelled. The conventional combined cycle power plant and the combined cycle power plant with waste heat recovery and utilization system are modelled in consideration of mass, energy and exergy balances for every component. Main parameters of both cycles for the calculations are listed in Table I. For the sake of convenient comparison, all the thermal conditions of the conventional and the proposed combined cycle are identical except the condenser pressure and the fuel temperature. The values within the parentheses in the proposed combined cycle

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Table I. Main assumptions for the calculations.

		Conventional combined cycle	Proposed combined cycle
Gas turbine	Turbine inlet temperature, TIT (°C)	1350	1350 (900–1400)
	Pressure ratio	15	15
	Compressor inlet air-flow rate (kg s^{-1})	430	430
	Turbine exhaust temperature, TET (°C)	593	593
	Total coolant fraction	0.17	0.17
	Pressure loss of the compressor inlet (%)	1.0	1.0
	Pressure loss of the combustor (%)	4.0	4.0
	Pressure loss of the exhaust (%)	6.0	6.0
	Isentropic gas turbine efficiency	0.89	0.89
	Isentropic compressor efficiency	0.86	0.86
	Combustion efficiency	0.995	0.995
	Fuel temperature (°C)	15	120
	Mechanical efficiency	0.99	0.99
	Generator efficiency	0.99	0.99
Dual pressure level HRSG	Pinch point temperature difference (°C)	10	10
	Subcooling temperature difference (°C)	5	5
	HP steam pressure (MPa)	6.3	6.3
	LP steam pressure (MPa)	0.76	0.76
Steam turbine	Isentropic efficiency	0.87	0.87
	Condenser pressure (kPa)	4	1 (1–4)
	Leaving velocity loss (kJ kg^{-1})	30	30
	Pressure drop of main steam valve (%)	2	2
Others	Isentropic efficiency of pump	0.8	0.8
	Mechanical efficiency of pump	0.92	0.92
	Temperature ambient air (°C)	20	20
	Pressure ambient air (kPa)	101.325	101.325
	Lower heating value of LNG (kJ kg^{-1})	50 056	50 056
	Pinch point temperature difference of condensing heat exchanger (°C)	—	5 (5–15)

represent the variable range for parametric analysis. For simplicity, it may be assumed that all components are well insulated.

The fuel, natural gas, is assumed to be pure methane. All gases are assumed to be the mixtures of ideal gases. Their temperature-dependent thermodynamic properties and the thermodynamic properties of water and steam are all calculated with the calculation code based on Liu *et al.* (1992). The thermodynamic properties of LNG are calculated from the NIST (2003).

The modular approach is adopted, each component being represented by a module. Each module comprises a set of equations for calculating the output variables from the given input variables. The output of one module serves as the input to the subsequent module along with additional input parameters that may be required. The sequence of calculation initially follows the gas path through the gas turbine cycle. The compressor module is calculated, followed by the combustor, and gas turbine. The HRSG module is calculated subsequently. On the steam side, HRSG module is the first one to be calculated, followed by the steam turbine, condenser, and feed water pump modules. For the proposed combined cycle, condensing heat exchanger and LNG vaporizer using spent steam as heat source are also modelled and calculated. Each component is modelled as follows.

3.1. Compressor

Air-flow rate, pressure ratio, compressor mechanical efficiency and isentropic compressor efficiency are used as input data. The power required by the compressor and the air temperature at compressor outlet are calculated as follows:

$$\dot{W}_{\mathrm{comp}} = \frac{1}{\eta_{\mathrm{comp}}} \dot{m}_{\mathrm{a}} T_1 c_p (\pi^{(k-1)/k} - 1)/\eta_{\mathrm{cm}} \tag{1}$$

$$T_2 = T_1 \left[1 + \frac{1}{\eta_{\mathrm{comp}}} (\pi^{(k-1)/k} - 1) \right] \tag{2}$$

where η_{cm} is taken to be 0.99.

3.2. Combustion chamber

The combustion equation is given below

$$\beta \mathrm{CH_4} + 2\mathrm{O_2} + 2d\mathrm{N_2} = \beta(\mathrm{CO_2} + 2\mathrm{H_2O}) + 2(1-\beta)\mathrm{O_2} + 2d\mathrm{N_2} \tag{3}$$

Theoretical number of moles of air n_{A} could be derived as

$$n_{\mathrm{A}} = 2(1 + d) \tag{4}$$

where nitrogen–oxygen ratio of air d is taken as 3.77382.

Fuel temperature, compressor outlet air temperature, pressure loss of combustor and T_3 are given as input data, while the following relationships are employed to calculate fuel–air ratio f and fuel coefficient β

$$(1+f)(h_3 - h_{\mathrm{g,273\,K}}) + (h_{\mathrm{a,273\,K}} - h_2) + f(h_{\mathrm{f,273\,K}} - h_{\mathrm{f}}) = \eta_{\mathrm{comb}} f\,\mathrm{LHV} \tag{5}$$

$$\beta = A_{\mathrm{M}} \cdot f \cdot n_{\mathrm{A}}/F_{\mathrm{M}} \tag{6}$$

The molecular weight of air A_{M} and fuel F_{M} are taken to be 28.965 and 16.0142 kg kmol^{-1}, respectively. The combustor efficiency is defined as

$$\eta_{\mathrm{comb}} = \dot{Q}_{\mathrm{ch}}/(\dot{m}_{\mathrm{f}}\,\mathrm{LHV}) \tag{7}$$

Equation (5) is solved for f by an iterative procedure (Ramaprabhu and Roy, 2004).

The excess air ratio is determined by the fuel coefficient. Then the dew point of the water vapour in flue gas is evaluated by combustion calculations, which are summarized in Table II.

Table II. Combustion calculations.

Item	Symbol	Unit	Source
Theoretical air quantity	V^0	$Nm^3\,kg^{-1}$	$(22.4\times 2/CH_4)/0.21$
Excess air ratio	α	—	$1/\beta$
Theoretical water vapour volume	$V^0_{H_2O}$	$Nm^3\,kg^{-1}$	$22.4\times 2/CH_4 + 22.4/H_2O \times V^0(10/1000)$
Theoretical nitrogen gas volume	$V^0_{N_2}$	$Nm^3\,kg^{-1}$	$0.79V^0$
CO_2 volume	V_{CO_2}	$Nm^3\,kg^{-1}$	$22.4/CH^4$
Actual water vapour volume	V_{H_2O}	$Nm^3\,kg^{-1}$	$V^0_{H_2O} + 0.0161(\alpha-1)V^0$
Total flue gas volume	V_y	$Nm^3\,kg^{-1}$	$V_{CO_2} + V^0_{N_2} + V^0_{H_2O} + (1+0.0161)(\alpha-1)V^0$
Volume fraction of H_2O	r_{H_2O}	—	V_{H_2O}/V_y
Dew point	T_d	°C	Water–steam property routine by r_{H_2O}

3.3. Gas turbine

The power produced by the turbine is calculated as follows, using T_3, T_4 and isentropic efficiency of gas turbine

$$\dot{W}_{GT} = [\dot{m}_a(1-R_{cl}) + \dot{m}_f]h_3 + \dot{m}_a R_{cl} h_2 - \dot{m}_g h_4 \tag{8}$$

Net output power and efficiency of gas turbine cycle are defined as follows, respectively:

$$\dot{W}_{gt} = (\dot{W}_{GT} - \dot{W}_{comp})\eta_{EG} \tag{9a}$$

$$\eta_{gt} = \dot{W}_{gt}/(\eta_{comb}\dot{m}_f \mathrm{LHV}) \tag{9b}$$

3.4. HRSG (heat recovery steam generator)

In this study a dual-pressure HRSG, which produces two pressure levels of steam to drive the steam turbine, is selected for the analysis. The gas and steam temperature profiles of the dual-pressure HRSG are presented in Figure 4. The HRSG heating surfaces arranged in the direction of the gas flow are HP superheater, HP evaporator, LP superheater, HP economizer, LP evaporator, LP economizer and condensing heat exchanger.

Based on Tomlinson *et al.* (1993), the following input data are selected for dual-pressure HRSG: the temperature difference between the steam at high-pressure superheater outlet and the flue gas at the gas turbine exhaust outlet is 25 K. The low-pressure superheated steam temperature is 11 K lower than that of the gas leaving the low pressure-superheater section of the HRSG. The pressures of low-pressure steam drum and high-pressure steam drum are estimated as 105% of those of the low-pressure steam and the high-pressure steam, respectively. The pressures of low-pressure feed water and the high-pressure feed water are calculated as 110% of those of the low-pressure steam and the high-pressure steam, respectively. The heat balance equation for the dual-pressure HRSG can be given as

$$\dot{m}_g(h_4 - h_{gH}) = \dot{m}_{sH}(h_7 - h_{wH}) \tag{10}$$

$$\dot{m}_g(h_{gH} - h_{gL}) = \dot{m}_{sL}(h_6 - h_{wL}) + \dot{m}_{sH}(h_{wH} - h_{wL}) \tag{11}$$

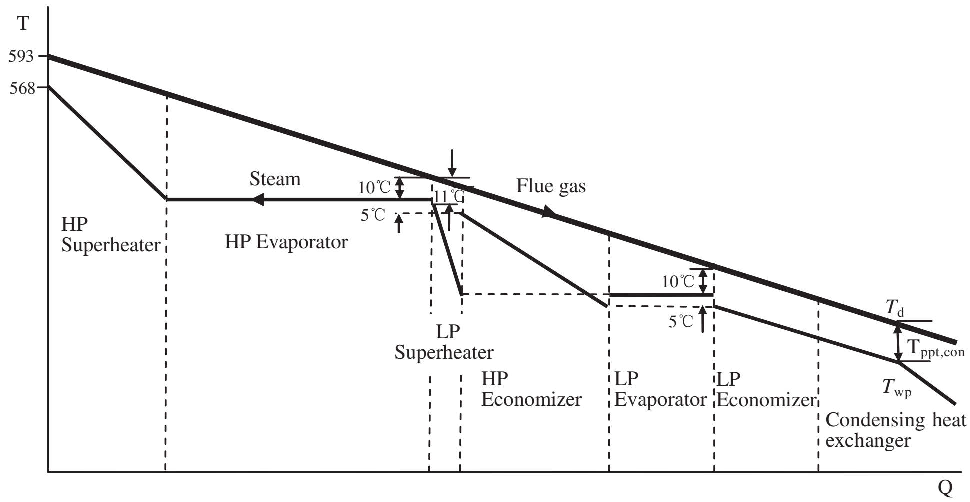

Figure 4. Gas and steam temperature profile of dual pressure HRSG.

$\dot{m}_{sH}$ and $\dot{m}_{sL}$ are calculated from Equations (10) and (11), the flue gas temperature at the LP economizer outlet can then be obtained (Yang and Xv, 2003).

The model for condensing heat exchanger is presented in Section 3.8.

3.5. Steam turbine

Steam turbine isentropic efficiency and back pressure are given as inputs. Pressure drop of main steam valve and leaving velocity loss is taken into consideration. Gross power output of steam turbine is

$$\dot{W}_{ST} = \dot{m}_{sH}(h_7 - h_8 - \Delta h_c) + \dot{m}_{sL}(h_6 - h_8 - \Delta h_c) \tag{12}$$

Net power output and efficiency of steam turbine cycle are defined as follows, respectively:

$$\dot{W}_{st} = \dot{W}_{ST}\eta_{EG} - \dot{W}_{pump} \tag{13a}$$

$$\eta_{st} = \dot{W}_{st}/(\eta_{comb}\dot{m}_f \mathrm{LHV}) \tag{13b}$$

3.6. Condenser

It is assumed that the condensate is at saturation temperature. The condenser pressure is used as input parameter in the calculations. The condenser heat duty is

$$\dot{Q}_{cond} = (\dot{m}_{sH} + \dot{m}_{sL})(h_8 - h_9) \tag{14}$$

In the proposed combined cycle power plant, the LNG in the storage tank is pumped to 3 MPa, enters the condenser where it absorbs the latent heat of the spent steam from the steam turbine and is converted into saturated gas. The mass flow rate of vaporized LNG is equal to $\dot{Q}_{cond}/r_{LNG}$, where r_{LNG} is the LNG latent heat of vaporization at the pressure of 3 MPa.

3.7. General pump model

Working fluid conditions at pump inlet, mass flow rate, pump head and pump isentropic efficiency are provided as input data. The pump power consumption is calculated as follows:

$$\dot{W}_{\text{pump}} = \dot{m}_{\text{wf}}(h_{\text{out}} - h_{\text{in}}) \tag{15}$$

3.8. Condensing heat exchanger

The water from the feed water tank is used as the cooling medium of the condensing heat exchanger which is made of corrosion resistant material. The flue gas is in counter flow with the cooling medium flow. After leaving the low pressure economizer, the flue gas enters the condensing heat exchanger and releases sensible heat to the cooling medium firstly. When flue gas temperature is reduced below the dew point, the water vapour in the flue gas condenses and latent heat is released. Therefore, the condensing heat exchanger is divided into two heat exchange regions: the single-phase heat exchange region and the region with water vapour condensing. Conceptual temperature profiles of gas and cooling medium in the condensing heat exchanger are presented in Figure 4. Where $T_{\text{ppt,con}}$ is the pinch point temperature difference of the condensing heat exchanger, which is defined as the temperature difference between the dew point T_{d} and the cooling medium temperature T_{wp} at the point where the water vapour in the flue gas begins to condense, i.e. $T_{\text{ppt,con}} = T_{\text{d}} – T_{\text{wp}}$. $T_{\text{ppt,con}}$ must not be too small, otherwise the required heat transfer area will be irrationally large.

The heat balance equation for single-phase heat exchange region

$$\dot{m}_{\text{g}}(h_{\text{gL}} - h_{\text{d}}) = \dot{m}_{\text{ew}}(h_{\text{wL}} - h_{\text{wp}}) \tag{16}$$

The heat balance equation for the region with water vapour condensing

$$\dot{m}_{\text{g}}(T_{\text{d}} - T_5) + \dot{m}_{\text{con}}r = \dot{m}_{\text{ew}}(h_{\text{wp}} - h_{10}) \tag{17}$$

where condensate mass flow rate m_{con} can be calculated in terms of the partial saturation pressures corresponding to exit flue gas temperature (Che *et al*., 2004).

From the above equation, stack gas temperature T_5 is determined by iteration, and then the recovered sensible and latent heat and HRSG efficiency can be estimated.

The HRSG efficiency is defined as

$$\eta_{\text{HRSG}} = (h_4 - h_5)/(h_4 - h_{\text{g,273 K}}) \tag{18}$$

3.9. HRSG waste heat utilization system

Part of the hot water leaving the low pressure economizer is delivered to the HRSG waste heat utilization system and its mass flow rate is $\dot{m}_{\text{ew}} - \dot{m}_{\text{sH}} - \dot{m}_{\text{sL}}$. The temperature of the cooled water leaving the system is equal to the temperature of the condensate water from the condenser. Part of vaporized LNG is heated up to the ambient temperature at constant pressure in the system and its mass flow rate is calculated from the simple energy balance equation, which has been omitted in this paper.

3.10. Exergy analysis

Exergy analysis can provide more realistic and accurate assessments of the efficiency and performance of thermal systems than those given by the more conventional energy analysis

(Rosen *et al.*, 1999; Sahin *et al.*, 2006). Therefore, it is necessary to conduct exergy analysis for both the conventional combined cycle and the proposed combined cycle.

For exergy analysis the reference conditions are taken as $T_0 = 293.15$ K and $P_0 = 1.01325$ bar. The specific exergy is defined as follows in the calculations:

For air and gas:

$$e = c_p(T - T_0) - T_0\left(c_p \ln\frac{T}{T_0} - R_g \ln\frac{p}{p_0}\right) \tag{19}$$

For LNG:

$$e = \left(\frac{T_0}{T} - 1\right)r - c_p(T_0 - T) + c_p T_0 \ln\frac{T_0}{T} + zR_g T_0 \ln\frac{p}{p_0} \tag{20}$$

For water and steam:

$$e = (h - h_0) - T_0(s - s_0) \tag{21}$$

Exergy balance equation (Dincer *et al.*, 2001):

$$\sum_i \dot{m}_i e_i + \sum_j (1 - T_0/T_j)\dot{Q} = \sum_e \dot{m}_e e_e + \dot{W} + \dot{I} \tag{22}$$

Estimation of exergy destruction in each component is given in Table III.

In the conventional combined cycle, the exergy destruction in condenser is calculated as presented in Table III because the spent steam from the steam turbine is condensed by the circulating water and most of its exergy is rejected. But for the combined cycle power plant with waste heat recovery and utilization system, the spent steam transfers heat to vaporize LNG. In order to compare with the conventional combined cycle conveniently, the same exergy destruction formula for the condenser is adopted as shown in Table III.

3.11. *Efficiency*

Power output of the conventional combined cycle power plant is

$$\dot{W}_{cc} = \dot{W}_{gt} + \dot{W}_{st} - \sum \dot{W}_{pump} \tag{23}$$

The net electrical efficiency of the conventional combined cycle power plant is defined as

$$\eta_{cc} = \dot{W}_{cc}/(\eta_{comb}\dot{m}_f \mathrm{LHV}) \tag{24}$$

Table III. Exergy destruction in each individual component.

Component	Irreversibility
Compressor	$\dot{W}_{GT} + \dot{m}_a e_{x1} - \dot{W}_{gt} - \dot{m}_a e_{x2}$
Combustion chamber	$1.04\eta_{comb}\dot{m}_f \mathrm{LHV} + \dot{m}_f e_{xf} + \dot{m}_a(1 - R_{cl})e_{x2} - [\dot{m}_a(1 - R_{cl}) + \dot{m}_f]e_{x3}$
Gas turbine	$[\dot{m}_a(1 - R_{cl}) + \dot{m}_f]e_{x3} + \dot{m}_a R_{cl} e_{x2} - \dot{m}_g e_{x4} - \dot{W}_{GT}$
HRSG	$\dot{m}_g(e_{x4} - e_{x5}) - \dot{m}_{sH}(e_{x7} - e_{x10}) - \dot{m}_{sL}(e_6 - e_{x10})$
Steam turbine	$\dot{m}_{sH}e_{x7} + \dot{m}_{sL}e_{x6} - \dot{W}_{st} - (\dot{m}_{sH} + \dot{m}_{sL})e_{x8}$
Condenser	$(\dot{m}_{sH} + \dot{m}_{sL})[(h_8 - h_9) - T_0(s_8 - s_9)]$
Mixture tank	$\dot{m}_I e_{xI} + \dot{m}_{II} e_{xII} - (\dot{m}_I + \dot{m}_{II})e_{xIII}$

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The corresponding exergy efficiency is

$$\eta_{cc,2} = \dot{W}_{cc}/\dot{E}_f \tag{25}$$

Chemical exergy of the fuel per unit time is

$$\dot{E}_f = \Delta\dot{G}_r \tag{26}$$

where the rate of Gibbs free energy decrease $\Delta\dot{G}_r$ is equal to $\Phi\eta_{com}\dot{m}_f$LHV (Kotas, 1985). For natural gas, $\Phi = 1.04 \pm 0.005$.

The useful products of the combined cycle power plant with waste heat recovery system are electric energy and thermal energy in the form of water approaching saturation at the pressure in low pressure steam drum. Based on the first law of thermodynamics the fuel utilization efficiency $\eta_{cc,fu}$ is defined by Bilgen (2000) as follows:

$$\eta_{cc,fu} = (\dot{W}_{cc} + \dot{Q}_p)/(\eta_{com}\dot{m}_f \cdot \mathrm{LHV}) \tag{27}$$

The net electrical efficiency of the proposed combined cycle is defined as

$$\eta_{cc} = \dot{W}_{cc}/(\eta_{com}\dot{m}_f \cdot \mathrm{LHV}) \tag{28}$$

Electric energy and thermal energy are at different grade. Therefore, in order to have deeper insight into the thermodynamic performance, exergy efficiency is defined as

$$\eta_{cc,2} = (\dot{W}_{cc} + \dot{B}_p)/\dot{E}_f \tag{29}$$

where the exergy content of process heat produced is evaluated as

$$\dot{B}_p = \dot{m}_w(e_{xh} - e_{xc}) \tag{30}$$

4. RESULTS AND DISCUSSION

4.1. *Results of the energy analysis*

Table IV shows the results of energy analysis for the conventional combined cycle and the proposed combined cycle based on the above-mentioned modules.

For the proposed combined cycle, the sensible heat and the latent heat of exhaust gas from the HRSG recovered by installing the condensing heat exchanger are $31.4\,\mathrm{MJ\,s^{-1}}$ in all. The condensate water temperature is reduced to 6.97°C (according to the condenser pressure 1 kPa). It is mixed with the water cooled by regasified LNG as cooling medium of the condensing heat exchanger. Then the stack temperature is lowered to 20.3°C in this ideal condition. Recovered

Table IV. Calculation results for the conventional combined cycle and the proposed combined cycle.

	$\dot{m}_f$ (kg/s)	$\dot{W}_{gt}$ (MW)	η_{gt} (%)	T_5 (°C)	η_{HRSG} (%)	$\dot{W}_{st}$ (MW)	η_{st} (%)	$\dot{W}_{cc}$ (MW)	η_{cc} (%)	$\eta_{cc,2}$ (%)	$\dot{Q}_p$ (MW)	$\eta_{cc,fu}$ (%)
Conventional combined cycle	8.834	155.95	35.27	91.1	85.97	82.24	18.6	238.18	53.9	52.05	—	—
Proposed combined cycle	8.785	155.80	35.43	20.3	96.94	88.44	20.1	244.24	55.5	54.89	31.4	62.88

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heat is used to produce hot water. The mass flow rate of the hot water entering the utilization system is equal to $46.1\,\mathrm{kg\,s^{-1}}$ under the conditions of 167.45°C, 0.84 MPa. Therefore, the efficiency of the HRSG is increased to 96.9% according to Equation (18). Compared to the conventional combined cycle, the output and efficiency of the steam turbine of the proposed combined cycle is increased by 6.6 MW and 1.6%, respectively, because the condenser pressure is reduced to 1 kPa by using the cold energy generated during the LNG vaporization. The efficiency of the gas turbine is increased by 0.16% due to the decreased fuel consumption by heating the fuel gas to 120°C in the proposed combined cycle, whereas the output of the gas turbine is decreased by 0.15 MW compared to the conventional combined cycle. The first law efficiency of the conventional combined cycle is estimated as 53.9%, the net electrical efficiency of the proposed combined cycle is estimated as 55.5%. The proposed combined cycle provides thermal energy as useful products while it mainly produces electric energy and its fuel utilization efficiency is estimated as 62.88%.

The energy flow diagram of the conventional combined cycle power plant is demonstrated in Figure 5. Each flow (in %) is obtained *via* dividing the energy at the outlet of each component by the fuel LHV. The flow represents the energy change because of heat, mass transfer or work. It can be seen from Figure 5 that the largest energy loss results from the condenser, in which 36.53% of the supplied fuel energy is lost and the stack gas energy flow leaving the HRSG is equal to 9.53%.

Figure 6 shows the energy flow of the proposed combined cycle. In contrast to 9.53% of the energy flow in the conventional combined cycle, only 2.07% of the supplied fuel energy is rejected as stack gas in the proposed combined cycle. Almost all energy of the spent steam is transferred to the LNG, so the energy loss in the condenser nearly vanishes.

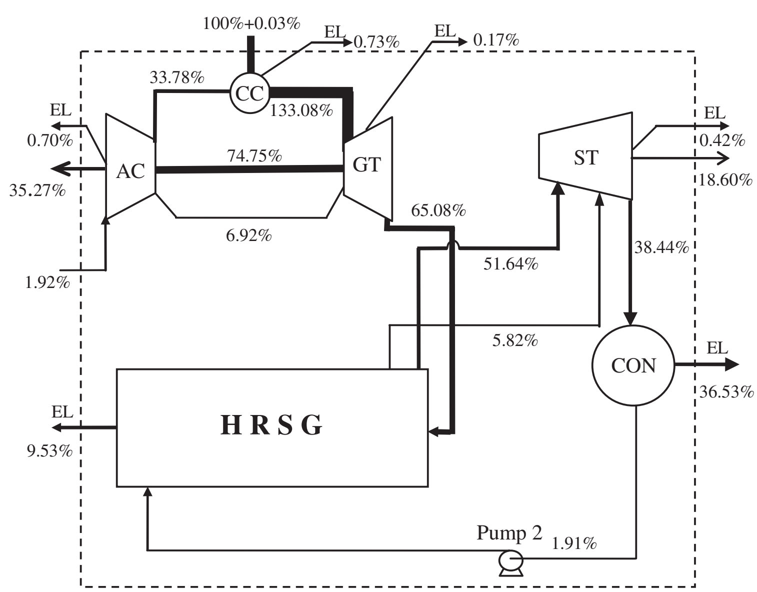

Figure 5. Energy flow diagram of the conventional combined cycle power plant. AC, air compressor; CC, combustion chamber; CON, condenser; EL, energy loss; GT, gas turbine; HRSG, heat recovery steam generator; ST, steam turbine.

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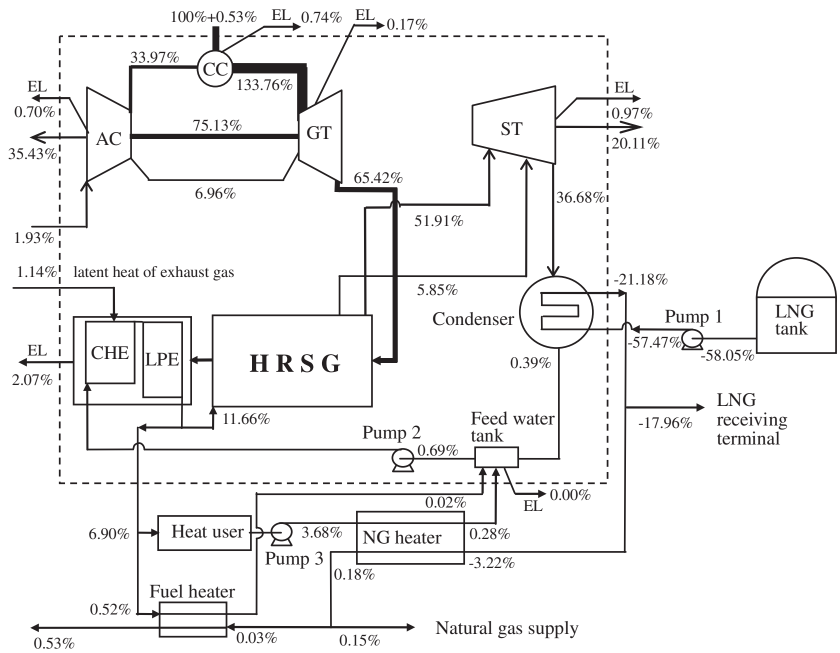

Figure 6. Energy flow diagram of the combined cycle power plant with waste heat recovery and utilization system. AC, air compressor; CC, combustion chamber; CHE, condensing heat exchanger; EL, energy loss; GT, gas turbine; HRSG, heat recovery steam generator; LPE, low pressure economizer; NG, natural gas; ST-steam turbine.

Table V. Irreversibility rate and relative irreversibility for both conventional and proposed combined cycle.

Component		Conventional combined cycle		Proposed combined cycle	
No.	Name	$\dot{I}$ (kW)	$\dot{I}/\sum\dot{I}$ (%)	$\dot{I}$ (kW)	$\dot{I}/\sum\dot{I}$ (%)
1	Compressor	12 624.2	5.75	12 622.7	6.02
2	Combustion chamber	121 201.8	55.24	123 307.9	58.77
3	Gas turbine	46 860.5	21.36	467 67.6	22.29
4	HRSG	11 593.0	5.28	105 24.2	5.02
5	Steam turbine	17 521.2	7.99	8837.9	4.21
6	Condenser	4795.7	2.19	7236.9	3.45
7	Stack	4793.9	2.19	507.3	0.24
8	Total	219 390.3	100	209 804.5	100

4.2. Results of the exergy analysis

The exergy destroyed due to irreversibility in each component of the conventional combined cycle and the proposed combined cycle is estimated in order to elucidate the impact of irreversibility on thermodynamic performance from the point of view of the second law of thermodynamics. Irreversibility rate and relative irreversibility for both considered combined cycles are shown in Table V. Figures 7 and 8 present the results of the analysis. The flows (in %)

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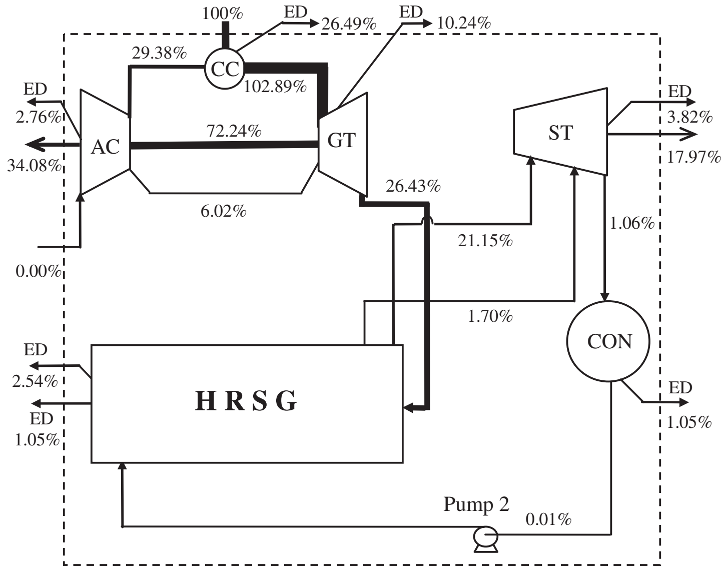

Figure 7. Exergy flow diagram of the conventional combined cycle power plant. AC, air compressor; CC, combustion chamber; CON, condenser; ED, exergy destruction; GT, gas turbine; HRSG, heat recovery steam generator; ST, steam turbine.

are computed *via* dividing the exergy at the outlet of each component by the fuel exergy. Each flow represents the exergy change because of heat, mass transfer or work, as well as the exergy destruction.

Figure 7 shows the exergy flow diagram of the conventional combined cycle power plant. The largest exergy destruction results from the combustion chamber, in which 26.49% of the supplied fuel exergy is lost, 1.05% of the supplied fuel exergy is rejected as flue gas and 1.06% of the supplied fuel exergy is released to the circulating water. The exergy efficiency of the conventional combined cycle is 52.05%.

It can be seen from Figure 8 that, similar to the conventional combined cycle power plant, the main exergy destruction of the proposed combined cycle is also from the combustion chamber. The exergy destruction of the released stack gas is nearly equal to 0.00% because the stack gas temperature drops almost to the ambient temperature by installing the condenser heat exchanger. The supplied fuel exergy of 1.21% produced by recovering the HRSG waste heat leaves the low-pressure economizer and enters the utilization system. Utilizing the cold energy generated during the LNG vaporization to reduce the condenser pressure from 4 to 1 kPa results in higher steam turbine output, and then the exergy transfer in the steam turbine cycle is increased by 1.47% compared to the equivalent exergy transfer in the conventional combined cycle power plant. Compared to the conventional combined cycle, where exergy transfer in gas turbine cycle is 34.08%, the gas turbine cycle exergy transfer of the proposed combined cycle is slightly higher (34.24%), since the fuel consumption is decreased by heating the fuel gas up to 120°C. The exergy efficiency of the proposed combined cycle is equal to 54.89%, based on Equation (29).

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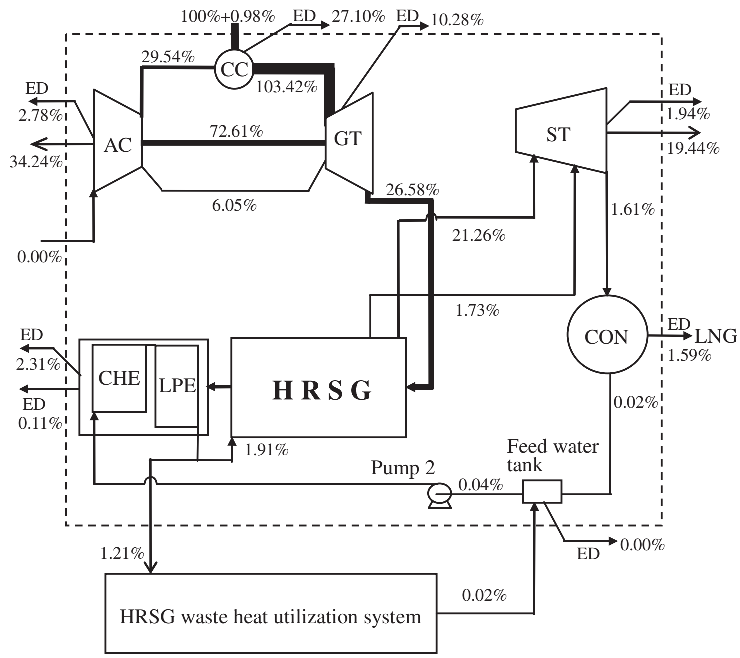

Figure 8. Exergy flow diagram of the combined cycle power plant with waste heat recovery and utilization system. AC, air compressor; CC, combustion chamber; CHE, condensing heat exchanger; ED, exergy destruction; GT, gas turbine; HRSG, heat recovery steam generator; LPE, low pressure economizer; NG, natural gas; ST, steam turbine.

4.3. Effect of condenser pressure

In the calculations, the gas turbine inlet temperature (TIT) is set to be 1350°C, the ambient temperature is taken as 15°C (it is widely accepted) and the fuel gas temperature at the combustor inlet is taken as 120°C. Other thermal conditions are still the input data shown in Table I. The performance of the combined cycle power plant with waste heat recovery and utilization system is evaluated for condenser pressures from 4 to 1 kPa. For the sake of convenient comparison, all the data are presented as the ratio of the value of the proposed combined cycle to the equivalent of the conventional combined cycle. The results are shown in Figure 9.

It can be seen that η_{cc}, W_{cc} and $\eta_{cc,fu}$ increase with decreased condenser pressure for given TIT. The effect of condenser pressure on the proposed combined cycle performance weakens as the condenser pressure increases. In the condenser pressure range of 4–1 kPa, η_{cc} is improved from 54.11 to 56.11%, W_{cc} is increased from 238 to 244 MW and $\eta_{cc,fu}$ is improved from 62.1 to 64.5%. This is mainly because lowered condenser pressure leads to higher output and efficiency of the steam turbine. Condenser pressure has no effect on the performance of gas turbine. Heating fuel can cause higher gas turbine efficiency due to the reduced fuel flow, which further increases the proposed combined cycle efficiency. Although fuel heating will result in slightly lowered gas turbine output because of decreased mass flow rate, the drop in gas turbine efficiency due to fuel heating is so small that it has negligible effect on combined cycle output.

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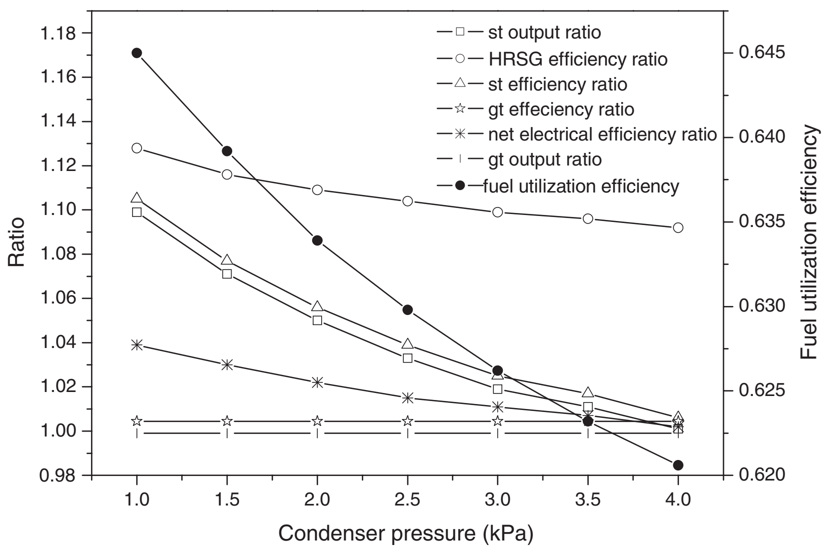

Figure 9. Effect of condenser pressure on the performance of the present combined cycle.

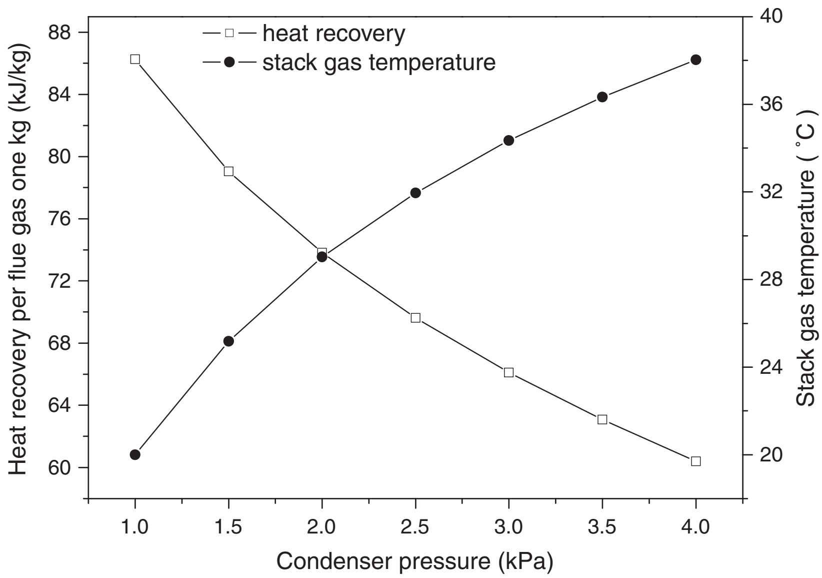

Figure 10. Effect of condenser pressure on the HRSG waste heat recovery.

For example, with the fuel gas heated to 120°C, the gas turbine output decreases only by 0.15 MW compared to the conventional combined cycle power plant.

Figure 10 gives the effect of condenser pressure on the HRSG waste heat recovery. The heat recovery ratio (heat recovery divided by the flue gas mass flow rate) increases rapidly as the condenser pressure decreases. This is because the condensate water temperature decreases with decreased condenser pressure, and lowered condensate water temperature results in a lower stack gas temperature, leading to a higher heat recovery. The variation of the stack gas temperature with condenser pressure is also demonstrated in Figure 10. For condenser pressure ranging from 4 to 1 kPa, stack gas temperature drop from 38 to 20°C, the heat recovery rate is increased from 60 to 86 kJ kg^{-1}. The maximum heat recovery (37 852.55 kJ s^{-1}) of the proposed combined cycle is reached at 1 kPa of condenser pressure.

4.4. *Effect of gas turbine inlet temperature*

The ambient temperature is 15°C, the fuel temperature at the combustor inlet is 120°C and the condenser pressure is set to be 2 kPa, other thermal conditions summarized in Table I are kept unchanged. When the pinch point temperature difference of the condensing heat exchanger is 5, 10 and 15°C, respectively, evaluation is made to clarify the effect of the gas TIT varying from 900 to 1400°C on the HRSG waste heat recovery. The calculated results are presented in Figures 11 and 12.

From the results, the heat recovered from the HRSG exit flue gas and the hot water mass flow rate almost linearly increase with the TIT for given pinch point temperature of the condensing heat exchanger. This is because the dew point and the mass flow rate of flue gas increase with increased temperature TIT, which is demonstrated in Figure 12. Increasing the dew point will cause the latent heat recovery to increase greatly in the condensing heat exchanger. The effect of the TIT on the latent heat recovery ratio (latent heat recovery divided by total heat recovery) weakens as the TIT increases, which is shown in Figure 13. The mass flow rate of flue gas and the stack temperature slightly increase as the TIT increases from 900 to 1400°C, both have an

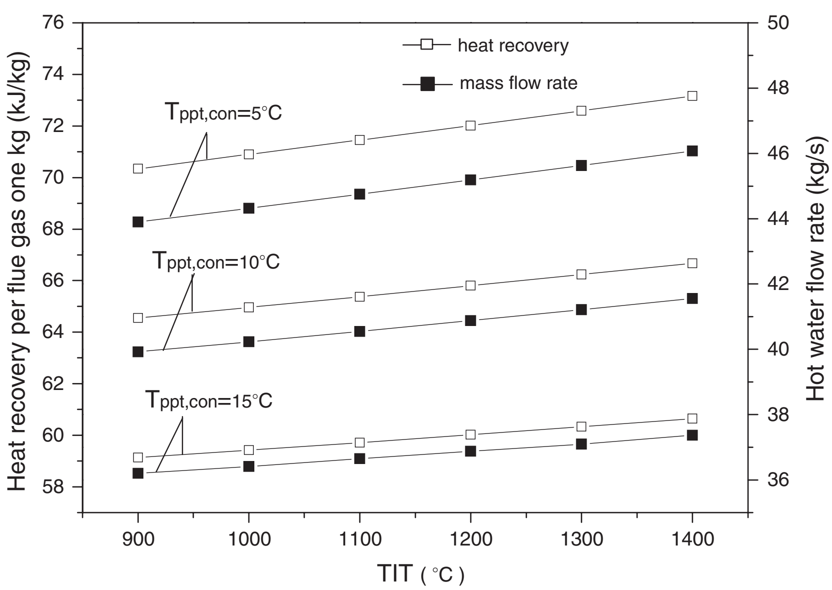

Figure 11. Effect of TIT on the HRSG waste heat recovery.

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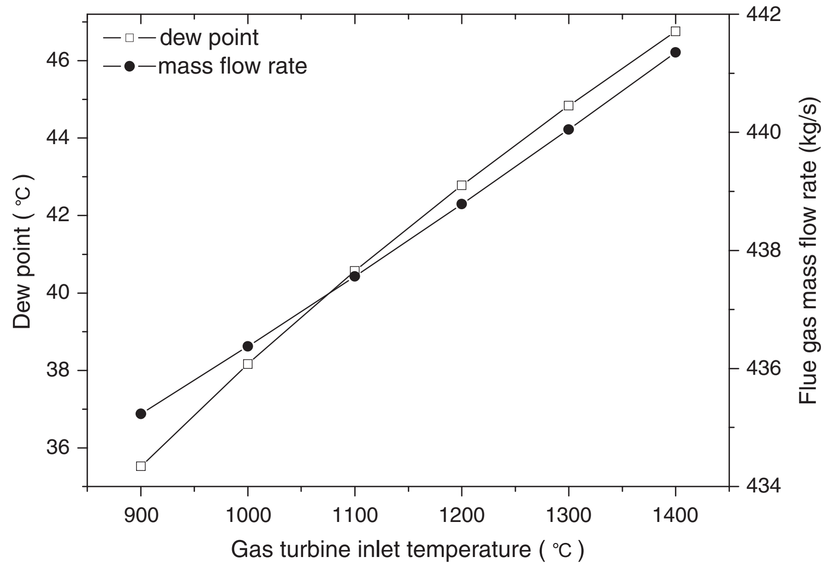

Figure 12. Effect of TIT on dew point and flue gas mass flow rate.

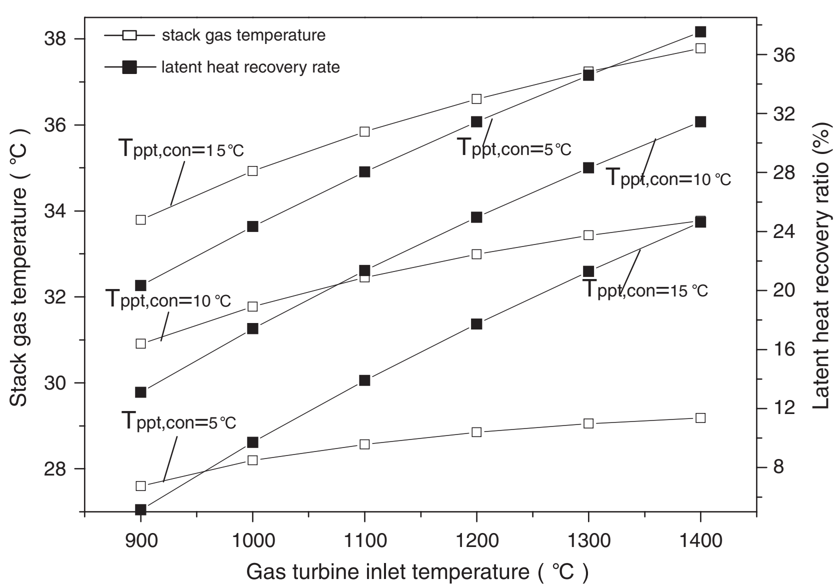

Figure 13. Effect of TIT on stack gas temperature and latent heat recovery.

insignificant effect on the heat recovery. The heat recovery and the hot water mass flow rate increase as the $T_{\text{ppt,con}}$ decreases. This is because the stack gas temperature decreases with decreased temperature $T_{\text{ppt,con}}$ after TIT, m_{g} and condensate temperature are known based on the energy balance equation of the condensing heat exchanger, and the lowered stack gas temperature results in more waste heat recovery. It can be seen that $T_{ppt,con}$ has a more significant effect on the HRSG waste heat recovery.

Increasing the gas TIT is limited by material and cooling methods. $T_{ppt,con}$ must not be decreased without limitation because heat transfer area is sharply increased with decreased temperature $T_{ppt,con}$, which results in the considerable increase in investment cost.

The hot water entering the HRSG waste heat utilization system comes from the low pressure economizer outlet, its thermal conditions are determined by low pressure and high pressure steam conditions, TET and m_g. Assume the above-mentioned thermal conditions do not change with the variation of TIT and $T_{ppt,con}$ throughout the calculations, the hot water should be kept at 167.45°C, 0.84 MPa.

4.5. *Effect of fuel heating*

The gas TIT of the proposed combined cycle is set to be 1350°C, keeping other thermal conditions unchanged as summarized in Table I. The fuel consumption rate (kg $MW^{-1} h^{-1}$) of the proposed combined cycle is calculated for different fuel gas temperatures (from the ambient temperature to 120°C). The calculated results are presented in Figure 14.

According to the calculated results, the fuel consumption rate is reduced by 0.55% and the combined cycle efficiency can be improved approximately by 0.1% when the fuel gas is preheated from 15 to 120°C. The cost for fuel gas is around US\$ 8.65/10^6 kJ (LHV). This estimation is based on the assumption that the fuel gas is pure methane and the fuel lower heating value is 50 056 kJ kg^{-1}. For a continuous operation of 24 h per day, the fuel consumption will be reduced by 4233.6 kg, the operating expense will be reduced by US\$ 1833 per day. If the service life of the power plant is 25 years, the total cost saving is US\$ 16.7 million when the fuel gas is heated to 120°C during the operating period.

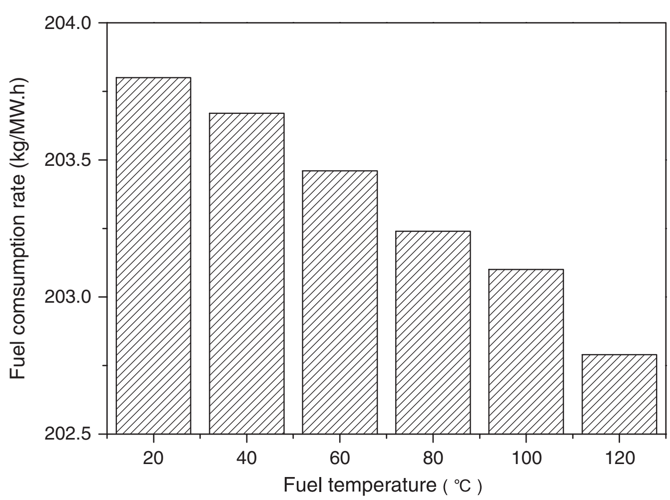

Figure 14. Relationship between fuel temperature and fuel consumption rate.

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4.6. Effect of exhaust gas pressure loss

It must be pointed out that the results discussed above are obtained without considering the exhaust gas pressure loss due to installation of the condensing heat exchanger in the downstream flue duct of the HRSG. Calculations have been made to understand the effect of the exhaust gas pressure loss on the power output of the gas turbine, based on the thermal conditions summarized in Table I. From the results, with a 1 kPa pressure drop at the exhaust gas outlet, the power output of the gas turbine can be decreased by 0.4%. A 4 kPa drop at the exhaust gas outlet may offset the power augmentation of the proposed combined cycle. However, the exhaust gas pressure drop may not be so great that power augmentation is still possible in the proposed combined cycle. In order to exactly evaluate the effect of realistic exhaust gas pressure drop caused by adding a condensing heat exchanger on the performance of the proposed combined cycle, the heat and mass transfer characteristics and the flow resistance characteristics of the condensing heat exchanger, which can be obtained by experiments, are required.

5. CONCLUSION

This paper has proposed an improved LNG fuelled combined cycle power plant with a waste heat recovery and utilization system and thermodynamic analyses have been carried out based on the model developed. The direct results of this proposed cycle include the increase total output and the additional production of hot water. The following conclusions can be drawn:

(1) When the condenser pressure is decreased to 1 kPa, the net electrical efficiency and the exergy efficiency of the proposed combined cycle are increased by 1.6 and 2.84% than those of the conventional combined cycle, respectively. The fuel utilization efficiency of the proposed combined cycle reaches 62.88%. About 46.1 $kg\,s^{-1}$ of hot water, whose conditions are 167.45°C, 0.84 MPa, can be extracted from the low pressure economizer outlet and is delivered to the utilization system and the heat recovery per kg of flue gas is equal to 86.27 $kJ\,s^{-1}$. About 88 $kg\,s^{-1}$ of LNG can be heated up to 20°C in the first HRSG waste heat utilization system.

(2) The net electrical efficiency and the heat recovery ratio increase as the condenser pressure decreases for a given TIT. This is mainly because that lowered condenser pressure results in a higher output of the steam turbine and lower stack gas temperature, leading to higher heat recovery.

(3) Higher heat recovery from the HRSG exit flue gas and higher hot water mass flow rate are achieved at higher gas TIT and lower pinch point temperature difference of the condensing heat exchanger $T_{ppt,con}$. This is because the dew point and the mass flow rate of flue gas increase with increased temperature TIT and the stack gas temperature decreases with decreased $T_{ppt,con}$. It can be seen that $T_{ppt,con}$ has a more significant effect on the HRSG waste heat recovery.

(4) Heating fuel results in higher gas turbine efficiency due to the reduced fuel flow, which further increases the combined cycle efficiency. The fuel consumption rate is reduced by 0.55% and the combined cycle efficiency can be improved by approximately 0.1% when the fuel gas is preheated from 15 to 120°C.

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(5) In the proposed combined cycle, low quality waste heat is used to vaporize LNG. About $300\,\mathrm{t\,h^{-1}}$ of LNG can be heated up to 20°C in the second HRSG waste heat utilization system, 1 MW of electric power for operating sea water pumps can be saved due to eliminating about $12\,000\,\mathrm{t\,h^{-1}}$ of sea water as the heat source to vaporize $300\,\mathrm{t\,h^{-1}}$ of LNG.

According to the analytical results, the low-quality waste heat deep recovery and utilization system in an LNG fuelled combined cycle power plant proposed in this paper appears feasible. However, in order to determine if this system is also profitable, a techno-economic study based on the results of the thermodynamic analyses is expected.

NOMENCLATURE

A_{M} = molecular weight of air ($\mathrm{kg\,kmol^{-1}}$)
$\dot{B}_{\mathrm{p}}$ = exergy content of process heat produced (kW)
c_p = specific heat at constant pressure ($\mathrm{kJ\,kg^{-1}\,K^{-1}}$)
d = nitrogen–oxygen ratio of air
e = specific exergy ($\mathrm{kJ\,kg^{-1}}$)
$\dot{E}_{\mathrm{f}}$ = chemical exergy of the fuel (kW)
e_{xh} = specific exergy of the supply water ($\mathrm{kJ\,kg^{-1}}$)
e_{xc} = specific exergy of the return water ($\mathrm{kJ\,kg^{-1}}$)
f = fuel–air ratio
F_{M} = the molecular weight of fuel ($\mathrm{kg\,kmol^{-1}}$)
h = specific enthalpy ($\mathrm{kJ\,kg^{-1}}$)
h_{d} = flue gas enthalpy at dew point ($\mathrm{kJ\,kg^{-1}}$)
h_{f} = physical enthalpy of fuel at the combustor inlet ($\mathrm{kJ\,kg^{-1}}$)
h_{gH} = enthalpy of flue gas leaving the HP evaporator ($\mathrm{kJ\,kg^{-1}}$)
h_{gL} = enthalpy of flue gas leaving the LP evaporator ($\mathrm{kJ\,kg^{-1}}$)
h_{wH} = feedwater enthalpy at HP evaporator inlet ($\mathrm{kJ\,kg^{-1}}$)
h_{wL} = feedwater enthalpy at LP evaporator inlet ($\mathrm{kJ\,kg^{-1}}$)
$\dot{I}$ = irreversibility (kW)
k = specific heat ratio of air
$\dot{m}$ = mass flow rate ($\mathrm{kg\ s^{-1}}$)
$\dot{m}_{\mathrm{con}}$ = condensate mass flow rate ($\mathrm{kg\,s^{-1}}$)
$\dot{m}_{\mathrm{ew}}$ = mass flow rate of the water leaving the low pressure economizer ($\mathrm{kg\,s^{-1}}$)
$\dot{m}_{\mathrm{w}}$ = the mass flow rate of hot water ($\mathrm{kg\,s^{-1}}$)
$\dot{m}_{\mathrm{wf}}$ = the mass flow rate of working fluid ($\mathrm{kg\,s^{-1}}$)
n_{A} = theoretical number of moles of air (mol)
P = pressure absolute (MPa)
$\dot{Q}$ = heat transfer rate (kW)
$\dot{Q}_{\mathrm{ch}}$ = chemical energy released rate of fuel in the combustor (kW)
$\dot{Q}_{\mathrm{p}}$ = thermal energy of process heat (kW)
r = latent heat of vaporization ($\mathrm{kJ\,kg^{-1}}$)
R_{g} = gas constant ($\mathrm{kJ\,kmol^{-1}\,K^{-1}}$)
R_{cl} = total cooling air-flow rate divided by compressor inlet air-flow rate

s = specific entropy ($\mathrm{kJ\,kg^{-1}\,K^{-1}}$)
T = temperature (K)
$T_{\mathrm{ppt,con}}$ = pinch point temperature difference of the condensing heat exchanger (K)
$\dot{W}$ = work (kW)
z = compress factor

Abbreviation

CHE = condensing heat exchanger
HRSG = heat recovery steam generator
LHV = lower heating value ($\mathrm{kJ\,kg^{-1}}$)
LNG = liquefied natural gas
LPE = low pressure economizer
TET = gas turbine exhaust temperature (°C)
TIT = gas turbine inlet temperature (°C)

Greek letters

β = fuel coefficient
$\Delta\dot{G}_{\mathrm{r}}$ = Gibbs free energy decrease rate (kW)
Δh_{c} = leaving velocity loss ($\mathrm{kJ\,kg^{-1}}$)
η = efficiency
$\eta_{\mathrm{cc,2}}$ = exergy efficiency of combined cycle
$\eta_{\mathrm{cc,fu}}$ = fuel utilization efficiency
η_{EG} = generator efficiency
η_{m} = mechanical efficiency
η_{s} = isentropic gas turbine efficiency
π = pressure ratio

Subscripts

I, II, III = fluid
a = air
cc = combined cycle
comb = combustor
comp = air compressor
cond = condenser
e = exit condition
f = fuel
g = gas
GT = gas turbine
gt = gas turbine cycle
i = inlet condition
j = surface condition
pump = pump
sL = low pressure steam

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ST = steam turbine
st = steam turbine cycle
sH = high pressure steam
0 = reference state

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