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Engineering design and exergy analyses for combustion gas turbine based power generation system

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Abstract

This paper presents the engineering design and theoretical exergetic analyses of the plant for combustion gas turbine based power generation systems. Exergy analysis is performed based on the first and second laws of thermodynamics for power generation systems. The results show the exergy analyses for a steam cycle system predict the plant efficiency more precisely. The plant efficiency for partial load operation is lower than full load operation. Increasing the pinch points will decrease the combined cycle plant efficiency. The engineering design is based on inlet air-cooling and natural gas preheating for increasing the net power output and efficiency. To evaluate the energy utilization, one combined cycle unit and one cogeneration system, consisting of gas turbine generators, heat recovery steam generators, one steam turbine generator with steam extracted for process have been analyzed. The analytical results are used for engineering design and component selection.

Keywords: Exergy; Gas turbine; Inlet air-cooling; Natural gas preheating; Combined cycle

1. Introduction

Studies of engineering designs and exergy analyses for power generation systems are of scientific interest and also essential for the efficient utilization of energy resources. For this reason, the exergy analysis has drawn much attention by scientists and system designers in recent years. Some devoted their studies [1,2] to component exergy analyses [3] and efficiency improvement; others concentrate on systems design and analyses [4–8].

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Nomenclature

B_f	fuel exergy (kJ/s)
B_p	process heat exergy (kJ/s)
ε	exergy (kJ/kg)
ε_f	fuel exergy factor
ε_s	steam/water heat exergy factor
E_f	fuel energy (kJ/s)
H	enthalpy (kJ/s)
h	enthalpy of produced steam/water (kJ/kg)
hc	enthalpy of feed water return to HRSG (kJ/kg)
m_a	mass flow rate of air (kg/s)
m_f	mass flow rate of fuel (kg/s)
m_e	mass flow rate of exhaust flue gas (kg/s)
m_s	mass flow rate of steam/water (kg/s)
m_{sH}	mass flow rate of high-pressure steam (kg/s)
m_{sL}	mass flow rate of low-pressure steam (kg/s)
p	pressure absolute (MPa)
Q_p	process heat $= H_o - H_i$ (kJ/s)
R	power-to-heat ratio
s	entropy (kJ/kg $^\circ$C)
T	temperature ($^\circ$C)
T_o	atmospheric temperature ($^\circ$C)
W_E	electrical gross power output of plant (kW)
W_{GT}	gas turbines gross power output (kW)
W_{ST}	steam turbine gross power output (kW)
η_{CC}	thermal efficiency for combined cycle plant
η_{GT}	thermal efficiency for gas turbine
η_{HRSG}	thermal efficiency for heat recovery steam generator
η_{ST}	thermal efficiency for steam cycle
$\eta_{Rankine}$	thermal efficiency for Rankine cycle
η_{1st}	first law efficiency
η_{2nd}	second law efficiency

Subscript

i	inlet
o	outlet

Abbreviation

CCPP	combined cycle power plant
CP	condensate pump
GTG	gas turbine generator

HP	high pressure
HRSG	heat recovery steam generator
HT	heater
ISO	International Standards Organization
LP	low pressure
LHV	lower heating value
MSL	mean sea level
RH	relative humidity
SC	simple cycle
STG	steam turbine generator

A computerized thermodynamic analyses of a gas turbine based combined cycle power plant (CCPP) fitted with a triple pressure HRSG and hot reheat foresees the efficiency reaching a value of 62% [4]. Huang [5] shows that the performance evaluation of a CCPP based only on the first law of thermodynamics is not adequate, but the second law of thermodynamics must be taken into consideration to get a better evaluation. Verkhivker and Kosoy [6] pointed out the principal processes which cause the destruction of exergy in a power generation cycle are the combustion process, the subsequent heating of the working fluid and the heat transfer in the heat exchangers. For a combined triple (Brayton/Rankine/Rankine)/(gas/steam/ammonia) power cycle, Marrero et al. [7] also confirms that the largest irreversibility is produced in the combustion process. It decreases with the increase of the gas turbine inlet temperature. Parametric studies performed by Bilgen [8] show that the first law and second law efficiencies are decreased with the increase of the power-to-heat ratio. The first law efficiency is strongly related to the power-to-heat ratio in a cogeneration plant. The efficiency is reduced around 40% when the power-to-heat ratio increases from 1 to 20. On the other hand, the second law efficiency is degraded only about 2% when the power-to-heat ratio increases from 1 to 20. It complies with the second law of thermodynamics—work is the valuable commodity of a power plant. Work can be completely and continuously converted to heat. Heat is not completely converted to work in a cycle.

For engineering and system design, gas turbines are the main power producers in the CCPP, and great progress has been made in recent decades to enhance the efficiency of gas turbines. On the other hand, researchers also look for methods to improve the power output and thermal efficiency of already installed gas turbines. System modifications can include cooling the compressor inlet air and preheating of the fuel gas. Such modifications have been applied to some power plants installed in Taiwan [9,10]. Power output and efficiency enhancement methods proposed in the past years [9–12] include mechanical refrigeration or absorption chillers for cooling the inlet air entering the compressor, preheating the fuel gas, and lastly, supplemental firing of the HRSG to increase the steam turbine power output. The related literature [11] shows that CCPP performance is strongly related to the temperature of the ambient air drawn into the gas turbine compressor inlet. Gas turbines typically produce 20% more power when supplied with ambient air at a temperature of 7.2 °C than at 35 °C [12]. Although the benefit of inlet air-cooling for CCPP has been explored in many investigations, we still need to consider the specific climatic data and the gas turbine unit characteristic to assess the benefits for individual power plants. Additionally, several CCPPs in Taiwan are designed with a natural gas (NG) preheating system to improve the efficiency. Flue gas heat energy is recovered to preheat the NG. This energy recovery decreases the flue gas exhaust temperature by only 3 °C, keeping the flue gas higher than its dew point. Therefore, the steam turbine power output will not be affected. However, such an application has seldom been discussed in related studies.

The power generation industry in Taiwan, at the end of year 2001, had the total installed capacity of about 30,203 MW. Statistical data [13] reveal that the capacity provided by CCPP is 21.1% (6367 MW) of this capacity. Also, the CCPP share will continue to expand, reaching an estimated 34.5% (14,294 MW), raising Taiwan's total installed capacity to an estimated 41,428 MW in the coming 10 years. Of the projected increase, around 70% will be CCPP installations. In addition, other predictions indicate that over 50% of new power plants in America will be CCPP installations [14]. Therefore, the power producers are interested in power output and efficiency improvement technologies for these CCPPs.

This study investigates the gas turbine based power generation systems, and the exergy analyses are compared with the CCPP operating data for engineering design improvement. Then, the engineering designs and analyses for compressor inlet air-cooling and fuel gas preheating systems are presented to study the feasibility of the two approaches to gain gas turbine power output and efficiency improvement in operating power plants.

2. Power plants description

To analyze the engineering system design and enhance gas turbine performance, two different power plants were selected to perform the experiments in this study. The first one is combined cycle Plant A; the other one is cogeneration Plant B. Fig. 1 illustrates the schematic flow diagram for Plant A; Fig. 2 shows the schematic flow diagram for Plant B. Plant B is not equipped with NG preheating. The processes for Plants A and B are similar when no steam is extracted for offsite steam users from the Plant B steam turbine. These plants are briefly described in Table 1.

Taiwan is located in a sub-tropical zone; the temperature difference between summer and winter is moderate. However, the power output decreases when operated in the warmer ambient air conditions due to the lower air mass flow rate.

2.1. Gas turbine design information

The gas turbines information of Plants A and B are presented in Table 2.

Normally, the gas turbine manufacturers quote performance based on ISO and LHV conditions. ISO conditions mean 15 °C ambient air temperature, 101.325 kPa barometric pressure, and 60% relative humidity. LHV means the lower heating value of the fuel being used. ASTM D3588 is the method used to determine the lower heating value of the NG fuel supplied to both Plants A and B.

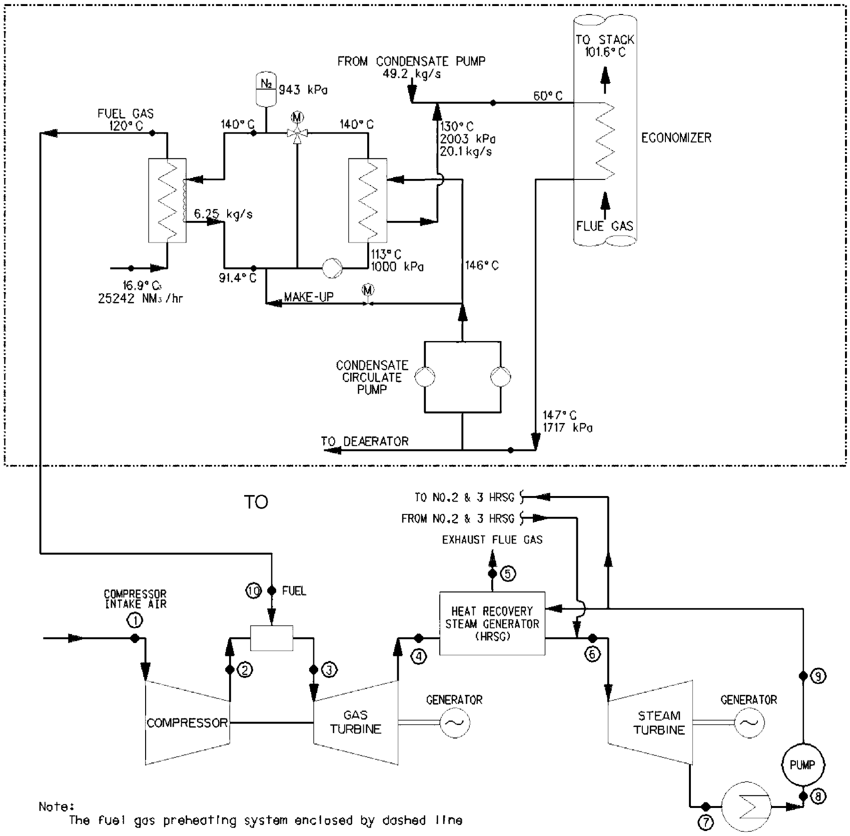

Fig. 1. Schematic flow diagram for Plant A.

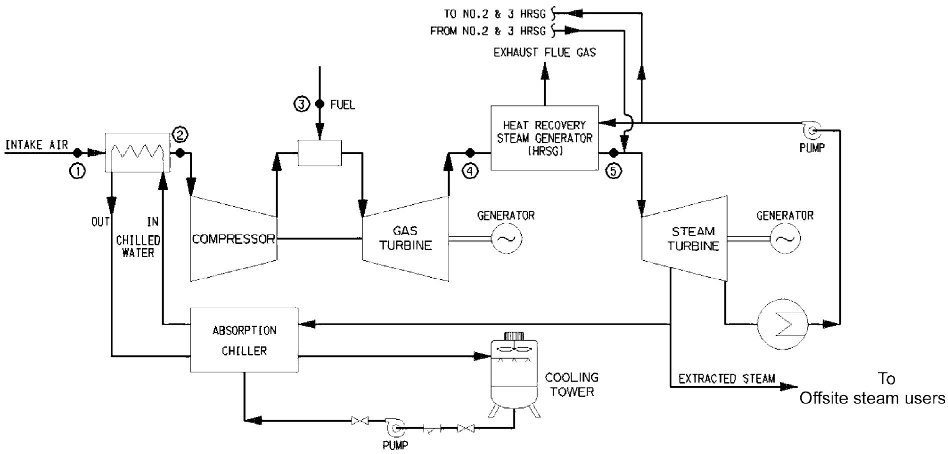

Fig. 2. Schematic flow diagram for Plant B.

Table 1
Description of Plants A and B

Plant type	Plant A	Plant B
	5 combined cycle units	1 cogeneration cycle units
Combination	3 GTGs + 3 HRSGs + 1 STG	3 aero derivative GTGs + 3 HRSGs + 1 STG
Total installed capacity (MW)	2200	150 + 45 ton/h steam
Site elevation (meter above MSL)	4	138
Annual average RH (%)	80	78.5
Highest/lowest daily T (°C)	35.4/7.4	38/6
Average annual T (°C)	25.1	22.3
Design condition (°C and %RH)	32% and 90%	32% and 90%
Preheated fuel gas T (°C)	120	N/A
Air inlet cooling T (°C)	N/A	5.5 + 0.5
Commercially operated date	January 1999	January 2000

Table 2
Gas turbine design information for Plants A and B

Type	Plant A	Plant B
	Heavy duty	Aero derivative
No. of compressor stages per GTG	17	19 (14 HP + 5 LP)
No. of turbine stages per GTG	4	7 (2 HP + 5 LP)
No. of shaft per GTG	1	2
Shaft rotor	Solid	Hollow concentric
Shaft speed (rpm)	3600	HP = 10, 000;LP = 3, 600
No. of combustors	2	1 annular
No. of burners per combustor	6	30
Compression ratio of GT	10.2	29.4

3. Results and discussion

3.1. Exergy analyses of CCPP

A CCPP as shown in Fig. 1 includes both the Brayton cycle and Rankine cycle. It joins operation of the gas turbine at the "hot end" and the steam turbine at the "cold end." The scale diagram of temperature vs. entropy for Plant A is plotted on Fig. 3. Referring to Figs. 1 and 3, gas turbine operates on the Brayton Cycle, i.e., intake air is compressed nearly isentropically from point 1 to 2, combusted at constant pressure from point 2 to 3, and then expanded nearly isentropically in the gas turbine from point 3 to 4, exhausting gas from point 4 to 5. The Brayton cycle as applied to a gas turbine is an open cycle, temperature at point 5 is higher than 100 °C and does not form a closed loop with the inlet air of point 1. For the Rankine cycle, points 9 to 6 shown on Fig. 3 are a heating process at constant pressure (pressure drop is shown

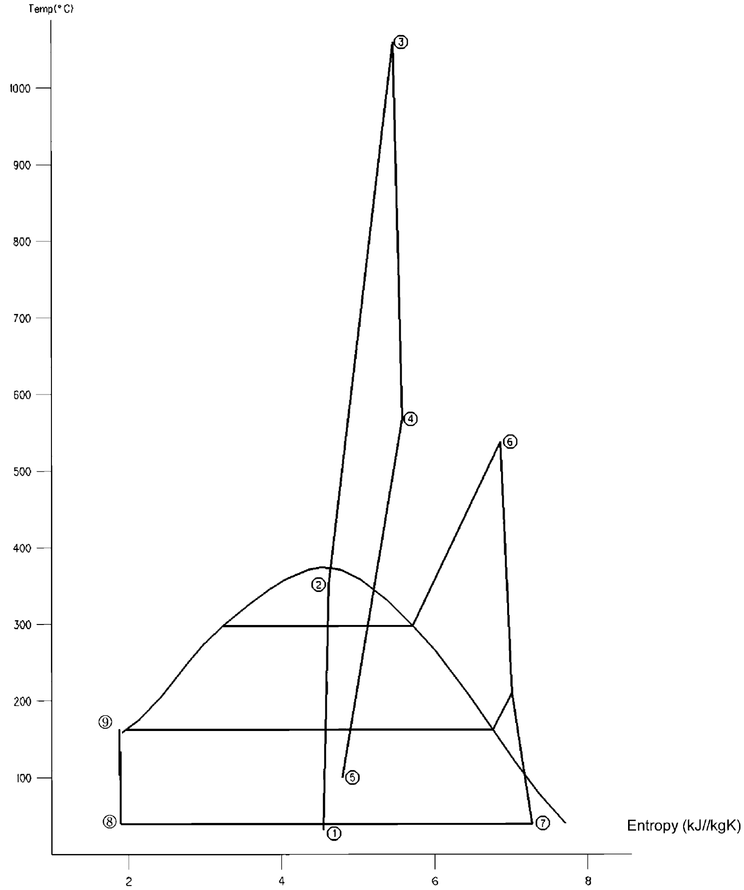

Fig. 3. T–S diagram of combined cycle plant.

on Table 3). The actual turbine expansion from point 6 to 7 is an irreversible (non-isentropic) process. No pressure losses are encountered in the condenser process from point 7 to 8 because it is a two-phase condensation process. The processes of pumps and heaters from point 8 to 9 are nearly isentropic. The working fluid of the Rankine cycle is a vapor–liquid and the Brayton cycle is a gas. The two different working fluids are shown on Fig. 3, which is temperature and entropy (T–S) diagram showing the operating data at each process.

The useful products of a CCPP are electrical energy, $W_{E,}$ and thermal energy, Q_P, in the form of superheated steam. The thermodynamic performance is based on the first law efficiency, and is defined as

$$\eta 1\mathrm{st} = \frac{W_E + Q\mathrm{p}}{E_f} \quad (1)$$

For simple cycle, W_E is equal to W_{GT} and for combined cycle, W_E is equal to $W_{GT} + W_{ST}$. Actually, the efficiency of a CCPP is reduced by the various inherent losses. One unavoidable loss is heat lost by radiation and/or convection, while a second is the internal loss caused by

Table 3
Exergy analyses for steam cycle of Plant A (100% power output)

Equipment			Design conditions									
			M (kg/s)	T (°C)	P (MPa)	h (kJ/kg)	H (kJ/s)	Q_p (kJ/s)	s (kJ/kg.K)	ε (kJ/kg)	B_p (kJ/s)	$\sum B_p$ (kJ/s)
HRSG	HP	i	136.258	158.1	13.025	674.8	91,947	383,839	1.9096	92.37	12,586	−194,170
		o	136.258	538	8.004	3491.8	475,786		6.8418	1405.1	191,456	
	LP	i	23.362	155.9	1.091	658.2	15,377	51,709	1.9011	78.36	1831	
		o	23.362	210.1	0.601	2871.6	67,086		7.0108	733.28	17,131	
STG	HP	i	136.258	536.3	7.604	3491.8	475,786	180,426	6.8643	1398.2	190,516	198,040
	LP	i	20.911	208.8	0.542	2871.7	60,050		7.0573	719.22	15,040	
	ST	o	156.81	40	0.0074	2266.5	355,410		7.2740	47.93	7516	
Con-		i	156.81	40.1	0.0074	2267	355,488	329,131	7.2740	48.43	7594	8699
denser		o	157.169	40.1	0.0074	167.7	26,357		0.5729	−7.03	−1105	
CP		i	157.169	40.1	0.0074	167.7	26,357	267	0.5729	−7.03	−1105	−190
		o	157.169	40.2	1.20	169.4	26,624		0.5745	−5.82	−915	
HT	LP	i	2.451	209.3	0.565	2871.6	7038	71272	7.0386	724.8	1776	−11,535
	Cond	i	157.169	40.2	1.20	169.4	26,624		0.5745	−5.82	−915	
	FW	o	159.62	155.8	0.555	657.4	104,934		1.9008	77.66	12,396	
FP	HP	i	136.258	155.8	0.555	657.4	89,576	2371	1.9008	77.66	10,582	−2021
		o	136.258	158.1	13.025	674.8	91,947		1.9096	92.37	12,586	
	LP	i	23.362	155.8	0.555	657.4	15,358	19	1.9008	77.66	1814	
		o	23.362	155.9	1.091	658.2	15,377		1.9011	78.36	1831	

Note: (1) $T_o = 32\ ^\circ C = 305$ K.
(2) ε is calculated from Eq. (2), B_P—exergy is calculated from Eq. (3) of 2nd law.

irreversible processes as discussed in the second law of thermodynamics. The exergy of the steam/water is defined as:

$$\varepsilon = h - T_o s \tag{2}$$

The exergy of steam/water produced [8] is

$$B_p = h_i - T_o s_i - (h_o - T_o s_o) = (h_i - h_o) - T_o\ (s_i - s_o) \tag{3}$$

Q_p is defined as the energy of steam generated in the HRSG or equipment and is calculated from high-pressure steam $m_{sH}(h - hc)_H$ plus low-pressure steam $m_{sL\ (h-hc)_L}$. The HRSG of Plant A is a pure energy converter transferring heat from the GTG exhaust gas without supplemental fuel combustion, therefore, no combustion takes place in the HRSG. Exergy analyses of Plant A operated steam cycle for 100% load and 50% load are shown on Tables 3 and 4, respectively. Plant A consists of three GTGs and one STG forming a combined cycle unit. When Plant A operates at 50% load, two GTGs and associated HRSGs operate at 75% load and the STG operates in the sliding pressure mode at 50% STG load. GTG control is achieved by variable inlet guide vanes, adjusted to keep the exhaust gas temperature constant at variable GT loads. The exergy evaluation in the system has to consider the mass flow rate of the steam/water. Theoretically in a steam/water cycle, the exergy of the steam/water produced in the HRSG plus power consumption of the pumps should be equal to the steam turbine gross power output plus the steam extracted by the heaters and exhausted to the condenser. However, from the second law of thermodynamics, the $\Sigma_j \Delta s_j \geq 0$ and compared with Eq. (2) then $\Sigma_j \Delta \varepsilon_j \leq 0$. Tables 3 and 4 show the amount of ΣB_p lost in the whole steam/water cycle amounts to less than 0.6% and 1.8%, respectively, which supports the second law of thermodynamics.

The power-to-heat ratio for simple cycle (GT only), $R = \dfrac{W_{GT}}{Q_P}$ (4)

and for combined cycle, $R = \dfrac{W_E}{Q'_P}$ (5)

wherein $Q'_p = Q_p - W_{ST}$. From Table 6, only 32.74% of steam energy can be converted to electrical power, which is the efficiency of Rankine cycle portion of Plant A at 100% load and at design conditions. If the electrical power and steam thermal energy are treated as equivalent energy levels, it is called the law of the energy conservation, part of the first law of thermodynamics. However, electrical power is much more valuable than steam/water thermal energy in a power generating plant. The energy can proceed in a certain direction but not in the reverse direction.

Exergy, the essential concept in second law analysis, is always consumed or destroyed in any process. If less exergy is consumed, a cycle can produce more efficiently. Therefore, by using exergy to evaluate the power plant cycles, a more accurate performance of the system can be obtained

$$\eta_{2nd} = \frac{W_E + B_P}{B_f} \tag{6}$$

Table 4
Exergy analyses for steam cycle of Plant A (50% power output)

Equipment			Design conditions									
			M (kg/s)	T (°C)	P (MPa)	h (kJ/kg)	H (kJ/s)	Q_p (kJ/s)	s (kJ/kg K)	ε (kJ/kg)	B_p (kJ/s)	$\sum B_p$ (kJ/s)
HRSG	HP	i	76.401	159.4	12.575	680.2	51,968	217,216	1.9230	93.69	7158	−103,280
		o	76.401	538	4.8	3523.3	269,184		7.1083	1355.27	103,544	
	LP	i	10.643	157.3	1.0903	663.9	7066	23,328	1.9146	79.95	851	
		o	10.643	202.8	0.6	2855.8	30,394		6.9791	727.71	7745	
STG	HP	i	76.401	535.9	4.3173	3523.3	269,184	97,456	7.1558	1340.78	102,437	106,066
	LP	i	8.597	195.2	0.293	2855.8	24,551		7.3023	628.60	5404	
	ST	o	84.848	35.9	0.0059	2313.3	196,279		7.5160	20.92	1775	
Con-denser		i	84.848	35.9	0.0059	2313.3	196,279	183,504	7.5160	20.92	1775	2400
		o	84.998	35.9	0.0059	150.3	12,775		0.5169	−7.35	−625	
CP		i	84.998	35.9	0.0059	150.3	12,775	128	0.5169	−7.35	−625	−112
		o	84.998	36	1.1342	151.8	12,903		0.5175	−6.04	−513	
HT	LP	i	2.046	202.4	0.5847	2855.8	5843	38,973	6.9908	723.61	1481	−5940
	Cond	i	84.998	36	1.1342	151.8	12,903		0.5175	−6.04	−513	
	FW	o	87.044	157.2	0.5747	663.1	57,719		1.9139	79.36	6908	
FP	HP	i	76.401	157.2	0.5747	663.1	50,662	1306	1.9139	79.36	6063	
		o	76.401	159.4	12.575	680.2	51,968		1.9230	93.69	7158	
	LP	i	10.643	157.2	0.5747	663.1	7057	9	1.9139	79.36	845	−1101
		o	10.643	157.3	1.0903	663.9	7066		1.9146	79.95	851	

Note: (1) T_o= 32 °C = 305 K. (2) ε is calculated from Eq. (2), B_P—exergy is calculated from Eq. (3) of 2nd law.

The exergy factor of generated steam/water (ε_s) and the exergy factor of fuel input (ε_f) can be expressed as follows:

$$\varepsilon_s = \frac{B_P}{Q_P} \tag{7}$$

$$\varepsilon_f = \frac{B_f}{E_f} = 1 \tag{8}$$

[5]. Substituting in Eq. (6)

$$\begin{aligned}\eta_{2nd} &= \frac{W_E + \varepsilon_s Q_P + Q_P - Q_P}{\varepsilon_f E_f} = \frac{W_E + Q_P + (\varepsilon_s - 1) Q_P}{\varepsilon_f E_f} = \frac{\eta_{1st}}{\varepsilon_f}\left[1 + \frac{\varepsilon_f}{\eta_{1st}} h \frac{(\varepsilon_s - 1) Q_P)}{\varepsilon_f E_f}\right] \\ &= \frac{\eta_{1st}}{\varepsilon_f}\left[\frac{W_E + B_P}{\eta_{1st} E_f}\right] = \frac{\eta_{1st}}{\varepsilon_f}\left[\frac{R + \varepsilon_s}{R + 1}\right]\end{aligned} \tag{9}$$

3.1.1. *Brayton cycle theoretical efficiency*

The ideal Brayton cycle is composed of two adiabatic–reversible (isentropic) and two constant pressure processes (as shown in Fig. 3). The theoretical Plant A GTG efficiency (η_{GT}) at ISO conditions with a flue gas inlet temperature of 1060 °C and exhaust temperature of 556.4 °C is 37.78%. This theoretical efficiency is 5.39% higher than the actual gas turbine operating efficiency of 32.39% due to the mechanical losses and non-isentropic processes existing in the operation.

3.1.2. *Rankine cycle theoretical efficiency*

The conditions of the steam generated from the HRSG will affect the steam cycle efficiency. The Rankine cycle of the steam turbine for Plant A utilizes steam as a working fluid and the condenser as a heat rejection reservoir, operating at constant pressure. Assume the turbine expansion and the feed water pumping processes are isentropic. The Rankine cycle efficiency is defined as the turbine work (enthalpy difference between the steam at the turbine inlet and at the turbine exhaust outlet) divided by the inlet steam enthalpy. The ideal Rankine cycle efficiency for Plant A with superheated dual pressure steam conditions is 38.37% at a condenser pressure of 0.0074 MPa.

Three cases of inlet steam conditions are analyzed, two at different saturated steam pressures of 7.604 and 0.542 MPa and one with superheated steam at 7.604 MPa and 536.3 °C to evaluate the Rankine cycle efficiency as shown in Table 5. All three cases supply single pressure steam to the turbine. Using superheated steam (7.604 MPa and 536.3 °C) definitely promotes a higher steam cycle thermal efficiency of 3.41% and 16.2% over using saturated steam at 7.604 and 0.542 MPa pressure, respectively. Similarly, a higher saturated steam at 7.604 MPa pressure has 12.79% better efficiency than the lower pressure saturated steam at 0.542 MPa due to its higher temperature.

Plant A is a dual pressure steam cycle, with the HP steam conditions the same as the superheated steam of the single pressure case and the LP steam flow is 15% of the total steam flow of the cycle. But, the Plant A efficiency of 38.37% is 2.44% lower than the theoretical efficiency when using single pressure superheated steam. However, adding the LP steam features recovers more flue gas energy in the LP stage of the HRSG with less heat loss to the stack. The gain in

Table 5
Comparison of steam conditions vs. Rankine cycle efficiency

	Pressure (MPa)	Temperature (°C)	η (%)
Case 1	7.604	291.5	37.4
Case 2	0.542	208.8	24.61
Case 3	7.604	536.3	40.81

Note: Condenser at 0.0074 MPa and 40 °C conditions.

LP steam energy (Q_P) in HRSG is shown in Table 6. The pinch points limit the steam temperature produced by the HRSG.

3.2. Effect of partial load on CCPP efficiency

Using the first law to evaluate the efficiency of Plant A at 100% load (see Table 3), the total steam energy at the three HRSG's HP and LP superheater outlets is 542,872 kJ/s; the net energy output from steam turbine is 180,426 kJ/s. The calculated first law efficiency (η) of steam cycle is 33.24%. Using the second law, the total outlet exergy of HP and LP steam from the HRSG is 208,587 kJ/s, the inlet exergy to steam turbine and heaters is 205,556 and 861 kJ/s, respectively. The destroyed exergy from HRSG, heaters and pumps to steam turbine and condenser is 2170 kJ/s (1.04%) due to the piping insulation thermal loss. The destroyed exergy at the condenser is 8699 kJ/s (4.17%). The total destroyed exergy in the steam cycle is 10,869 kJ/s (5.21%). The exergy analyses of Plant A operating at 50% load is shown in Table 4 and the component exergy analyses is shown in Table 6.

From Table 6, the overall efficiency of Plant A at 100% power output of CCPP is 53.78%. The Plant A actual efficiency (η_{cc}) is 51.27% [449,972 kW/(877,605 kJ/s)] for 100% power output from the operation record. For the 50% power output of the CCPP, the Plant A overall efficiency is 51.35% and the actual operation efficiency is 48.87%. The efficiency at ISO conditions is 54.32% and actual operation efficiency is 51.12%. The evaluated efficiency is 2.5–3.2% higher than the actual operation efficiency since the stack energy and exergy losses are not considered. The Plant A performance have been verified by the acceptance tests in 1999, which revealed that the efficiency and power output have 0.9% and 1.06% better than the original vendor guarantee, respectively.

The exergy loss at 50% load is three times that of 100% load due to the lower steam pressure in the HRSG. It can be verified from the steam T–S diagram, as the LP steam has higher entropy value than HP steam at the same temperature. Therefore, the Plant A operating efficiency at 100% load is 2.4% higher than at 50% load.

3.2.1. Thermodynamics analyses

The first law efficiency of simple cycle $\eta_{1st} = 80.91\%$ is calculated by Eq. (1), where the W_{GT}, Q_p and E_f are shown in Table 6, item 1. The second law efficiency of simple cycle $\eta_{2nd} = 53.4\%$, wherein the heat to power ratio (R) is equal to 0.63 which is calculated by Eq. (4) and $\varepsilon_s = 0.4458$ which is calculated by Eq. (7). B_p is listed in Table 3.

Table 6
Exergy analyses of main equipment for Plant A

Load (%)	m (kg/s)		h (kJ/kg)		E_f (kJ/s)	W_{GT} (kW)	η_{GT} (%)	Exhaust gas			Q_p (kJ/s)	η_{1st} (%)
	Fuel	Air	Fuel	Air				m_e (kg/s)	H (kJ/kg)	Energy (kJ/s)		
1. *Gas turbine (one set)*												
100[a]	6.276	342.85	50,083	31.2	314,321	101,810	32.39	349.1	618.8	216,023	147,022	79.16
100[b]	5.841	321.66	50,083	102.8	292,535	91,513	31.28	327.5	631.8	206,915	145,190	80.91
50[c]	4.599	264.130	50,083	102.8	230,332	65,681	28.52	268.73	631.8	169,784	120,267	80.73

Load (%)	Exhaust gas					HP steam				LP steam			
	m_e (kg/s)	h_i (kJ/kg)	h_o (kJ/kg)	Energy (kJ/s)	m_s (kg/s)	h_o (kJ/kg)	h_i (kJ/kg)	Q_p (kJ/s)	m_s (kg/s)	h_o (kJ/kg)	h_i (kJ/kg)	Q_p (kJ/s)	η_{HRSG} (%)
2. *Heat recovery steam generator (one set)*													
100[a]	349.1	618.8	105.9	179,053	46.11	3466.5	679.5	128,509	8.37	2875.2	663.4	18,513	82.1
100[b]	327.5	631.8	105.9	172,232	45.42	3491.8	674.8	127,948	7.79	2871.6	658.2	17,242	84.3
50[c]	268.73	631.8	105.9	141,325	38.2	3523.3	680.2	108,606	5.32	2855.8	663.9	11,661	85.1

Load (%)	HP steam H_i (kJ/s)	LP steam H_i (kJ/s)	$\sum H_i$ (kJ/s)	W_{ST} (kW)	$\eta_{Rankine}$ (%)
3. *Steam turbine set (combined by 3 HRSGs)*					
100[a]	479,524	64,977	544,501	176,599	32.43
100[b]	475,786	60,050	535,836	175,433	32.74
50[c]	269,184	24,551	293,735	93,810	31.94

[a] 100% load at ISO condition.
[b] 100% load at 32 °C ambient temperature.
[c] 50% load is 2 GTGs operated at 32 °C ambient temperature.

The first law efficiency of combined cycle is same as simple cycle; however, the steam turbine power output needs to be deducted from Q_p to obtain Q'_p, which is equal to 260,115 kJ/s. The second law efficiency of combined cycle $\eta_{2nd} = 73.4\%$, wherein R is equal to $W_E/Q'_p = 1.73$ and $\varepsilon_s = B_p/Q'_p = 0.7465$.

The exhaust gas leaves the gas turbine through a horizontal outlet. Then the gas enters a 90° upward bend at the base of the vertical tower section, which contains the HRSG heating surfaces. The HRSG heating surfaces arranged in the direction of gas flow are HP superheater, HP evaporator, HP economizer, LP superheater, LP evaporator, and a common condensate preheater, forming a dual pressure boiler. The three main parts producing HP steam are firstly, the HP economizer, secondly, the HP evaporator, and the lastly, HP superheater, all working at a HP steam pressure. The condensate at 60 °C enters the common condensate preheater and exits (at 147 °C) to the feed water tank. It is heated to saturation and deaerated in the feed water tank. Then, the low-pressure feed pump takes some deaerated water from the feed tank and discharges this deaerated feed to the LP evaporator T_{fL} at 162.5 °C and the feed water is evaporated. The vapor after being separated from the unevaporated water in the LP steam drum passes to the LP superheater. Finally, the vapor passes through the LP superheater and exits at a superheated temperature of $T = 210$ °C. The HP feedwater pump takes the balance of the deaerated water from the feed tank and discharges it into the HP economizer. The feedwater temperature at HP economizer inlet is T_{cH} (156 °C), a sub-cooled liquid state. The sub-cooled liquid is heated in the economizer and exits at T_{fH} (298 °C) to the HP evaporator, where it is changed to the saturated vapor state. The vapor is separated from the liquid in the HP drum and the saturated vapor enters the HP superheater exiting at a superheated temperature of 537.3 °C.

The combustion gas, in counter flow, enters the HP superheater at the gas turbine exit condition of T_t (568 °C), goes through the HP superheater and then the HP evaporator where it exits at a temperature of T_{ppH} (304 °C). The flue gas enters the common condensate preheater at T_{ppL} and exit at T_e (100 °C). The gas and steam temperatures profile of the HRSG are shown in Fig. 4.

For simplicity, it is assumed that the pressure drop in the flue gas side of the HRSG is negligible and the HRSG is well insulated. Energy balance of the flue gas in the single pressure HRSG consisting of a superheater and evaporator becomes

$$m_s(h - h_f) = m_a(1 + r_{fa})(h_t - h_{pp}) \tag{10}$$

where m_a is the mass of air in gas turbine, r_{fa} is the fuel–air ratio used in combustion process ($r_{fa} = 0.01816$ for Plant A), h_t is the enthalpy of flue gas at the turbine exit, h_{pp} is the enthalpy of flue gas leaving the evaporator.

In a dual pressure system, there is a substantial increase in the amount of heat recovered in the HRSG. Usually, the HP steam can be optimized at a higher pressure than the single pressure cycle; more of the energy is transferred into exergy. In addition, LP steam is produced, recovering the heat at the low temperature end of the HRSG and lowering the stack gas temperature by the addition of economizer and preheater.

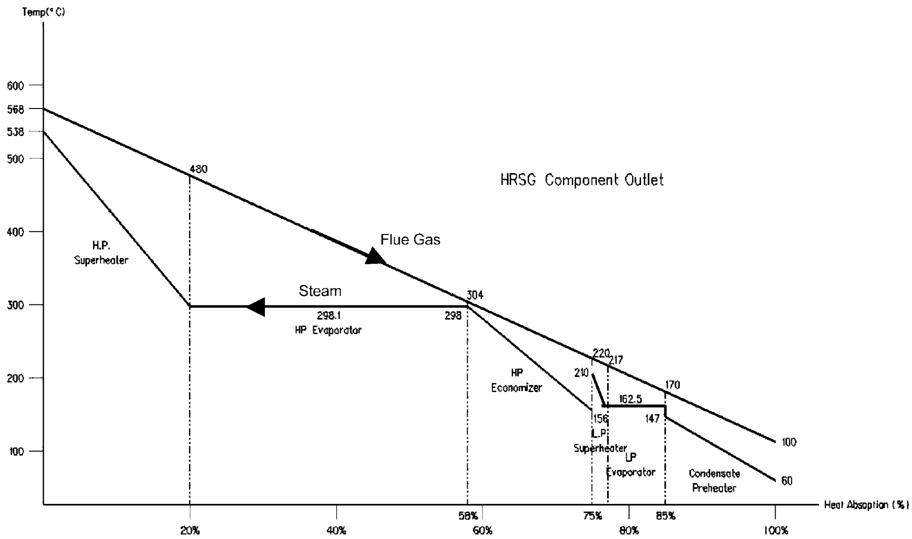

Fig. 4. Gas and steam temperature profile of HRSG.

For a dual pressure level HRSG producing HP and LP steam, Eq. (10) can be developed as

$$m_{sH}(h - h_f)_H = m_a(1 + r_{fa})(h_{tH} - h_{ppH}) \tag{11}$$

$$m_{sL}(h - h_f)_L = m_a(1 + r_{fa})(h_{tL} - h_{ppL}) \tag{12}$$

From Fig. 4, for the HP portion, h is the steam enthalpy at superheater outlet (538 °C), h_f is the feedwater enthalpy at economizer inlet (158.1 °C). For LP portion, h is the steam enthalpy at superheater outlet (210 °C), h_f is the feedwater enthalpy at evaporator inlet (155.9 °C).

Solving the steam generated in HRSG from equation $Q_p = m_{sH}\ (h - hc)_H\ + m_{sL}(h - hc)_L$ for m_{sH} and m_{sL} and inserting it into Eqs. (11) and (12), and rearranging, the steam/water produced per unit mass of air flow is found as

$$Q\text{pH} = m_a(1 + r_{fa})(h_{tH} - h_{ppH})[(h - hc)_H/(h - h_f)]_H \tag{13}$$

$$Q_{pL} = m_a(1 + r_{fa})(h_{tL} - h_{ppL})[(h - hc)_L/(h - h_f)_L] \tag{14}$$

$$\begin{aligned} Q_P = m_a(1 + r_{fa})\{&(h_{th} - h_{ppH})[(h - hc)_H/(h - h_f)_H] \\ &+ (h_{tL} - h_{PPL})[(h - hc)_L/(h - h_f)_L]\} \end{aligned} \tag{15}$$

In Eq. (15), Q_P is the steam/water heat. The efficiency analyses is listed in Fig. 5.

Fig. 5 is based on the temperature of the flue gas, which at pinch point (T_{PPH}) of HP portion is 304 °C and (T_{PPL}) of LP portion is 170 °C. The pinch point is fixed at 6 °C of HP portion and 7.5 °C of LP portion for Plant A. The actual second law efficiency of the simple cycle is lower than the combined cycle because the power-to-heat ratio (R) is only 0.63 instead of 1.73. The higher exergy ratio of generated steam/water and the power-to-heat ratio both increase the thermal efficiency. Fig. 5 depicts the efficiency evaluation based on first law without considering the exergy of generated steam/water. Bilgen [8] evaluated the exergetic and engineering analyses

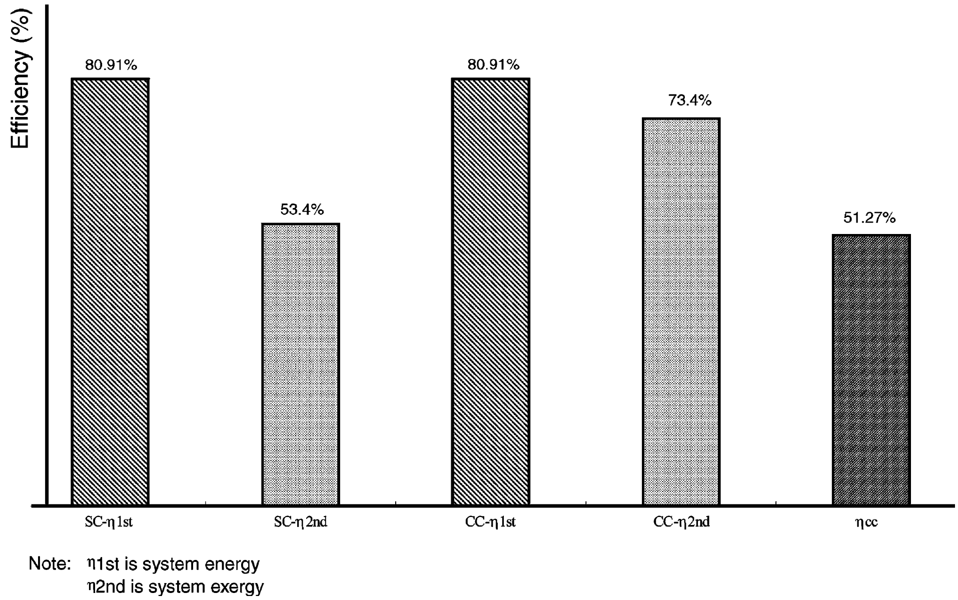

Fig. 5. Thermodynamic analyses of Plant A.

for saturated single pressure steam of a cogeneration system. However, Plant A is a combined cycle installation generating superheated steam at dual pressure conditions. This study analyses the actual operation conditions of this steam cycle. The analytical data are compared to the plant performance test results. Deviations can be studied and understood to allow improvements to be made during the engineering design stage.

3.2.2. Evaluation of temperature and heat absorption for HRSG

The temperature vs. heat absorption diagram (Fig. 4) shows the transferred heat flow Q (horizontal axis), as the flue gas passes through the HRSG in percent of heat absorbed and the vertical scale is temperature. The key parameters Q (heat flow) and C_p (specific heat) are nearly constant over the entire passage through the HRSG resulting in a straight line representation of the flue gas temperature. The temperature drop over each heating surface is therefore proportional to the heat absorbed by the steam/water system. The highest quantity of heat being transferred is in the HP evaporator and the smallest is in the LP superheater. The quantity of heat transferred in the LP superheater appears with a very short and narrow space.

In designing the HRSG, the incoming condensate is first preheated in the common condensate preheater before entering the feedwater tank. The feedwater tank is designed to operate as a deaerator. To achieve proper deaeration in the feedwater tank, a temperature difference of approximately 8 °C is required between the preheated condensate entering the feedwater tank and the feedwater tank temperature itself. Both the LP and HP feedwater pumps take suction from the feedwater tank and supply feedwater at essentially feedwater tank temperature to the LP drum/evaporator and to the HP economizer and drum/evaporator, respectively.

The inlet temperature of the feedwater entering the HP economizer is nearly the same as the feedwater entering the LP drum, as both are fed from the common feedwater tank via separate feedwater pumps. Due to the different pressure levels, the HP feedwater will be heated slightly more by the feedwater pump than the LP feedwater.

3.3. Effect of pinch point on plant performance

The pinch point is the minimum difference between the gas temperature leaving the evaporator section of the HRSG and the saturation temperature corresponding to the steam pressure in the evaporator section. Generally, lowering the pinch point results in an increase of total heat recovered in the evaporator section. However, lowering the pinch point will decrease the logarithmic mean temperature difference (LMTD) in the HRSG, requiring more heat transfer surface area. This significantly increases the equipment cost.

The pinch point for the HP portion is $\Delta T_H = 6\ ^\circ C$ (T_{ppH}–$T_{fH} = 304 – 298\ ^\circ C$) and for the LP portion is $\Delta T_L = 7.5\ ^\circ C$ ($T_{ppL} - T_{fL} = 170 – 162.5\ ^\circ C$). These pinch points are important parameters affecting the thermal performance of the system.

If the pinch point of an HRSG is changed to 10, 20 or 30 °C and the pressure of process steam/water is fixed, the total required heating surface of HRSG will be reduced, leading to a reduction in steam temperature. This change will affect the steam cycle thermal performance. The theoretical energy and exergy efficiencies are shown in Table 7.

Table 7 shows both the HP and LP pinch points increasing from 10 to 30 °C. The energy efficiency of first law and the exergy efficiency of second law for the combined cycle will decrease 1.07% and 0.31%, respectively. The results, compared to Huang's [5] analysis of 0.75% for first law and 0.5% for second law, have minor differences as Huang's [5] analyses are based on single pressure with saturated steam, whereas this study is based on dual pressure with superheated steam. This result will be used as the basis for the optimal heating surface design of HRSG. When the plant is operated at the same GTG power output, increasing the pinch point will cause the power-to-heat ratio to increase, thereby decreasing the efficiency. By the way, the pinch point increase will also increase the exergy loss of steam/water and slightly decrease the efficiency. The first law efficiency decrease is 2.7–3.3 times of the efficiency decreases for the second law at the same power-to-heat ratio. It is consistent with Bilgen [8] study results.

Table 7
Comparisons of pinch point change and efficiency variation

$\Delta T_H/\Delta T_L$	η (%) Combined cycle			
	η_{1st}	η_{2nd}	ε_s	R
6/7.5 (°C)	80.98	77.52	0.8834	1.7257
10/10 (°C)	80.56	77.40	0.8922	1.7059
20/20 (°C)	79.49	77.09	0.9151	1.8170
30/30 (°C)	78.49	76.80	0.9378	1.8841

3.4. Effect of fuel gas preheating on turbine performance

In order to have more insight into the fuel gas preheating effect on turbine efficiency, a series of experiment were conducted on Plant A. The fuel gas is heated to various temperatures, from ambient temperature to 118 °C, while keeping the other parameters constant. The flue gas exit temperature at the stack without the fuel gas preheating system is 104.5 °C. When heating the fuel gas with the flue gas, the exit temperature at stack decreases by 3 °C. The experimental data shown in Fig. 6 depict the average fuel consumption rate (kg/MW·h) at the constant intake air temperature of 26.8 °C is 211.37 kg/MW·h when the fuel gas temperature is 22.5 °C. If the fuel gas is heated to 70.2 and 118 °C, the average fuel consumption rate reduces to 210.98 and 210.37 kg/MW·h, respectively. The fuel consumption rate is reduced 0.4% when the fuel gas is preheated from 22.5 to 118 °C. Preheating the fuel to 118 °C is equivalent to increasing the fuel LHV by 170.96 kJ/m^3. This estimated result is based on the assumption that the composition of fuel gas is 100% CH_4. This implies that the fuel LHV can be treated as 51,146.7 kJ/kg before the combustion process. In general, the power demands during summer require the gas turbine to operate at its full load (nearly 90 MW). For a 24 h operating period, the fuel gas consumption will be reduced by 502.2 kg when the fuel gas is heated to 70.2 °C and reduced by 1830.6 kg when the fuel gas is heated to 118 °C. The cost for fuel gas is around US\$ 8.65/10^6 kJ(LHV), implying by installing the gas preheating facilities, the operating expense will be reduced by US\$ 221.10 per GTG per day when the fuel gas is heated to 70.2 °C. For operating

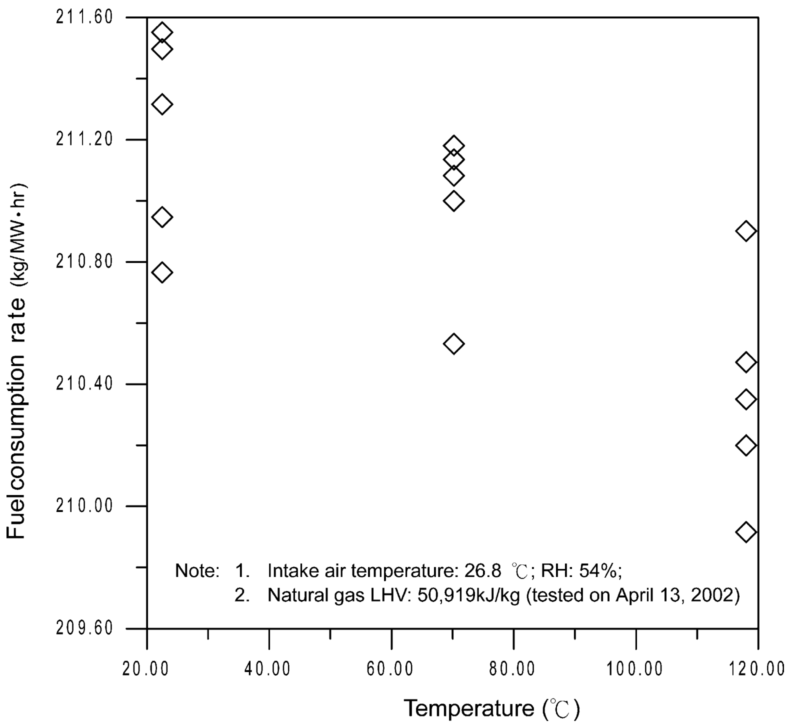

Fig. 6. Relationship between NG temperature and fuel consumption rate.

with the fuel gas heated to 118 °C, the savings in operating cost is US$ 805.90 per GTG per day. The Plant A currently contains 15 sets of gas turbine with 40% capacity factor and 25 years design operation life. The total cost saving for this plant is US$ 44.1 million when the fuel gas is heated to 118 °C during the operating period.

3.5. *Effect of inlet air-cooling on gas turbine performance*

A previous study [1] shows that the location of the power station plays an important role on its performance. The power output of a gas turbine increases as the inlet air temperature decreases. Plant B uses a mechanical refrigerating system to cool the ambient air from 30 °C to 5.5 ± 0.5 °C.

Fig. 7 presents the gas turbine performance data for Plant B. The ambient temperature range of these measurements is from 15 to 30 °C with a relative humidity of $66 \pm 2\%$. It reveals that the inlet air temperature effect on power output is not significant for this turbine unit. It is because the centrifugal water chillers are installed to cool the inlet air to 5.5 ± 0.5 °Cwith $96 \pm 2\%$ RH. The extra power consumption of centrifugal water chillers is only 17.6% of the total power augmentation gained [9]. The gas turbines performance has been verified during the acceptance tests, the power output decreased 18% when the inlet air temperature increased from 15 to 30 °C without installing the intake air-cooling system. Gas turbine output can be maintained nearly constant by controlling the inlet air for combustion to a specific condition. By installing the inlet air-cooling system, it is possible to provide maximum electrical power during hot summer days, when the demand on the Taiwan power grid is the highest.

Currently, Plant B has installed two sets of 1500 refrigeration ton (RT) absorption chillers replacing part of the existing centrifugal water chillers (each with 1000 RT) in 2001. The

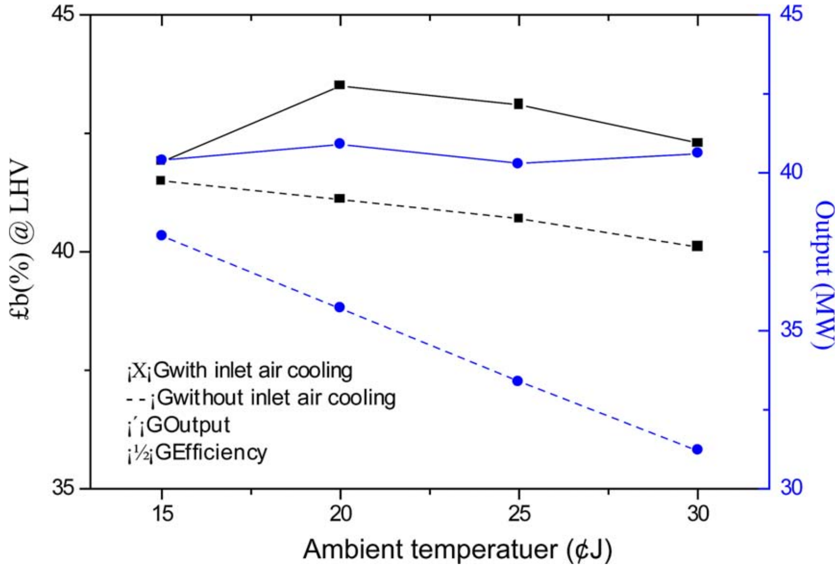

Fig. 7. Plant B gas turbine performance chart.

operation records show that the auxiliary power consumption is reduced about 2215 kW due to the introduction of two (2) sets of 1500 RT absorption chillers. For the sizing of the new chillers, another issue is to select an optimum inlet air temperature to gain the maximum power output without additional power consumption. If we provide the chilled water instead of the process steam to the offsite steam users, many problems with steam supply (traps and expansion) and condensate recovery (contamination, water hammer and flashing water) will be avoided. It is relatively cheap to pump cold water (no user tanks, or pumps, only flow controls at user's end to regulate room temperature).

Fig. 8 displays the measured turbine net power output and other data at two compressor intake air temperatures. The ambient temperature is $14 \pm 2^{\circ}C$ with the average relative humidity of 60% during the measurement period. For Case 1, the inlet air to the compressor is cooled to 5 °C, and cooled to 10 °C for Case 2. The turbine exhaust temperature, 458 ± 3 °C is same for both cases. Meeting the operational requirements of nearly constant turbine exhaust temperature and low NO_x emission, results in more fuel being consumed in the combustor. The power output also is found greater in Case 1 than that in Case 2. However, the gas turbine efficiency of Case 2 is 1% higher than that of Case 1. That is to say, power can be gained by inlet air-cooling. However, for this type gas turbine, the efficiency decreases when the inlet air is cooled to 5 °C.

From Table 6, Plant A operation record at 100% load shows that the gross efficiency of the Brayton cycle is 32.39% at ISO conditions and 31.28% with 32 °C ambient air temperature. Even though the power output of a GTG at ISO conditions was only 10.11% (10,297 kW) higher than the 32 °C (design) conditions, the high exhaust temperature at the GTG exit decreases the gas turbine power output factor. In a CCPP, higher exhaust temperature at the GTG outlet can generate more steam energy for the steam cycle. The higher the efficiency of the GTG, the greater will be the reduction in efficiency of the steam cycle. After installing the steam turbine to utilize the waste heat from the gas turbine exhaust gases, the total combined gross efficiency increased to 51.12% at ISO conditions and 51.27% at 32 °C (design) conditions. The reason of ISO efficiency even slightly lower than the Plant A designed temperature is to ensure

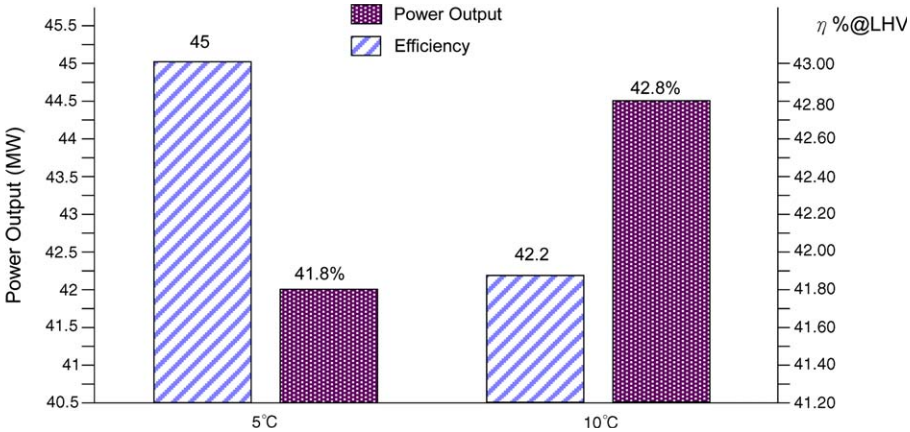

Fig. 8. Plant B GT power output and efficiency at various intake air temperatures.

that Plant A has a better performance during summer operation. It is an optimum engineering design for CCPP in Taiwan.

Ondryas et al. [11] indicated gas turbine output could be increased by 0.72% with each 1 °C inlet air temperature reduction for the Model 7E gas turbines. Analytical data reveal that the selected gas turbine can gain 1.12% output with each 1 °C inlet air temperature reduction in the ambient temperature range between 15 and 40 °C. This difference is mainly attributed to the different gas turbines in original design including the blade geometry and the combustion features and is not discussed herein. The unit capital cost increment is US$ 230/kW [9] for inlet air-cooling system, which is less than half of the unit cost of power generating equipment installation.

3.6. Effect of exergy efficiency with variation of the compressor inlet and of the fuel temperatures

Fig. 7 shows that an ambient temperature change from 15 to 30 °C decreases the efficiency by 1.5%. With the increase in ambient temperature, the airflow mass and power density are both decreased. The power output is decreased 18% with only a 6% decrease in fuel consumption. When the gas turbine operates at varying ambient temperatures, the exergy loss at a lower ambient temperature is less than at a higher ambient temperature as indicated by Eq. (2). The gas turbine is a volume displacement system operating at a constant speed and without inlet control vanes. The inlet air is induced into the compressor at a constant volume during operation. Ambient air density varies inversely with its temperature, directly affecting the inlet air mass flow rate and thus directly impacting the power output [9]. Therefore, by cooling the inlet air, the inlet air mass flow is increased, enabling the gas turbine to operate at higher power output and higher exergy efficiency as in Plant B operating condition.

Recovering some energy from the exhaust flue gas before it enters the stack to preheat the fuel returns otherwise wasted energy back into the cycle. The fuel gas, after conditioning at the facility, averages to 118 °C. Any increase in temperature, prior to entering the combustion process, reduces the amount of fuel needed to bring the incoming fuel up to the combustion temperature. Transferring heat from the stack gases to preheat the incoming fuel results in a direct fuel saving. Lowering the temperature at the exhaust stack reduces the overall energy loss, which enhances the system performance proportionally.

4. Conclusions

Based on first law (energy) and second law (exergy) analyses, the formulas for dual pressure HRSG have been developed for thermodynamic performance and engineering betterment of combustion gas turbine based power generation systems. The plant performance can be improved by improving the system design such as using dual pressure steam cycles, inlet air-cooling for the gas turbine, and fuel gas preheating, etc. From Fig. 5, the efficiency difference of CCPP between second law and actual plant operation at design condition is 22.13%, which is caused by a steam cycle efficiency of only 32.74% as shown in Table 6. The efficiency deviation between exergy analyses and performance test results can be improved by engineering design. The higher exergy destruction occurs because the lower pressure steam has high entropy value at same temperature level. Therefore, the CCPP operated at 50% load has an efficiency of 2.4%

lower than the 100% load. The HRSG pinch point increased 10 °C only affects the overall combined cycle efficiency by 0.3%. The pinch point design value has to be carefully evaluated based on anticipated operating factors to obtain an optimum design.

Experiments conducted on the 90 MW rated gas turbines of Plant A, show that, with the installation of a fuel gas preheating system from 22.5 to 118 °C, less fuel consumption can be achieved under the same power output. Operating records indicate fuel savings and a slight improvement in efficiency (0.06%) for the whole plant.

The gas turbine Plant B produces 18% greater power output using 15 °C intake air than 30 °C air. In many power purchase agreements (PPA), the capacity charge is based on the peak output that can be demonstrated on a predetermined hot summer day. In such cases, the power output reduction in summer is economically unacceptable to the power producer. On the other hand, the plant operational data depict that the efficiency will be 1% lower when compressor inlet air is cooled from 10 to 5 °C. The trend of energy efficiency is not proportional to the power output. Therefore, the optimum operational inlet air temperature to the compressor will depend on the gas turbine design under the consideration of energy saving and contractual requirements.

From the engineering design point of view, if the offsite steam user uses process steam exclusively for absorption chillers associated with air conditioning, the power facility can install the chillers and directly provide the chilled water instead of the process steam. It will eliminate many problems such as steam traps, expansion joints, and return condensate contamination, etc.

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