This is the Pre-Published Version.
https://doi.org/10.1016/j.applthermaleng.2015.10.078

Study on Energy and Economic Benefits of Converting a Combined Heating and Power System to a Tri-generation System for Sewage Treatment Plants in Subtropical Area

Yi CHEN[a], Tiantian ZHANG[b], Hongxing YANG[a,*] and Jinqing PENG[a],

[a]*Renewable Energy Research Group (RERG), Department of Building Services Engineering, The Hong Kong Polytechnic University, Hong Kong*
[b]*School of Municipal and Environmental Engineerimg, Harbin Institute of Technology, Harbin, PR China*

Abstract

A feasibility study was conducted on converting an existing combined heat and power (CHP) system into a tri-generation system for sewage treatment plant application. Usually, a biogas-driven CHP is used for both electricity generation and digester heating. However, a huge amount of heat has to be released during the summer in subtropical areas when the heating demand is low. So the tri-generation system, created by enhancing the CHP scheme with an absorption chiller, was proposed to address the defect. In summer, the huge amount of waste heat drives the chiller and produces chilled water for space cooling. Four possible tri-generation retrofitting schemes were proposed with different types of absorption chiller. The hourly dynamic energy performances of the four systems were simulated by the established mathematic models under Hong Kong weather data. Then detail economic analysis was conducted to the four systems. The results show that the double effect absorption chiller driven by bypass of 450°C flue gas is the optimal retrofitting scheme for its highest thermal efficiency, shortest payback period, better stability and easier control. The annual average thermal efficiency can be improved from 20.8% to 38.3%.

* Corresponding author. Tel.: +852 2766 5816; fax: +852 2765 7198
E-mail address:behxyang@polyu.edu.hk (H. Yang)

1. Introduction

In Hong Kong, over 98% of the sewage produced is collected and treated before being drained into the sea to minimize pollution [1]. In the sewage treatment works, a mixture of thickened primary effluent and activated sludge is produced in the sewage treatment process. This mixture needs to be broken down by biochemical reaction, which takes place in the digesters which are fully enclosed and insulated at 35℃ in anaerobic conditions. The output is a stable and inoffensive sludge which is then transported to landfill for disposal. In the digestion process, biogas consisting mostly of methane (CH4) is produced.

In order to make full use of the biogas and improve the efficiency of energy utilization, combined heat and power (CHP) system is usually implemented in large sewage treatment plants [2~3], providing effective solution for generating electricity by burning the biogas and producing hot water as by-product [4]. The hot water produced by CHP is used to maintain the required temperature of the sludge digester. However, one major problem of this kind of CHP system can be detected in subtropical areas: a large amount of heat has to be released to the environment through the radiators due to the low heating demand of digesters in summer, resulting in significant energy waste. Besides, the power generating efficiency may decrease in summer because the inlet air temperature of the CHP is higher than the designed working temperature. It was reported that power output de-rating can be determined for each project if the intake air temperature > 30℃ for Jenbacher gas internal combustion engine [5]. For the gas turbine engine, the influence by high inlet air temperature is even larger [6~8].

To solve the above problems, a tri-generation system is proposed as a retrofitting scheme in this paper to enhance the overall performance of the original CHP system. Tri-generation system, which is also named combined cooling, heating and power (CCHP) system, mainly comprises a CHP module for generating electricity and heat, and a thermally driven chiller for producing cooling [9~11]. The benefits of the system including energy saving and emission reduction have been studied for different climate zones and proved to be attractive [12]. In the proposed tri-generation system, the chilled water produced by the absorption chiller can be used for two purposes: (1) district cooling for nearby office buildings or residential buildings; (2) cooling the power house to maintain a standard working condition for the CHP unit, especially in summer. In this way, the huge amount of thermal energy from the original CHP system can be reused by regulating the cooling and heating in different seasons. In summer, more chilled water can be produced for air-conditioning, while less hot water is supplied to the digester. In winter, digester heating is given priority and the surplus waste heat is used for cooling. In sum, the naturally produced biogas and the simultaneous heating (digester heating) and cooling (air-conditioning) demands provide an opportunity for tri-generation application in sewage treatment plants.

Actually, the biomass fuelled tri-generation system has attracted increasing attention in recent years because it provides advantages such as primary energy savings and emissions reduction [13~15]. The energy performance, thermodynamic feasibility and thermo-economic potential of the biomass fuelled tri-generation system have been widely investigated based on theories and simulation. Huang present the work focuses on the modeling and simulation of a commercial building scale biomass fuelled tri-generation plant by using ECLIPSE package [16]. Lian evaluated the thermos-economic potential of four biomass tri-generation configuration using the second law of exergy [17]. A biomass tri-generation system using an organic Rankine cycle (ORC) was also investigated by exergy analysis considering four cases [18] as well as by ECLIPSE simulation [19]. Although tri-generation system has been widely investigated, the studies focused on the performances and benefits compared with traditional energy generation systems. Little research has been reported on converting an existing CHP system into a tri-generation system to solve the huge energy waste in summer in subtropical areas, as well as compare the energy performance and economic benefit of different types of tri-generation solutions.

The paper reports a feasibility study of converting an existing CHP system to tri-generation system at a sewage treatment plant. Firstly, the problems of energy utilization by CHP system were identified and analyzed in relation to sewage treatment works in subtropical regions. Four possible tri-generation retrofitting schemes were proposed to maximize the energy efficiency. The annual dynamic performances of the four systems were simulated under Hong Kong weather condition using the established mathematical models. The economic analysis was conducted among the four systems before the optimal one was selected considering the efficiency, stability, automatic control and payback period.

2. Problems of existing CHP system

The schematic diagram of the existing CHP system is shown in Fig.1 and the typical CHP model parameters provided by the GE Energy Co. are listed in Table 1 [20]. The CHP generator is a reciprocating internal combustion engine which is driven by the biogas produced in the digester to generate electricity. The jacket water circulates in the CHP system so as to, on the one hand, cools the generator to maintain it normal working state; and on the other hand, provide hot water as a byproduct.

For internal combustion engine, the typical temperature ranges of the flue gas and hot jacket water are 440 to 500℃ and 70℃ to 90℃, respectively [21]. For this existing CHP system, the temperatures of flue gas and hot jacket water from the generator are 450℃ and 79℃, respectively. As the temperature of the exhaust gas from generator is very high, a heat exchanger (HE1) was added to recover the heat from the exhaust gas and to further reheat the jacket water (primary water) to 85.9℃. In heat exchanger 2 (HE2), the hot jacket water first releases heat to the digester's water heating system (secondary water loop) to meet its heat demand, and then passes through a radiator to release excess heat if its temperature is higher than 70°C. If the thermal energy can't meet the heating load in extremely cold days, the backup electricity-driven boiler is put into operation for heating the digester.

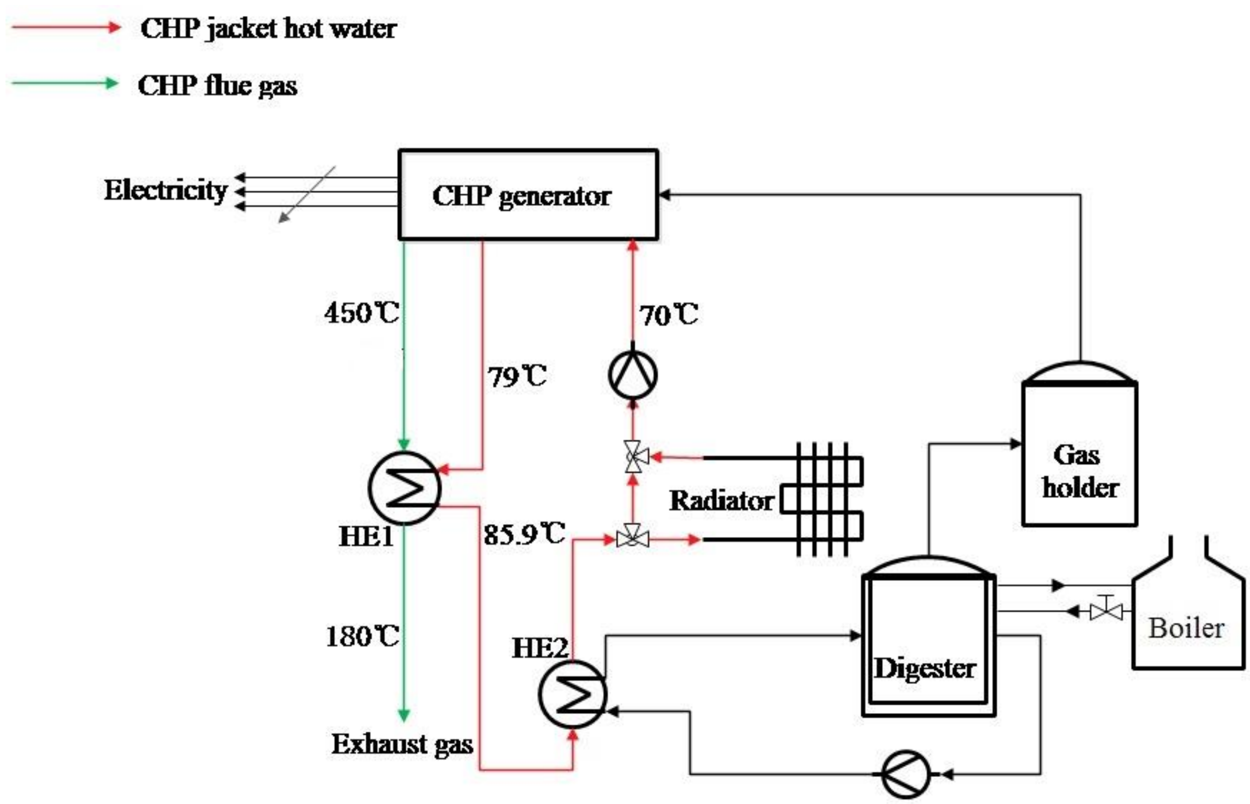

Fig.1 Schematic diagram of the existing CHP system

Table 1 Parameters of a typical CHP unit under nominal condition

CHP model: JMS 420 GS-B.L					
Nominal operation conditions: pressure 1000 mbar; temperature 25 °C; relative humidity 30%					
Electricity output	kW	1409	Primary water inlet temperature	℃	70
Exhaust gas before heat exchange 1	℃	450	Primary water before heat exchange 1	℃	79
Exhaust gas after heat exchange 1	℃	180	Primary water after heat exchange 1	℃	85.9
Biogas consumption	Nm^3/s	0.144	Primary water flow rate	m^3/h	75.5
Intake air volume	Nm^3/s	1.44	Fuel flow rate	Nm^3/s	1.48

However, a major problem of the CHP system can be detected in subtropical areas. Take Hong Kong as an example. By calculation, the total heating capacity provided by the exhaust gas and the jacket water is about 1360kW, which is much higher than the required heating load in summer. The excess thermal energy for each month is shown in Fig.2. It can be seen that even in the coldest month of January, the heating capacity provided can be higher than the required heating load by 200kW. More serious is that up to 1000kW of thermal energy is wasted in summer resulting in a very low thermal efficiency.

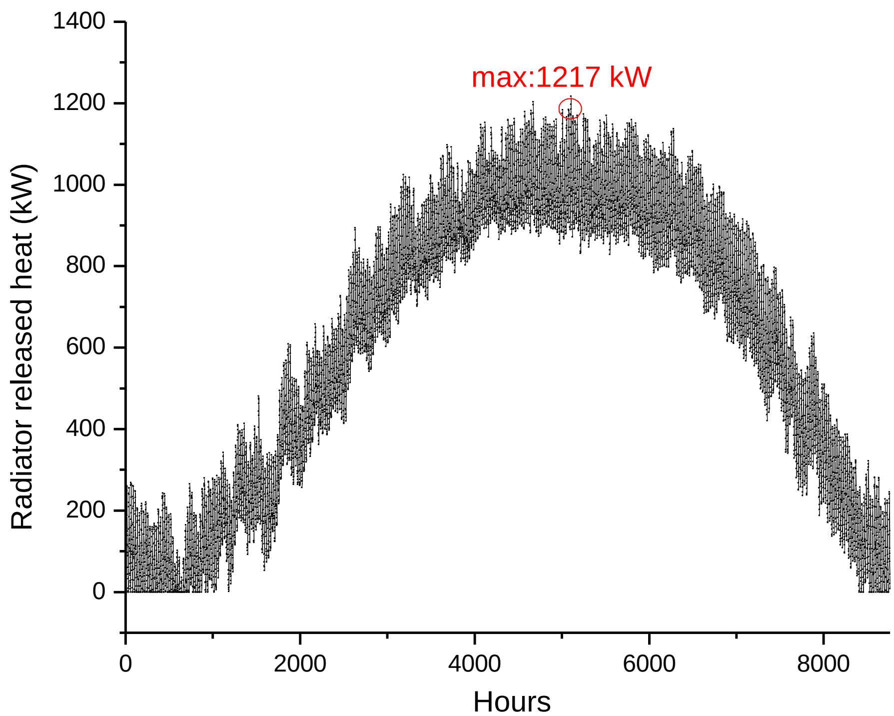

Fig.2 Excessive thermal energy released by radiator of existing system

3. Design of tri-generation systems

To solve the problem of existing CHP system, a tri-generation system is proposed to significantly improve the efficiency of thermal utilization. The proposed scheme is to use the excess thermal energy of the CHP system to drive a lithium-bromide absorption chiller. Its chilled water output is for space cooling of power house, office buildings or residential buildings nearby.

According to the different driving heat sources, absorption chillers can be classified into four types, namely hot-water driven, flue gas driven, steam driven and direct-fired absorption chiller. For the existing CHP system, the heat sources available for driving the chiller include the flue gas and jacket hot water. For different kinds of absorption chillers, the temperature requirements for the heat sources are different, which determines the design scheme of each tri-generation system as listed in Table 2. In the original CHP system (Fig.1), the low temperature heat source, such as 180℃ flue gas and 85.9℃ hot water can only drive the single effect absorption chiller. The high temperature 450℃ flue gas, in theory, can be further utilized in four ways: heating, cooling, producing water vapor and generating electricity. But it is an optimal choice for the 450℃ flue gas to drive the double effect absorption chiller directly for cooling considering the following aspects.

Table 2 Temperature requirements of the heat sources for different types of absorption chillers

	Single effect	Double effect
Flue gas driven absorption chiller	>180℃	430~520℃
Hot water driven absorption chiller	85~140℃	>140℃

Firstly, in real application, the 450℃ flue gas can hardly heat the steam to a temperature high enough for further driving the turbine for electricity generation; secondly, it is very difficult for electricity generation by using the 450℃ flue gas because the micro flue gas generator can be hardly found on the market nowadays. Moreover, as the tri-generation system discussed was a retrofitting scheme based on the existing CHP system, it should achieve higher benefits with less modification. Thus, the retrofitting investment is too large if adding another steam heater or gas engine. As a consequence, the 450℃ flue gas for driving the double effect absorption chiller is the optimal choice in the retrofitting scheme.

As analyzed above, four different types of tri-generation systems can be proposed according to the different driving heat source, and cooling and heating regulation scheme. The detail information of the four systems is shown in Table 3. The operational strategies for the proposed tri-generation system are as follows: whatever the conditions, priority is given to the digester's heating demand to maintain its temperature at 35°C and the excess thermal energy is used to drive the absorption chiller. The schematic diagrams of the four tri-generation systems are presented in Figs.3 to 6. For convenience, the four systems are denoted as systems 1 to 4, respectively.

Table 3 Information of the four proposed tri-generation systems

	System 1	System 2	System 3	System 4
Driving heat source	180℃ flue gas	85℃ hot jacket water	450℃ flue gas	450℃ flue gas
Absorption chiller	Flue gas driven single effect	Hot water driven single effect	Flue gas driven double effect	Flue gas driven double effect
Design temperature drop of the chiller	180℃ - 150℃	85.9℃ - 70℃	450℃ - 200℃	450℃ - 200℃
Cooling and heating regulation scheme	Cooling provided by flue gas driven chiller;	Distribution of 85.9℃ hot jacket water	Distributed by chiller (with hot water re-heater	Distribution of 450℃ flue gas

	Heating provided by jacket hot water		inside)	
Waste energy	150°C flue gas, all surplus hot water	180°C flue gas	150°C flue gas, part of hot water	150°C flue gas, part of hot water

- System 1: single effect absorption chiller driven by 180°C flue gas

System 1 aims to use the 180 °C exhaust fuel gas directly after the heat exchanger (HE1) to produce the 7/12°C chilled water. The thermal energy of exhaust gas is fully used and the temperature drops from 450°C to 150°C. As the jacket water system is unchanged, there is still a large amount of thermal energy in the hot water has to be released by radiator, especially in summer when the heating demand is low. In term of modification work, some supplementary equipment should be added, including a single effect flue gas absorption chiller, a cooling tower, two pumps, some valves and pipelines.

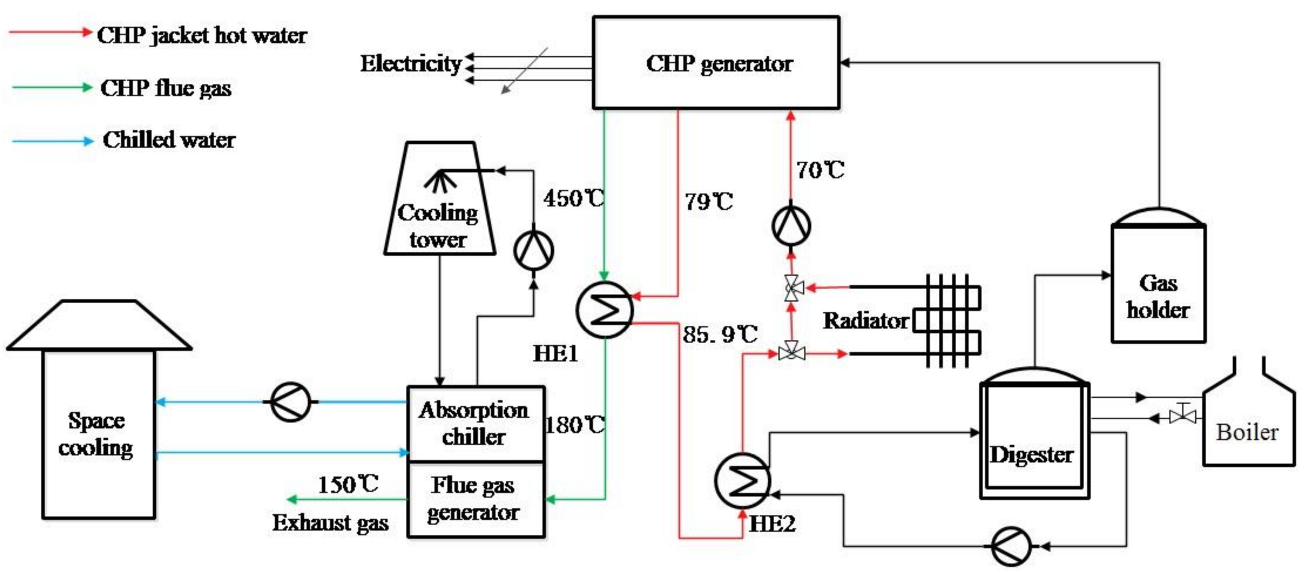

Fig.3 Schematic diagram of system 1

- System 2: single effect absorption chiller driven by surplus hot water

A single effect absorption chiller driven by 85.9℃ jacket water is proposed in system 2. On the basis of the existing system, the jacket water is divided into two parts: firstly, it is given the priority of satisfying the heating load of the digester and then the surplus hot water will be sent to the absorption chiller to produce chilled water. Compared with the existing CHP system, system 2 is able to make full use of the thermal energy of the jacket water, but the 180°C high temperature fuel gas is not fully recovered and exhausted to surrounding directly all year round.

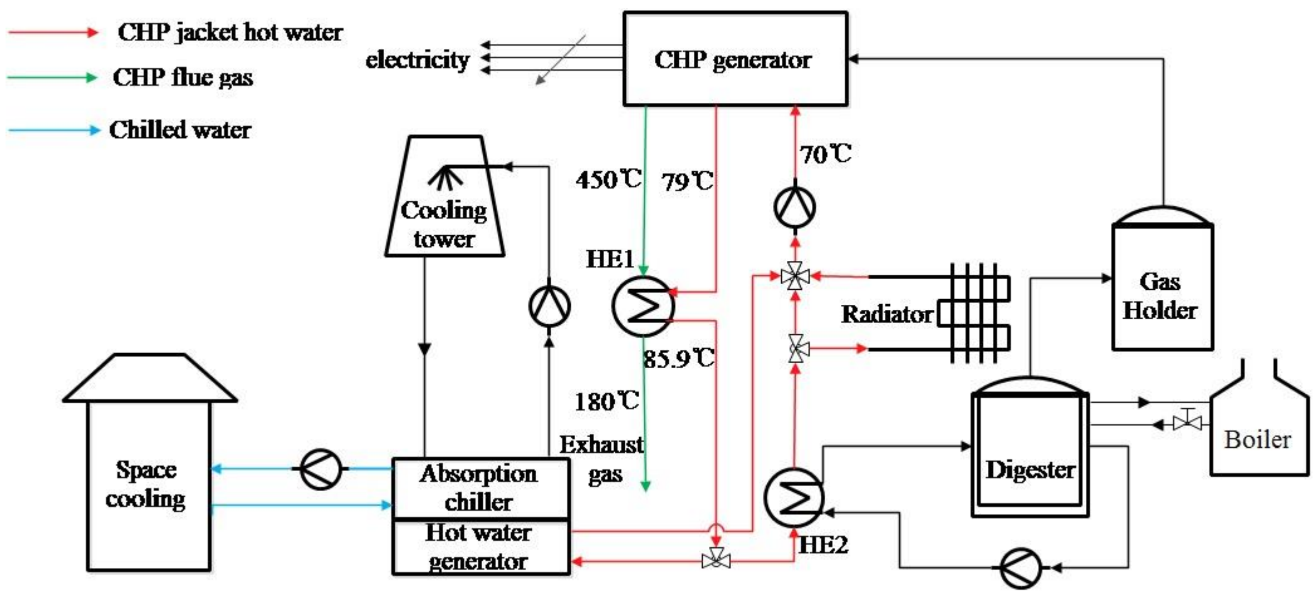

Fig.4 Schematic diagram of system 2

- System 3: double effect absorption chiller driven by 450℃ flue gas

A double-effect and dual-use absorption chiller, driven by the high temperature flue gas (450℃) is proposed in system 3, which could produce hot water and chilled water simultaneously. The original HE1 is removed so that the 450℃ flue gas from the generator is able to drive the double effect absorption chiller. The amount of cooling and heating capacity can be adjusted by the absorption chiller. When the required digester heating load is large, the jacket water from the CHP unit is first heated in the hot water heater of the absorption chiller and then further heated by the 200℃ exhaust gas. When the heating load is low, the jacket water will be directly transported to HE1 and HE2 to heat the digester. System 3 is the most complex in terms of control. The cooling/re-heating regulation inside the chiller will greatly affect the performance of the whole system.

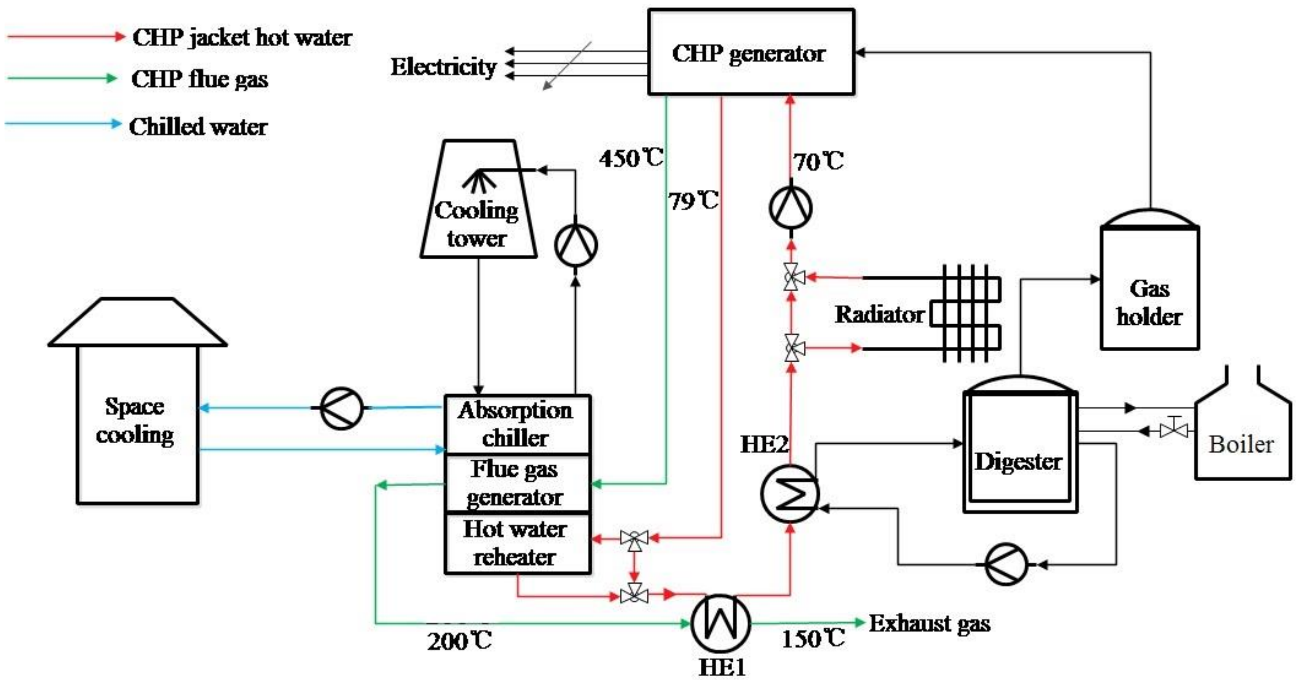

Fig.5 Schematic diagram of system 3

- System 4: double effect absorption chiller driven by bypass of 450°C flue gas

In system 4, the 450℃ flue gas from the generator is divided into two parts: one part is supplied to HE2 for heating the digester in priority and the other part is transported to the double-effect absorption chiller for producing chilled water. The cooling/heating outputs are regulated by distributing the flue gas according to the demands, thus the heating requirement of the digester could be guaranteed in priority. In cold days, the jacket hot water is firstly heated by the 200℃ exhaust gas from the chiller and then reheated by the 450℃ bypassed flue gas. In all, there are three heat exchangers in the primary water loop: HE1 is served to recover the exhaust heat in the flue gas; HE2 is to reheat the jacket water by the bypassed flue gas; HE3 is for heating the digester.

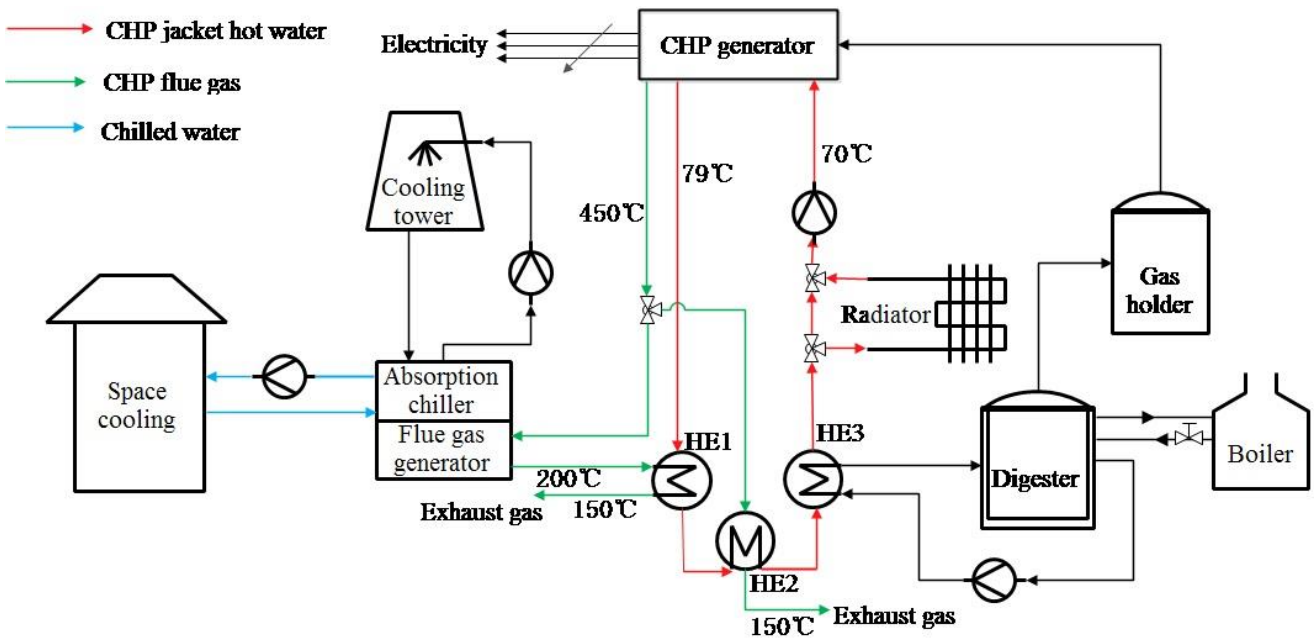

Fig.6 Schematic diagram of system 4

4. Mathematical modelling

A customer-defined code based on the following mathematic equations was programed by MATLAB, for simulating the thermal performances of the above four tri-generation systems. Firstly, the weather data, enthalpy of the flue gas, physical parameters of water, as well as the design parameters of each system were set as input for the system simulation. Then, the heating load was estimated before processing the corresponding simulation procedure for each system. The distribution of the flue gas or hot water was simulated thereafter according to different system's design. In the end, different energy utilization efficiencies can be obtained. Fig. 7 shows the simulation flow chart of the four tri-generation systems.

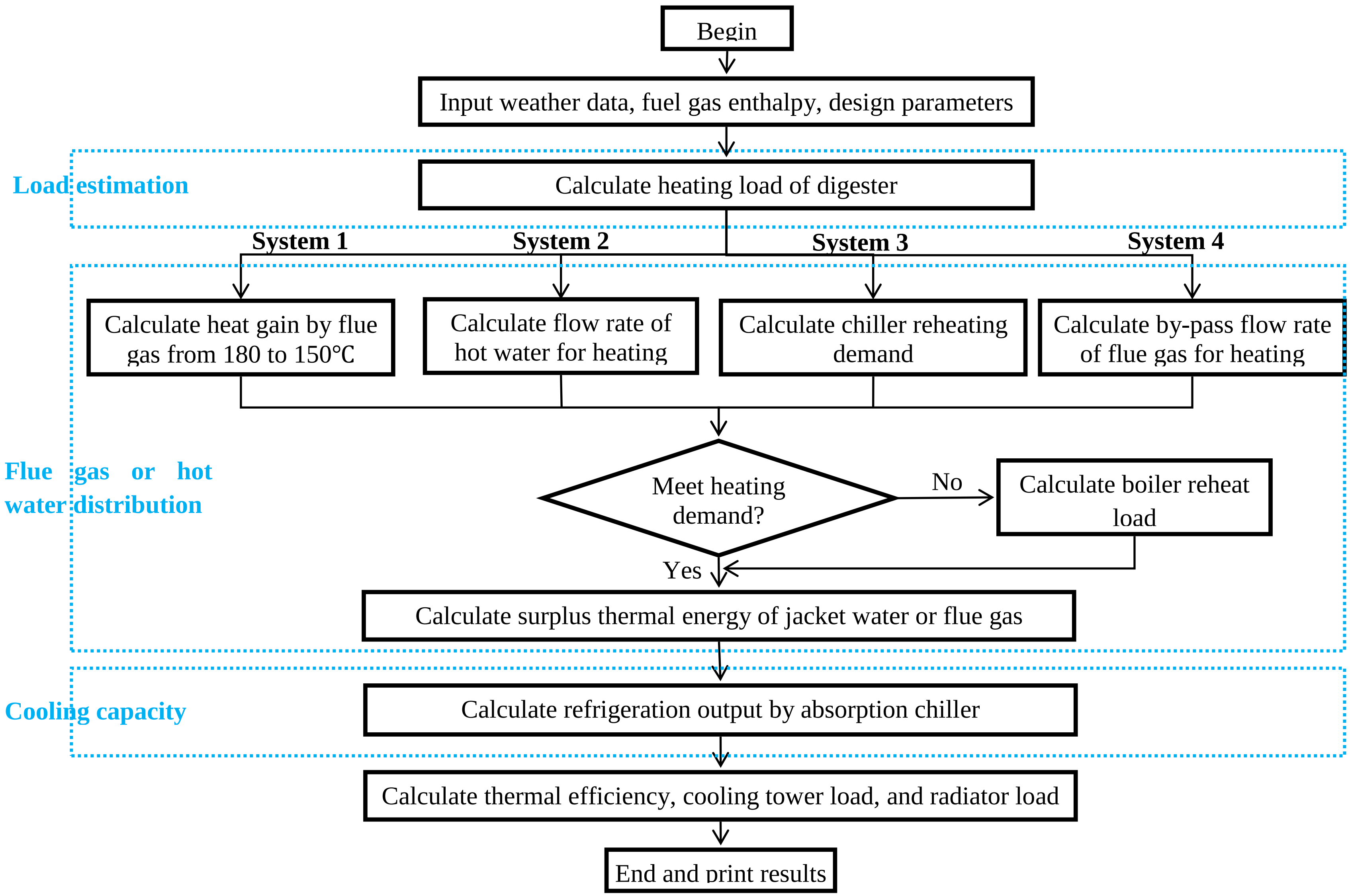

Fig.7 Simulation flow chart of the four tri-generation systems

4.1 Weather data, system operation schedule

Hong Kong is selected as a reprehensive subtropical area for study. It owns typical hot and humid weather all year around. The 20 years average hourly weather data of Hong Kong are shown in Fig.8. The temperature varies from 13.4℃ to 32.8℃ and the relatively humidity fluctuates from 45.1% to 87.3%.

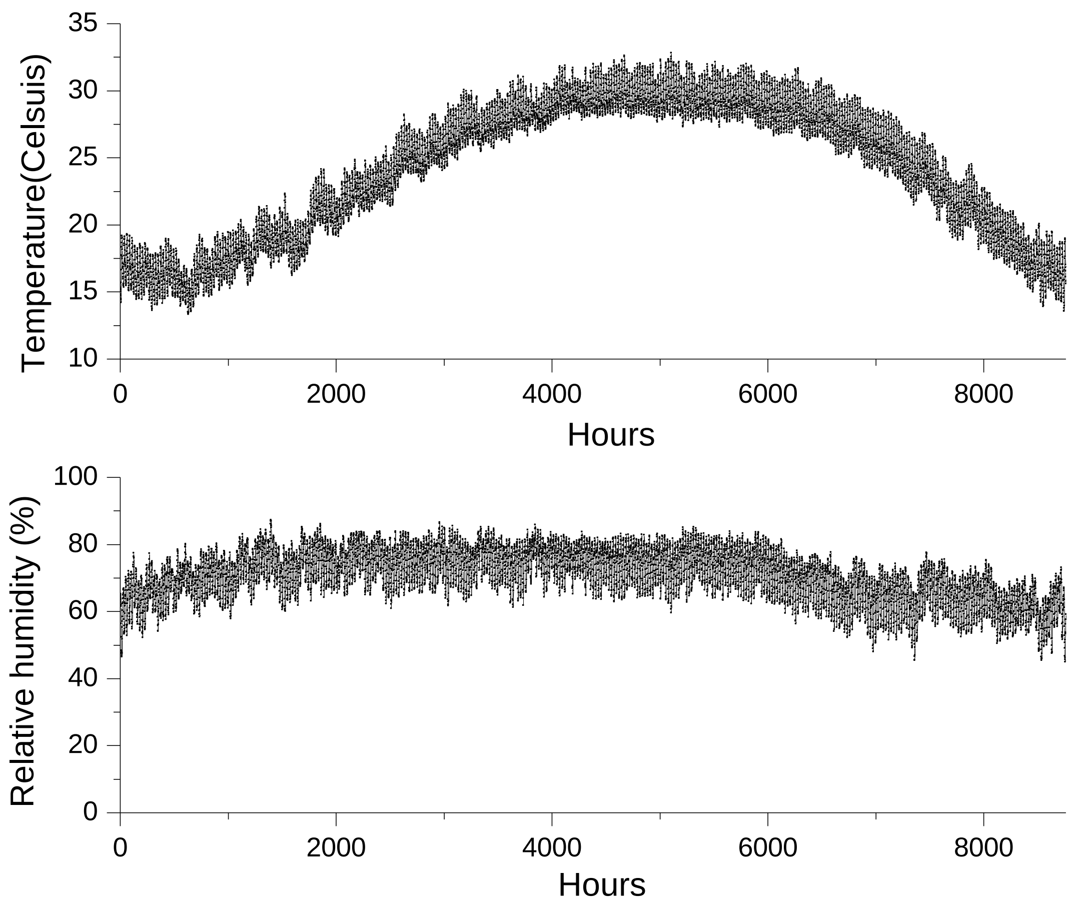

Fig.8 Hourly air temperature and humidity in Hong Kong

As the CHP requires scheduled maintenance so it can not operate at full load for 8760 hours in a year. According to the data provided by the GE Company, the annual operating hours of Jenbacher gas engine can be more than 8000 hours with an availability of over 95% under full load conditions [22]. So the simulation assumptions can be made as: the annual operation rate of the CHP unit is set as 95% and always operates at full load (8320 hours in total). Besides, according to the maintenance guide of the gas engine, attendance for replace of consumables such as oil and filters typically occurs every 1000 to 2000 operational hours [23]. So the scheduled maintenance is set to be executed every 2000 hours. In sum, the CHP annual operation schedule is plotted as Fig.9, in which '0' represents stop for maintenance while '1' represents operation at full load.

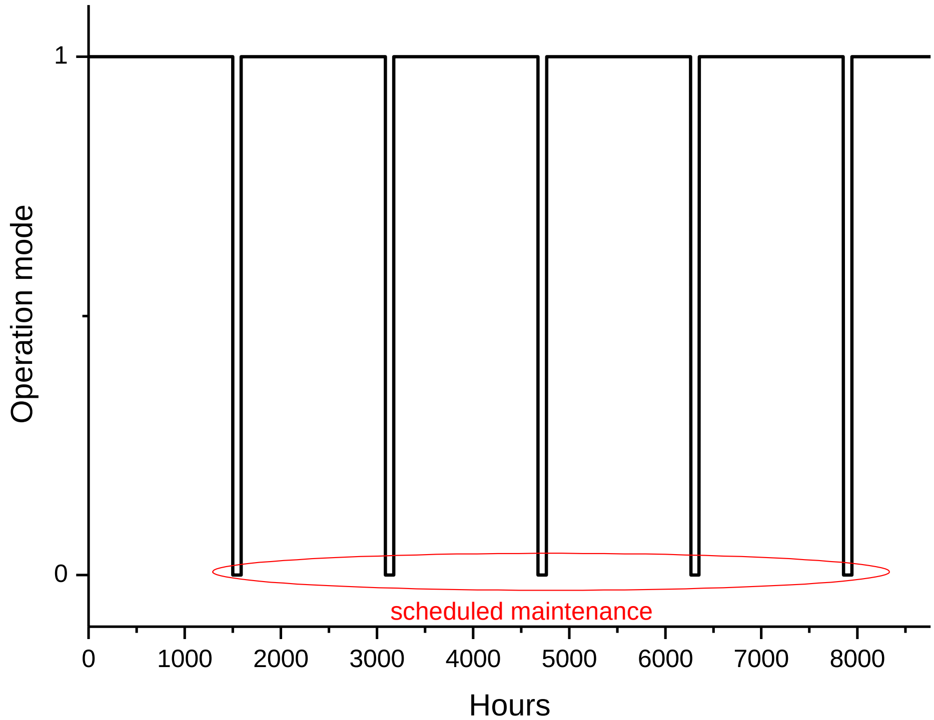

Fig.9 CHP operation schedule

4.2 Heating load of digester

A digester has to be operated at the design temperature of 35℃ to maintain a high biogas production rate. The digester heating loads, varying with the change of ambient temperature can be calculated by:

$$Q_{HL} = \sum Q_{HL_i} = \sum F_{digester} \cdot K_i \cdot \left(t_{set} - t_{ambient}\right) \quad (1)$$

where, Q_{HL} is the total heating load of the digesters, kW; Q_{HL_i} is the heating load for one particular digester, kW; t_{set} is the insulation temperature of the digester which is set as 35℃; $t_{ambient}$ is the ambient air temperature, ℃.

As the surface area of digester tank (F_{digester}) and its heat transfer coefficient (K_{i}) are difficult to obtain, the value of $\sum F_{digester} \cdot K_i$ are calculated through the heat balance equation of the secondary water loop, expressed as:

$$\sum F_{digester} \cdot K_i = \frac{\rho V_2 \cdot c_p \cdot (t_{in} - t_{out})}{t_{set} - t_{ambient}} \quad (2)$$

where, V_2 is the flow rate of the secondary heating water loop, m^3/s; t_{in} and t_{out} are the inlet and outlet temperature of the secondary water, ℃.

According to the data collection on site in Satin Sewage Treatment Work on 28th February, the recorded water flow rate and temperature for two times measurement were: V_2=60.6m^3/h, t_{in}=67.9℃, t_{out}=53.1℃ and V_2=62.0m^3/h, t_{in}=68.9℃, t_{out}=55.1℃, respectively. The recorded ambient temperature was 18.9℃. So $\sum F_{digester} \cdot K_i$ was calculated to be 63.9 W/℃ and 61.3 W/℃ and determined as 62.6 W/℃ for average.

4.3 Modeling of the tri-generation system

The simulation model of the tri-generation system was developed based on the heat balance equations of each component, which are elaborated as follows.

1） Energy balance in the jacket water loop

For system 1, system 2 and system 3, the energy balance in the jacket water loop can be expressed as:

$$Q_{CHP} + Q_{HE1} = Q_{HE2} + Q_r \quad (3)$$

For system 4, the energy balance in the jacket water loop can be expressed as:

$$Q_{CHP} + Q_{HE1} + Q_{HE2} = Q_{HE3} + Q_r \quad (4)$$

where, Q_{CHP} represents the heat gain from CHP unit, kW; Q_{HE1}, Q_{HE2} and Q_{HE3} are the heat exchange rate by the heat exchanger 1, 2 and 3, respectively, kW; Q_r is the excess thermal energy released by the radiator, kW. Each item can be further calculated as follows.

$$Q_{CHP} = m_w \cdot c_p \cdot (t_{out} - t_{in}) \quad (5)$$

where, m_w is the mass flow rate of jacket water, kg/s; t_{in} and t_{out} is the temperature difference between the inlet and outlet of the jacket hot water, which are 70℃ and 79℃, respectively.

The energy balance in the heat exchanger 1 (HE1) can be calculated as:

$$Q_{HE1} = \eta_1 \cdot V_{g1} \cdot \Delta h = c \cdot m_{w1} \cdot \Delta t_w \quad (6)$$

where, Δh is the enthalpy difference between the inlet and outlet flue gas, kJ/Nm^3; η_1 is the heat transfer efficiency of HE1, which is set as 0.9; Δt_w is the temperature difference between the inlet and outlet jacket water, ℃; c is the specific heat of water, kJ/kg·℃. V_{g1} and m_{w1} are the flue gas flow rate and jacket water flow rate through the HE1, Nm^3/s and kg/s. For system 1~4, m_{w1} is equal to the total flow rate of jacket water, 20.3kg/s (75.5m^3/h); For system 1~3, V_{g1} is equal to the total flow rate of exhaust fuel gas, 1.48Nm^3/s; while for system 4, the fuel gas flow rate supplied to the HE1 (V_{g1}) and to HE2 (V_{g2}) satisfies:

$$V_{g1} + V_{g2} = 1.48 \quad (7)$$

The energy balance equation in the heat exchanger 2 (HE2) in system 1~3 and heat exchanger 3 (HE3) in system 4 are the same. Taking HE2 as an example:

$$Q_{HL} = Q_{HE2} + Q_{boiler} = \eta_2 \cdot c \cdot m_{w2} \cdot \Delta t_w + Q_{boiler} \quad (8)$$

where, Q_{boiler} is the heating load of the boiler, kW. η_2 is the heat transfer efficiency of HE2, which is set as 0.9; Δt_w is the temperature difference between the inlet and outlet jacket water, ℃; m_{w2} is mass flow rate of the jacket hot water through the HE2, kg/s. For system 1, 3 and 4, m_{w1} is equal to the total flow rate of jacket water, 20.3kg/s (75.5m3/h); while for system 2, the flow rate of jacket hot water supplied to chiller (m_{wc}) and HE2 (m_{w2}) satisfies:

$$m_{wc} + m_{w2} = m_{w1} = 20.3 \quad (9)$$

The excess thermal energy in the jacket water is released to surrounding through the radiator, and this part of the wasted energy can be calculated by:

$$Q_r = c \cdot m_{w2} \cdot \Delta t_r \quad (10)$$

where, Δt_r is the temperature difference between the inlet and outlet temperature (70℃) of the jacket water, ℃.

In the above equations, the enthalpy of flue gas (*h*) at certain temperature is determined by the volume content of each component of the fuel gas. Through the measurement, the biogas produced in the Satin Sewage Treatment Works mainly consists of 53% CH_4 and 47% CO_2. After the CH_4 combustion with 10 times intake air volume, the fuel gas contains 15.9% CO_2, 16.8% N_2 and 67.3% water vapor. The fuel gas enthalpy at different temperature (shown in Table 4) can be calculated by the weighted average method using the enthalpy of each component and its corresponding volume content.

Table 4 Enthalpy of flue gas at different temperatures

Temperature (℃)	100	150	180	200	300	400	450	500
Enthalpy (kJ/Nm^3)	150.1	227.7	274.3	305.4	466.1	632.6	718.5	804.5

2） Energy balance in the absorption chiller

The energy balance in the absorption chiller is given by:

$$Q_e + Q_g = Q_c + Q_a \quad (11)$$

where, Q_e represents the load of the evaporator, kW; Q_g is the load of the generator of the absorption chiller, kW; Q_c and Q_a are the loads of the condenser and the absorber, respectively, kW.

For system 3, the energy balance in the absorption chiller is a little different and can be expressed as:

$$Q_e + Q_g = Q_c + Q_{rh} + Q_a \quad (12)$$

where, Q_{rh} is the load of the hot water re-heater inside the chiller, kW.

For the fuel gas driven absorption chiller as system 1, 3 and 4, the load of the generator can be calculated as:

$$Q_g = V_{g1} \cdot \Delta h' \quad (13)$$

where, $\Delta h'$ represents the enthalpy difference between the inlet and outlet flue gas, kJ/Nm^3. The designed temperature drop of the fuel gas in each system refers to Table 3.

For the hot water driven absorption chiller as system 2, the load of the generator can be calculated as:

$$Q_g = m_{wc} \cdot c \cdot \Delta t_w \quad (14)$$

where, Δt_w is the temperature difference between the inlet and outlet hot jacket water of the chiller.

The load of the evaporator or the refrigeration output of the chiller can be calculated as:

$$Q_e = Q_g \cdot COP = c \cdot m_{chw} \cdot \Delta t_{chw} \quad (15)$$

where, m_{chw} is the mass flow rate of chilled water, kg/s; Δt_{chw} is the temperature difference between the inlet (7°C) and outlet (12°C) chilled water of the evaporator, °C. The coefficient of performance (COP) of both the single effect absorption chiller and double effect absorption chiller vary according to the working load. The part load performances of the two kinds of absorption chiller are plotted in Fig.10 [24]. Through the polynomial fitting of the curves, the mathematic expression of COP (y) under various working load (x) can be calculated by:

$$y = 0.0001x^3 - 0.0061x^2 + 0.0578x + 0.8114 \quad (16)$$

and

$$y = 0.0004x^3 - 0.0125x^2 + 0.0912x + 1.1867 \quad (17)$$

for single effect absorption chiller and double effect absorption chiller with the R^2 of 0.9982 and 0.9873, respectively. The Equation (16) and (17) were programmed into the simulation code for predicting the chiller performances under the changing load.

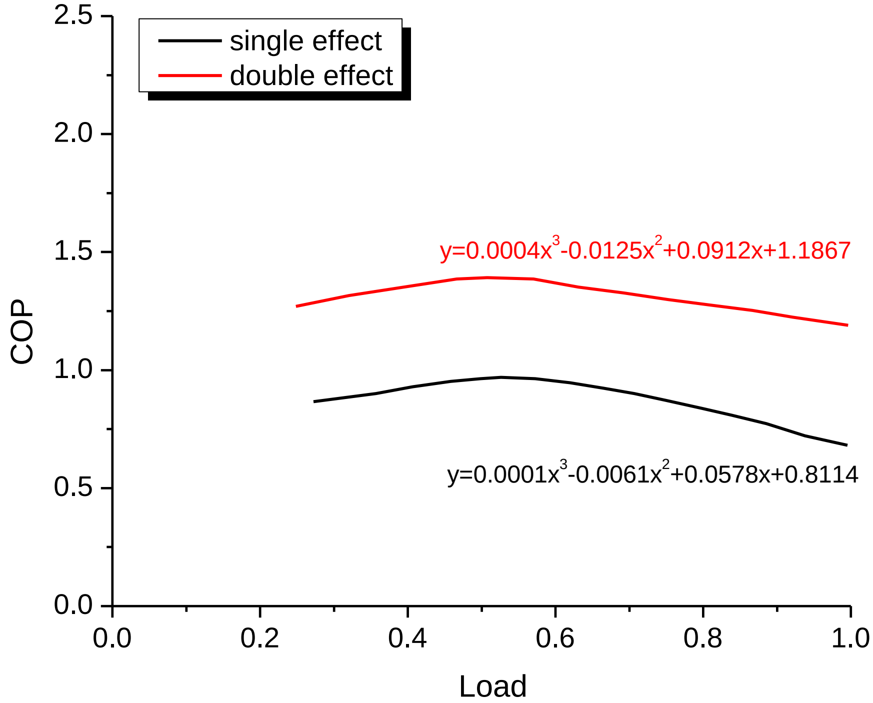

Fig.10 COP of absorption chiller under different working load

3） Energy balance in the cooling tower

The cooling capacity of the cooling tower Q_{ct} can be calculated by:

$$Q_{ct} = Q_c + Q_a = cm_{cw}\Delta t_{cw} \quad (18)$$

where, m_{cw} is the flow rate of cooling water, kg/s; Δt_{cw} is the temperature difference between the inlet (37℃) and outlet (32℃) cooling water of the cooling tower.

4） Thermal efficiency

The thermal efficiency of the tri-generation system is calculated as:

$$\eta = (Q_{HL} + Q_e - Q_{boiler}) / Q_b \quad (16)$$

where, Q_b represents the combustion heat of the biogas, kW, calculated as:

$$Q_b = V \cdot H \quad (17)$$

where, V represents the biogas consumption rate, which is 0.144 Nm^3/s; H is the heat value of biogas combustion, which is 22.572 MJ/Nm^3;

5. Simulation results and discussion

The annual dynamic simulation results of four tri-generation systems were presented. The simulations were conducted at one hour step for accurately predicting the performances of absorption chiller, cooling tower, radiator and boiler. In order to illustrate the year round performances of each components clearly and continuously, the scheduled maintenance were not displayed in the below figures, but summarized in the corresponding descriptions.

5.1 Heating load profile

The hourly heating load of digester is shown in Fig.11. It can be seen that the heating load varies greatly from season to season and even in a day. The maximum heating load is 1355 kW in January and the minimum load is only 137 kW in July. As the heat gain of the jacket water from CHP unit and exhaust fuel gas was calculated to be 1360 kW, in theory, it can just cover the peak heating load of the digester. However, the heat loss is inevitable during the heat transfer process in HE2, so the total heat provided is only 1224 kW when 0.9 heat transfer efficiency is considered. As a result, the heating capacity provided by the CHP can’t meet the peak digester heating demand in some extremely cold days, so supplementary heating should be provided by auxiliary electricity-driven boiler. On the other hand, as the lowest heating load accounts for only

1/10 of the total thermal energy provided, the excess thermal energy has to be wasted in original system. But by adopting absorption chiller, the excess thermal energy can be used for space cooling, which just matches the cooling demand trend around the year.

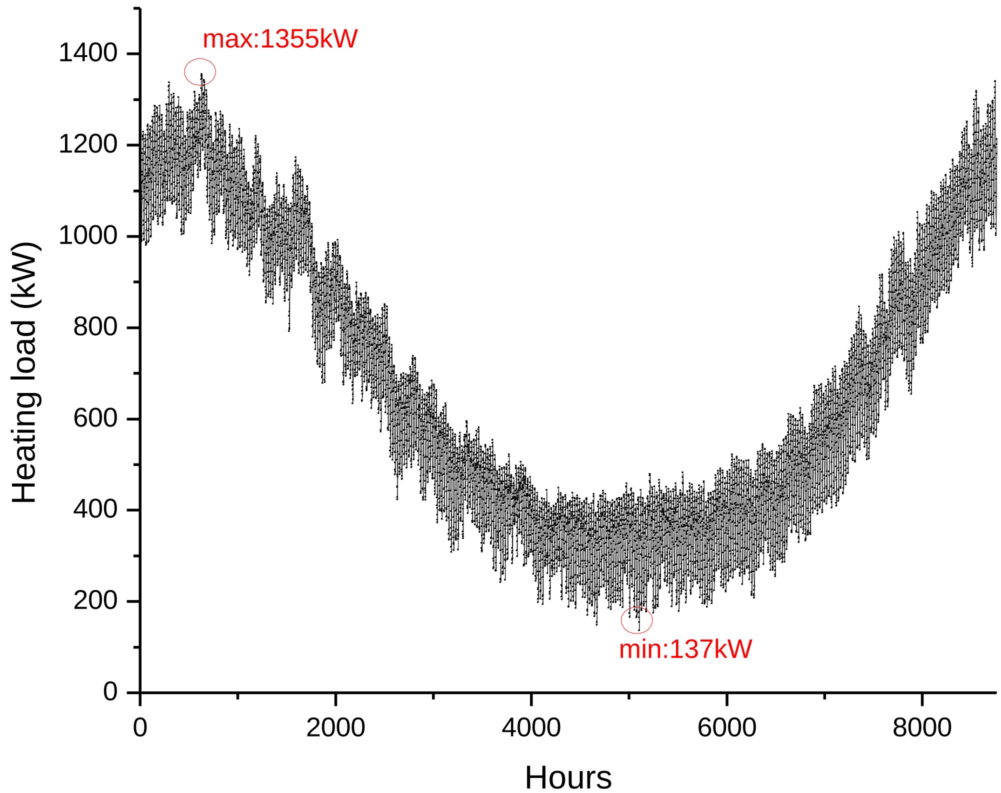

Fig. 11 Heating load of the digesters

5.2 Energy performance results for tri-generation systems

5.2.1 Cooling performance

Fig.12 shows the hourly COP variation of the absorption chiller in different tri-generation system. For system 1 and 2, the single effect absorption chiller is adopted. In system 1, the COP keeps a constant value of 0.68 all year around because the driving heat source is 180℃ fuel gas with constant flow rate of 1.48 Nm^3/s at CHP full load. In system 2, the COP fluctuates from 0.61 to 0.96 with the average value of 0.86. The COP varies because of the dynamic regulation of hot jacket water for heating and cooling. The highest COP occurs in mild weather conditions when the chiller operates at around 50% part load. It declines in both summer and winter when the working load is large or small. As the cooling capacity of the absorption chiller can be only adjusted within 10% to 100% working load, the chiller is off at some moments when the load is less than 10% in cold months.

For system 3 and 4, the double effect absorption chiller is adopted. The COP varies from 1.04 to 1.38 with the average value of 1.23. Unlike the chiller of system 2, in which the COP fluctuates around the year, the COP of the chiller in system 3 and 4 keeps constant at 1.19 in summer. It is because the 200℃ fuel gas from double effect absorption chiller can fully meet the low heating demand in summer, so that all the 450℃ fuel gas is bypassed for cooling and the chiller always operates at full load with constant COP value.

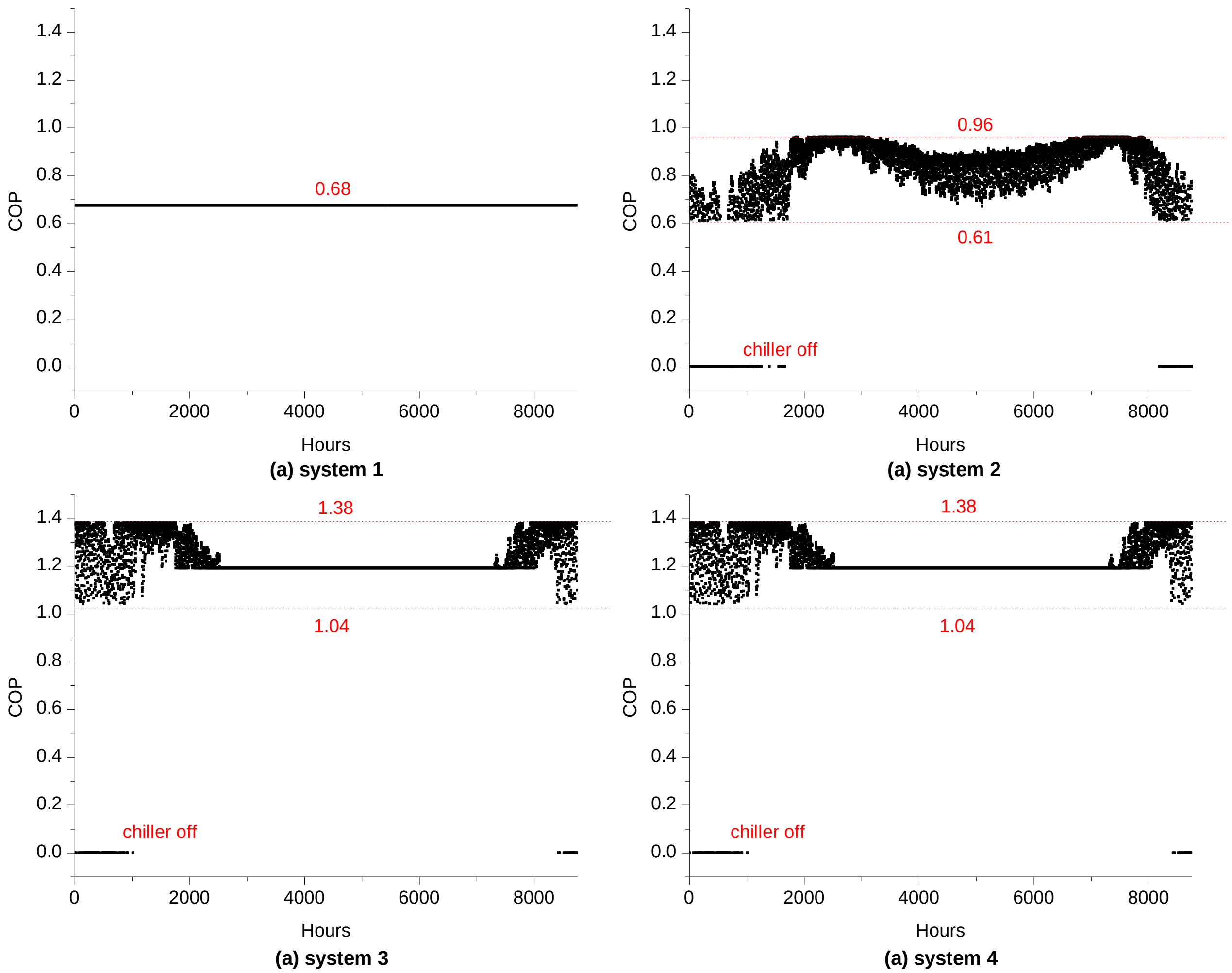

Fig.12 COP of the absorption chiller in different tri-generation system

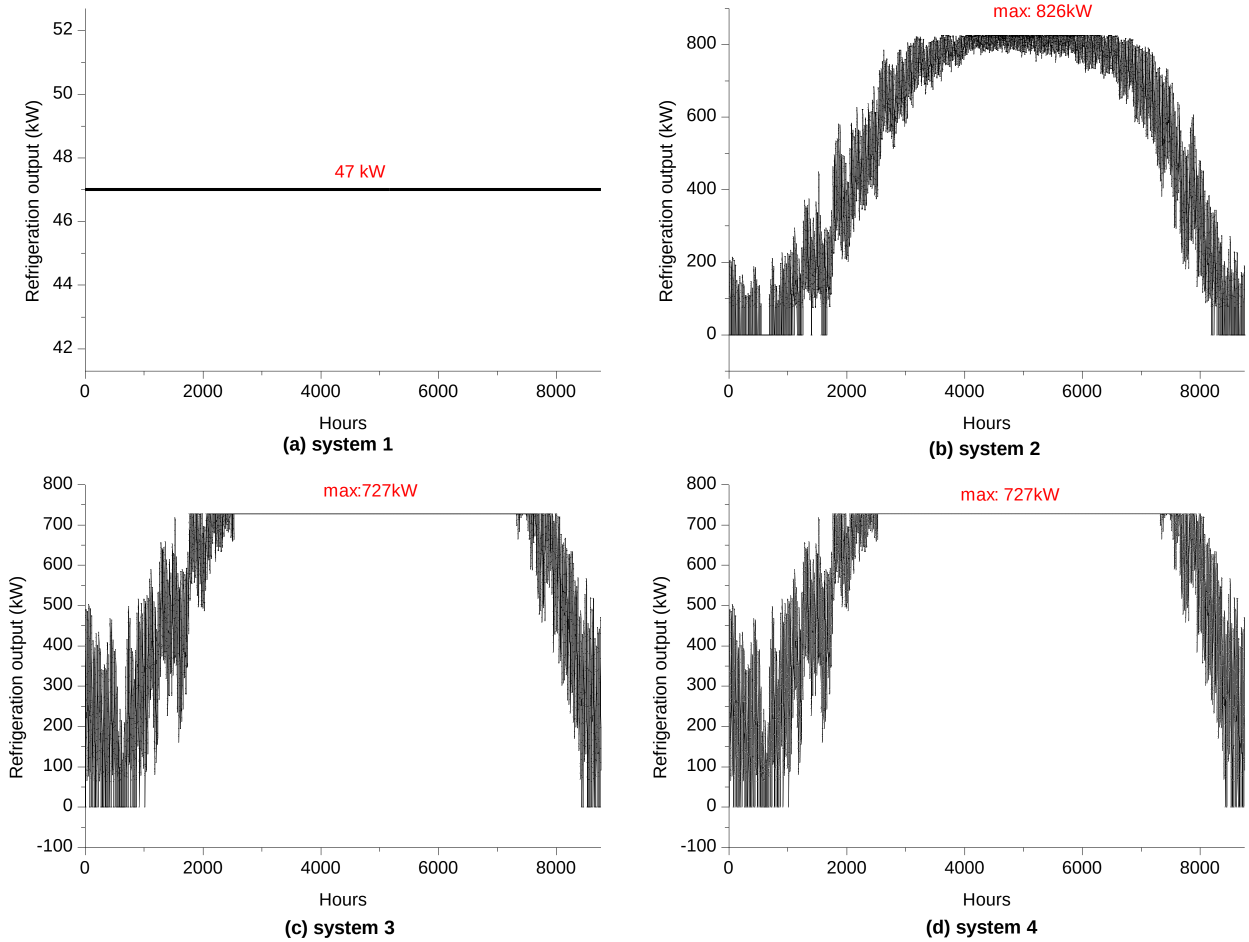

Fig.13 Refrigeration output of different tri-generation system

Fig.13 shows the hourly refrigeration output of different tri-generation system. Except system 1 which produces the cooling at constant 47 kW all year round, the rest systems produce more refrigeration in summer than winter because of more thermal energy input. The peak refrigeration output for system 2, 3 and 4 are 826 kW, 727 kW and 727 kW, respectively, which determines the cooling capacity of the absorption chiller. The annual total refrigeration output for the fours systems are: 3.91E+05 kWh (system 1), 4.30E+06 (system 2), 4.95E+06 (system 3) and 4.95E+06 (system 4), respectively.

Compared with system 1, the rest three tri-generation systems enjoy much larger refrigeration output because much more excess heat is effectively used. The thermal energy is the mostly fully used in system 2 for the reason that the hot jacket water is totally regulated based on the heating demand and all the excess heat is used for cooling without wasted by the radiator. But for system 3 and 4, all the fuel gas will be transported for producing cooling and discharged at 200℃ for heating the jacket water non-adjustably, when the heating load is low in summer. Therefore, the excess thermal energy has to be released by radiator if the heating capacity provided is larger than the heating demand. However, although the thermal energy input of system 2 is larger than that of system 3 and 4, the annual refrigeration output is less because of the lower COP of the single effect absorption chiller.

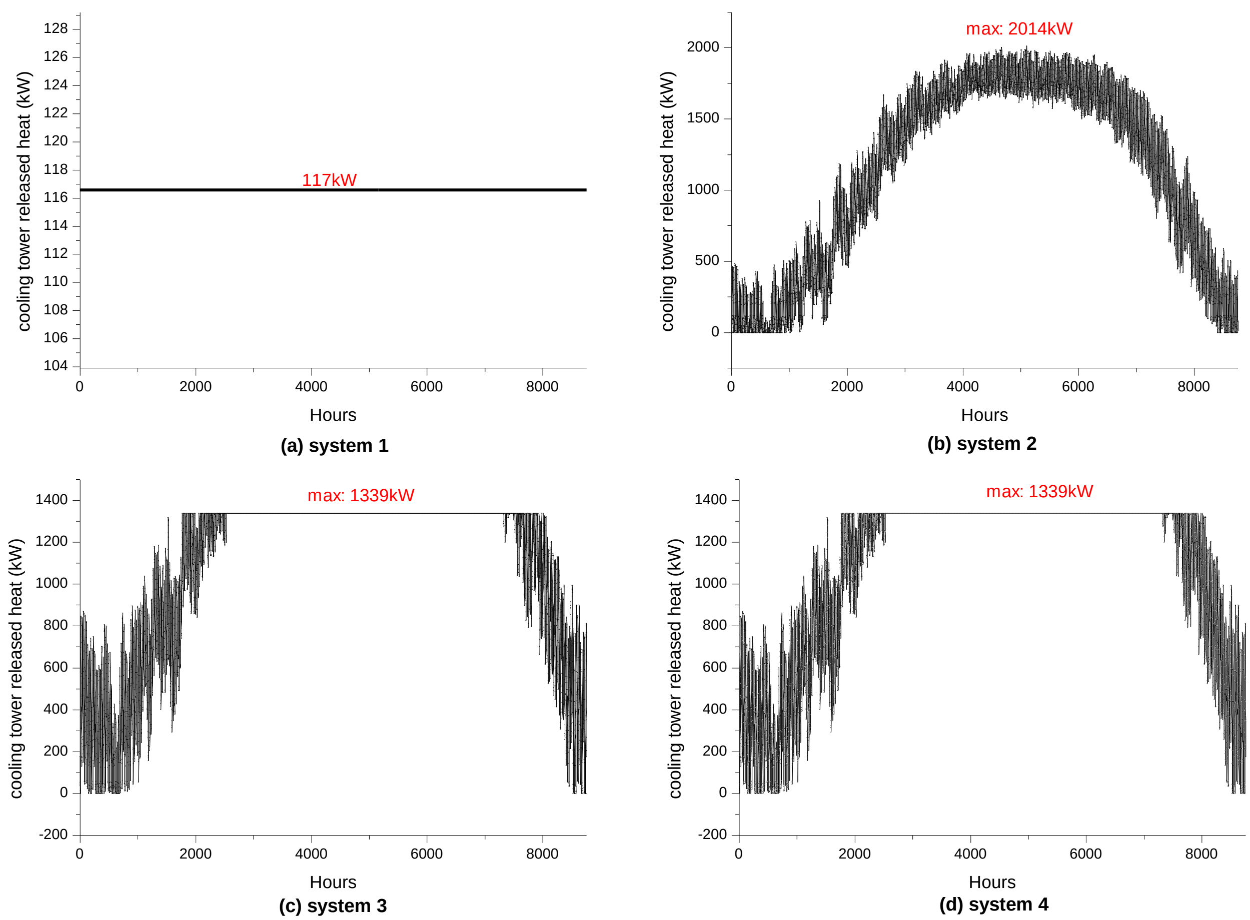

Fig.14 Cooling tower load of different tri-generation system

Fig.14 shows the cooling tower load of different tri-generation system. In system 1, the cooling tower load keeps constant at 117kW all year round; while for the any systems, the load is higher in summer than in winter. The peak loads of the cooling tower are 117 kW for system 1, 2014 kW for system 2, 1339 kW for system 3 and 4, respectively. Because more thermal energy is inputted in system 2 for generating refrigeration, the annual cooling load of the cooling tower is also larger than the other systems.

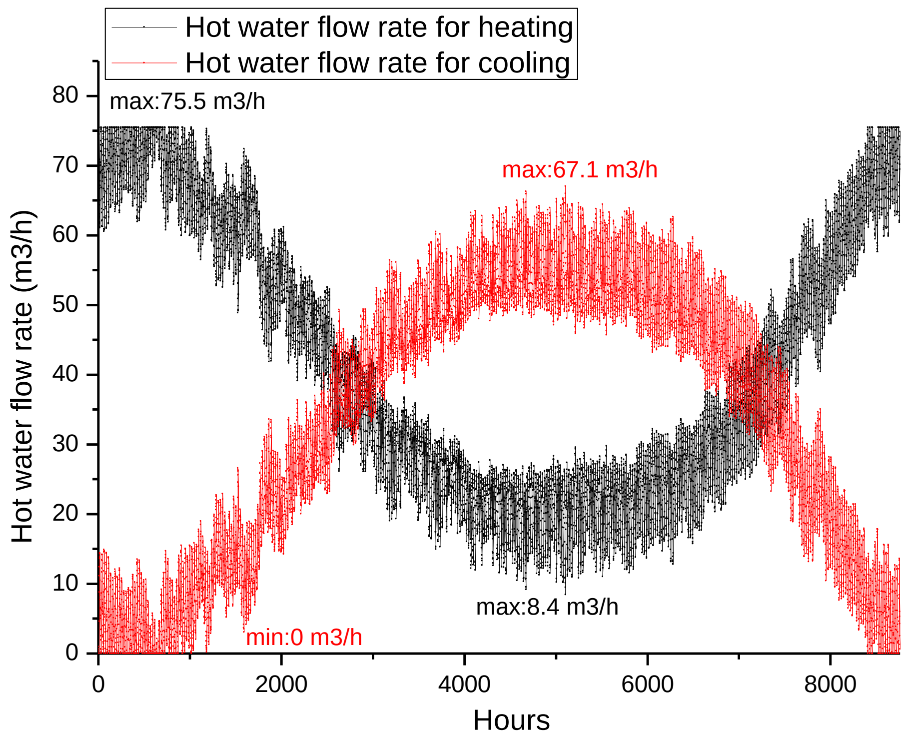

Fig. 15 Hot water regulation for cooling and heating in system 2

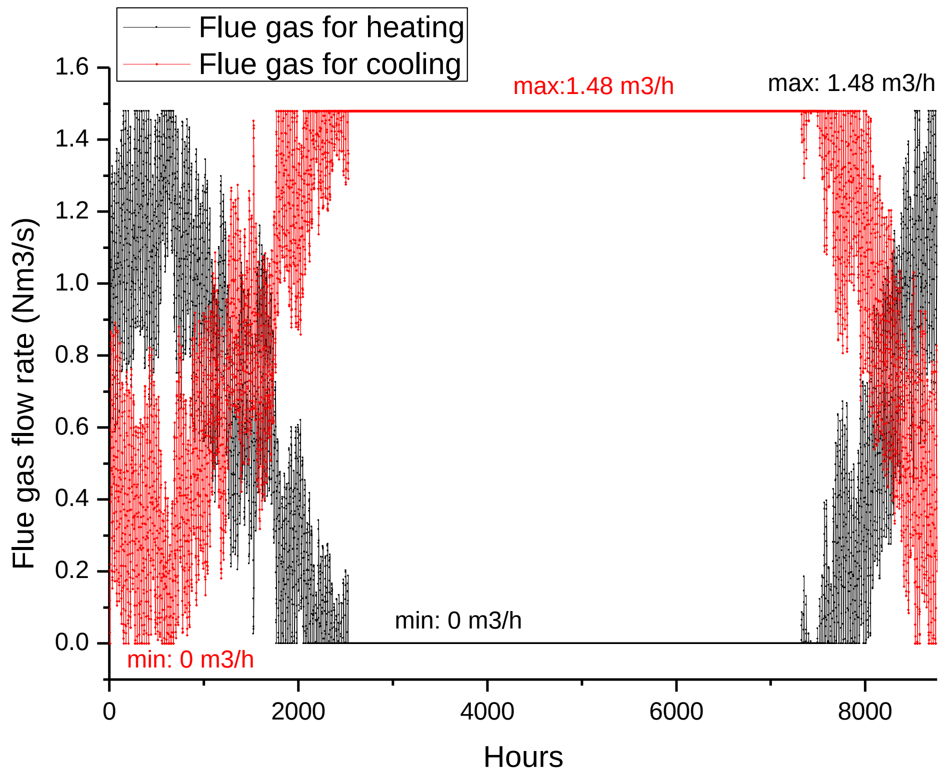

Fig. 16 Flue gas regulation for cooling and heating in system 4

Fig.15 and Fig.16 show the hourly regulation of the hot jacket water and flue gas in system 2 and system 4. As it can be seen from the figures, the annual regulation trend is similar. In system 2, most of the hot water is transported for digester heating and little is left for cooling in cold days. The peak hot water demand is 75.5 m^3/h and 0 m^3/h is left for cooling. While in hot days, heating demand is little with minimal flow rate of 8.4 m^3/h and the majority is used for cooling. In system 4, the minimal fuel gas flow rate is 0 Nm^3/s for heating in summer and is 0 Nm^3/s for cooling in winter.

Compare the system 3 and 4, the heat and cooling regulation methods are much different. In winter, if the 200 ℃ exhaust gas from the chiller cannot meet the digester heating load, the jacket hot water is therefore reheated but by different strategies. In system 3, the reheating process is completed by the water re-heating generator inside the absorption chiller, while in system 4, reheating is realized by bypassing a part of 450℃ flue gas through a three-way valve.

5.2.2 Boiler heating and radiator performance

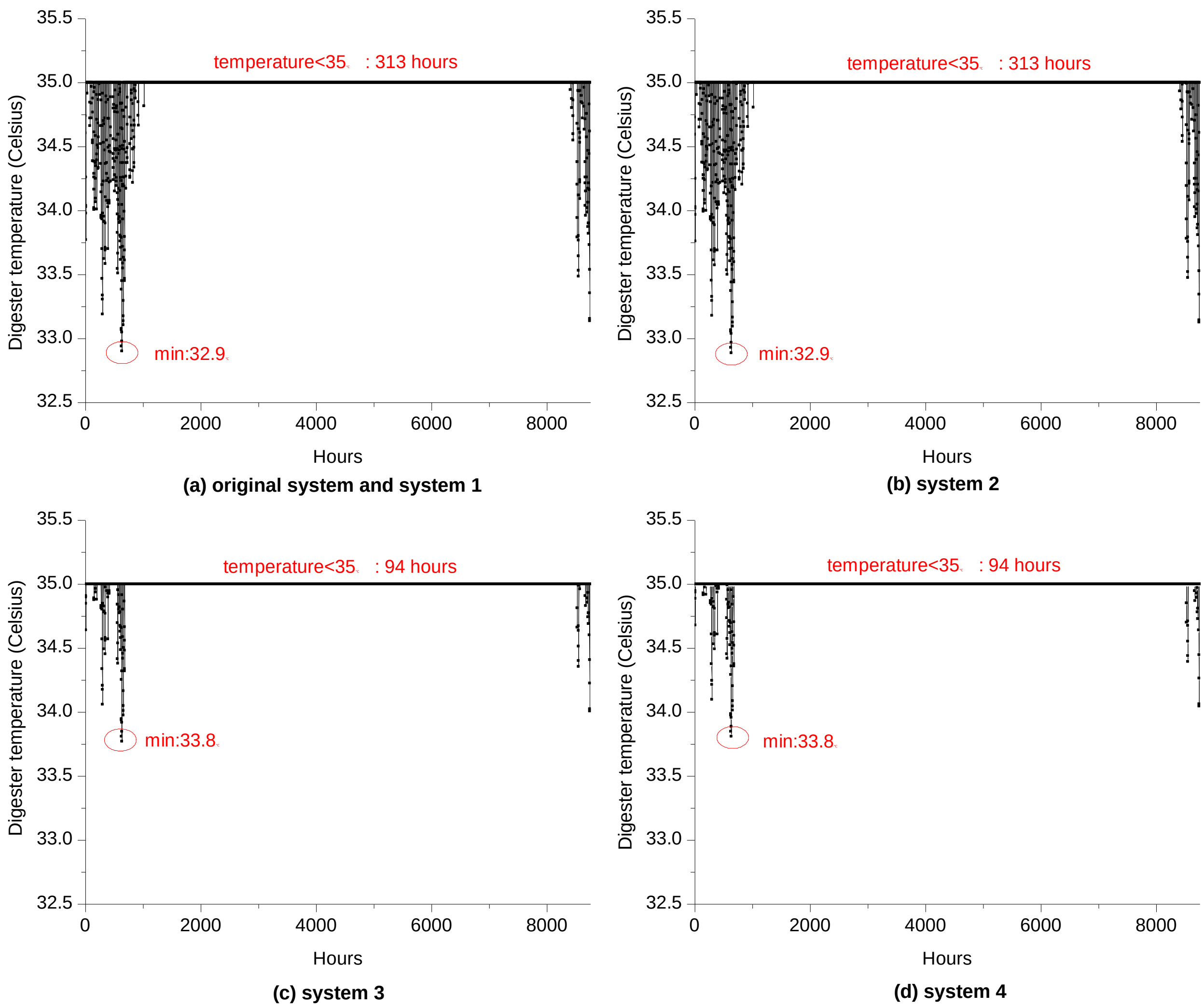

Fig. 17 Digester temperature in different tri-generation system without boiler heating

As it was analyzed above in Chapter 5.1, the heating load of the digester cannot be fully guaranteed on some extremely cold moments; therefore the digester can't be heated to 35℃.

Fig.17 shows the hourly digester temperature of different system if auxiliary boiler re-heating is not provided. For original system, system 1 and system 2, the lowest digester temperature can only reach 32.9℃, 2.1℃ lower than the required temperature, and the annual non-guarantee hours for digester temperature is 313 hours in total; while for system 3 and system 4, the lowest digester temperature is 33.8℃, still 1.2℃ lower than requirement, and the annual non-guarantee hours is 94 hours. The reason why system 3 and 4 achieve less non-guarantee hours is that the thermal energy in 180℃ fuel gas is fully used for re-heating the jacket water, but it is exhausted directly in system 2.

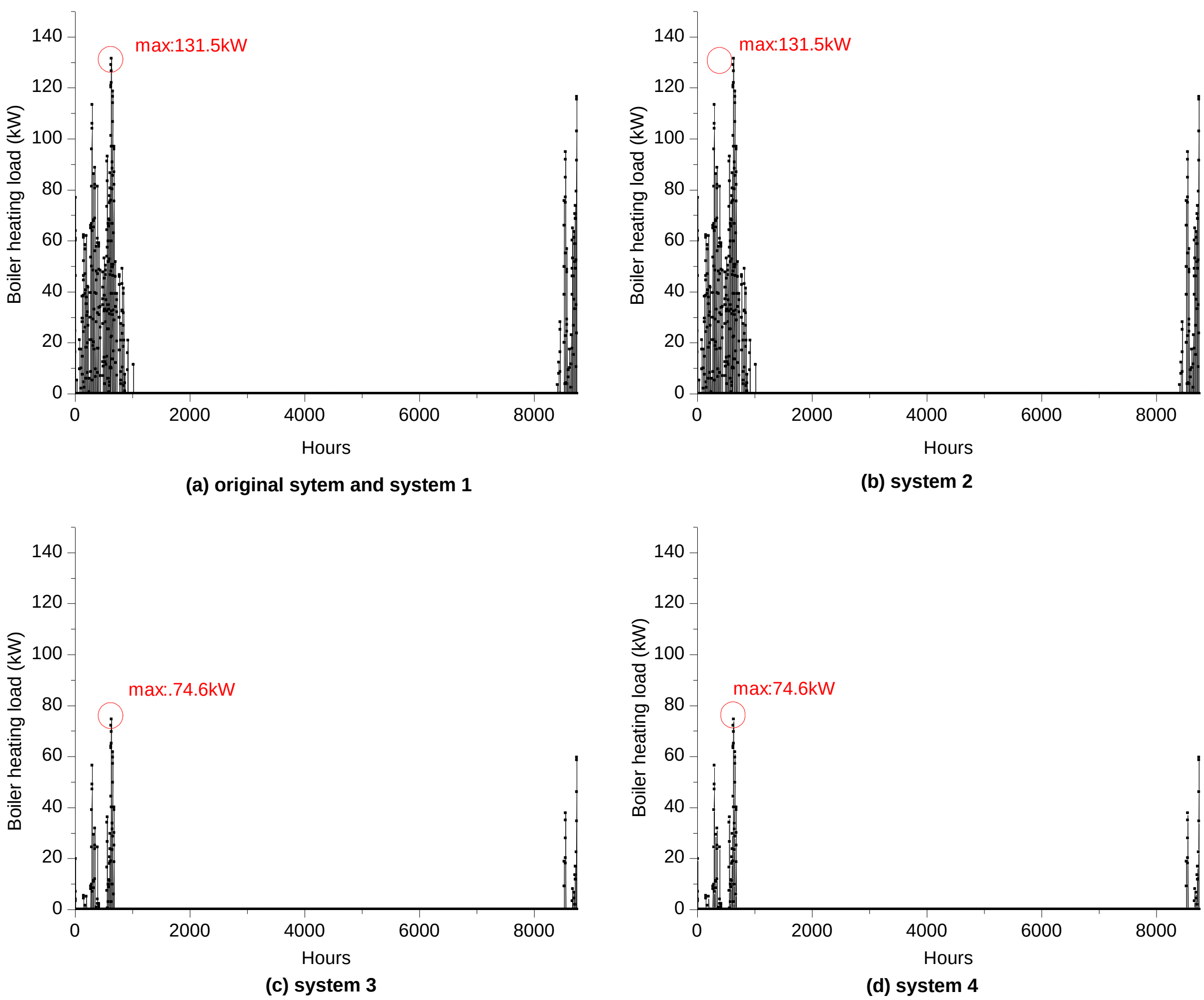

Fig.18 Boiler heating load in different tri-generation system

Based on the simulation results in Fig.17, the hourly boiler heating load in different tri-generation system is shown in Fig.18. The maximum heating load for original system, system 1 and system 2 is 131.5 kW and the annual total heating load is 1.36E+04 kWh; while the maximum heating load for system 3 and 4 is 74.5kW and the annual total heating load is 2.25E+03 kWh. From this aspect, system 3 and 4 are superior in fully recovering the thermal energy in high temperature fuel gas, resulting in decreasing of boiler re-heating load. The annual boiler heating saving of system 3 and 4 was calculated to be 1.14E+04 kWh/a.

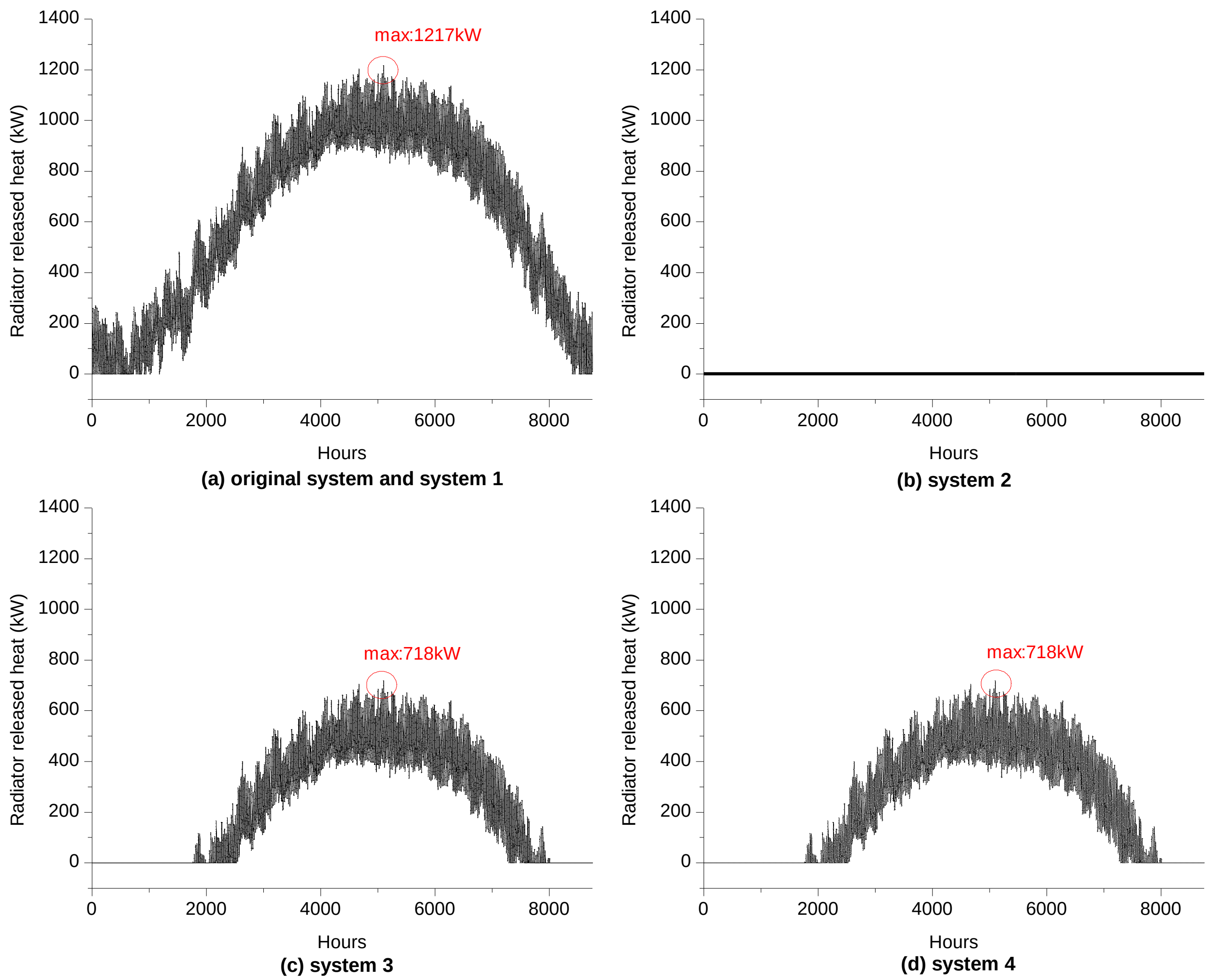

Fig.19 Radiator load in different tri-generation system

Fig.19 shows the hourly radiator load in different tri-generation system. The energy wasted by system 1 is the largest among the four systems with the peak radiator load of 1217 kW and annual load of 5.05E+06 kWh. It is because the jacket water system remains unchanged in system 1, only the thermal energy in 180℃ fuel gas is used but the rest in jacket water is wasted. System 2 shows its biggest advantage in minimizing thermal energy waste by radiator. There is no energy released in system 2 because its flexibility in regulation the heating demand as mentioned above. System 3 and 4 shows the same trend on radiator performance. The peak load of the radiator is 718kW and annual energy released is 1.85E+06, 63.3% less than the original system. The heat wasted by radiator focuses in hot months when the heat provided by the 200℃ fuel gas is larger than the requirement.

5.2.3 System efficiencies

Fig.20 illustrates the thermal efficiency of different tri-generation systems. The annual average thermal efficiencies of the original system and four tri-generation systems are 20.8%, 22.2%, 35.9%, 38.3% and 38.3%, respectively. So the efficiency of system 3 and 4 is the highest, while system 1 is the lowest. Compared with the original system, the efficiency of system 1 is improved but still very limited especially in hot months because the same huge amount of heat is released by radiator. The average efficiency of system 3 and 4 is the highest, but it fluctuates more signifinately than system 2 in a year, especially in summer. The declination can be observed because of the energy wasted by radiator. In contrast, the efficiency fluctuation of system 2 is smaller as no energy released by radiator all year round, but the annual refrigeration output is less than system 3 and 4 because of the lower COP of absorption chiller. In sum, by adopting the tri-generation scheme, the problem of low thermal efficiency of the original system can be solved effectly with the maximum improvement of 17.5%.

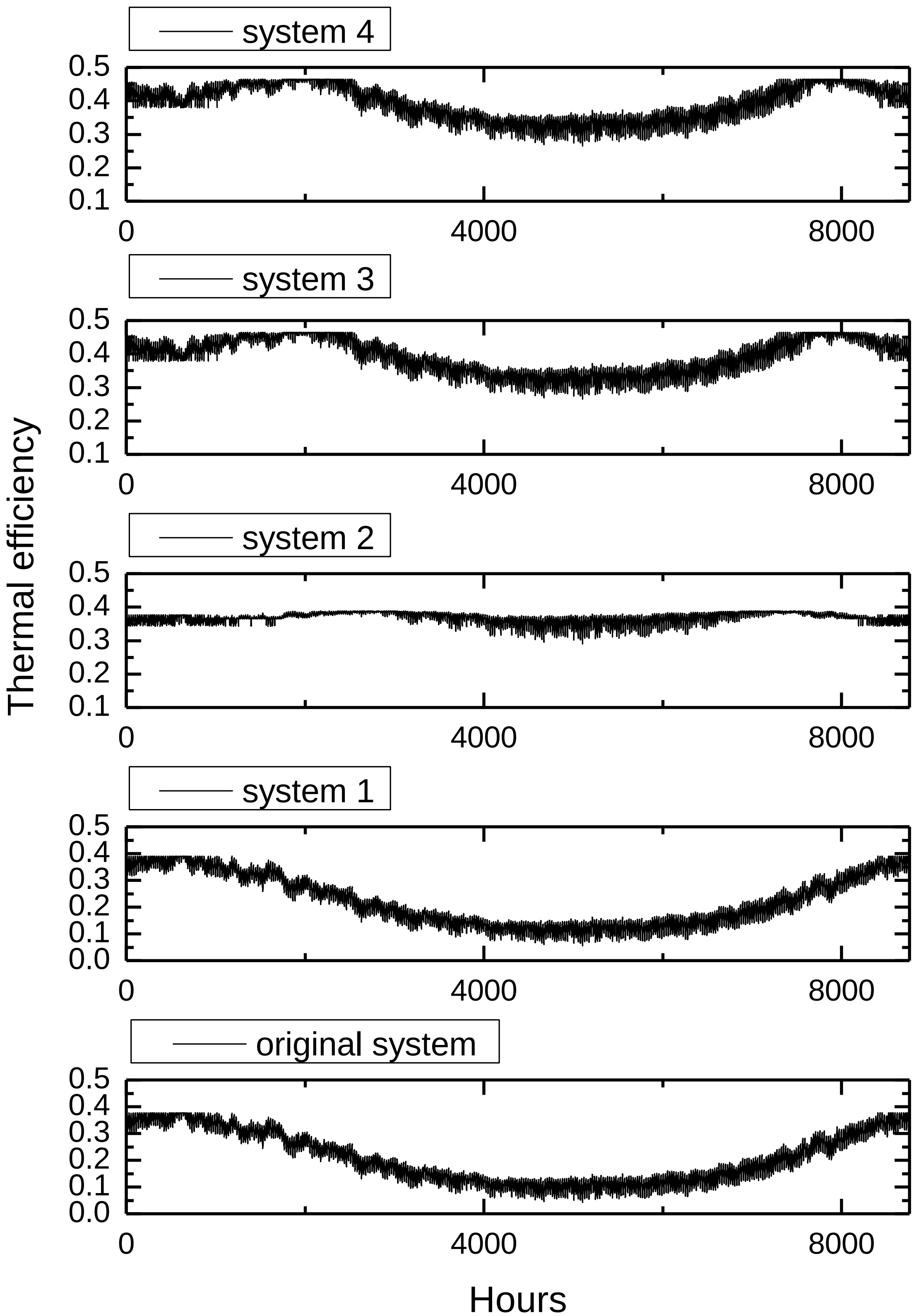

Fig. 20 Thermal efficiency in different tri-generation system

5.3 Economic analysis

Economic analysis was conducted among the four proposed tri-generation systems. The cost of converting the CHP system into a tri-generation system consists of the initial investment (including equipment initial investment and system installation fee), operational and maintenance costs. The main equipment in the retrofitting scheme include: absorption chiller and cooling tower.

As it is shown in Fig.13, the capacity of the absorption chiller is very different among different system, which has the major influence on its initial investment. According to a market study in the reference, the investment cost for the thermal absorption chillers with various cooling capacity is shown as Fig.21 [25]. It can be seen that, the specific investment per kilowatt is decrease with the increase of the chiller cooling power. Besides, the manufacture cost per kilowatt of single effect absorption chiller is only 70% of the double effect absorption chiller [26]. Moreover, the simultaneous cooling and heating absorption chiller is about 10% more expensive than the single cooling unit based on the quotation obtained by chiller manufactories. Based on the above three aspects, the initial investment of absorption chiller can be estimated to be 490 €/kW (system 1, single effect), 140 €/kW (system 2, single effect), 220 €/kW (system 3, double effect, simultaneous heating and cooling) and 200 €/kW (system 4, double effect), as the cooling capacity of the chiller are 47kW, 826kW, 727kW and 727kW, respectively, in the four systems. By converting the price into Hong Kong Dollar (HKD), the total investment of the four absorption chillers is 2.07E+05 HKD, 1.04E+05 HKD, 1.44E+05 HKD and 1.31E+05 HKD, respectively.

The investment of the cooling tower was estimated by the quotation from ONS (overnite supply) [27]. As the cooling capacity of the cooling tower was simulated to be 117 kW (system 1), 2014 kW (system 2), 1339 kW (system 3) and 1339 kW (system 4), the investment of cooling tower can be estimated to be 3.59E+04 HKD (system 1), 3.88E+05 HKD (system 2), 2.59E+05 HKD (system 3), 2.59E+05 HKD (system 4).

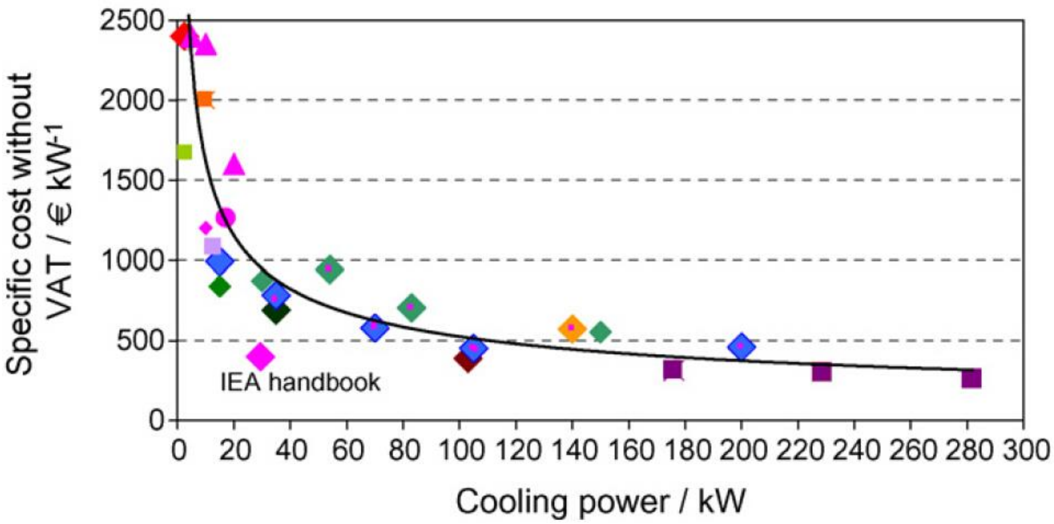

Fig. 21 Absorption chiller investment with different cooling power

Major unknowns are the installation fee of the initial investment, which varies greatly from region to region and site layout. Therefore, two calculations were conducted with different cost assumptions for installation and integration. The first case was done with extremely low installation and integration cost. The installation cost was selected to be 10% of the total investment of absorption chiller and cooling tower. Considering the modification workload of different systems, the integration cost is estimated to be 10% (system 1), 12% (system 2) and 17% (system 3 and 4). The secondary case was done with high installation and integration cost. The installation cost was 25% and the integration cost is estimated to be 18% (system 1), 20% (system 2) and 25% (system 3 and 4).

Annual expenses include the maintenance and operational costs. For maintenance costs, some companies offer constant cost maintenance and repair contracts: the costs vary between 0.5% for large chiller (up to 700 kW) to 3% for small power machine. Repair contracts are even more expensive with 2% for large chiller and up to 12% for a 100 kW machine [25]. In the calculation, 2.5% investment was used for estimating the annual total maintenance cost for system 2, 3 and 4; while 15% was used for estimating the maintenance cost for the small system 1. The operational costs for electrical pumps were set to be 2%. The operational costs for absorption chiller and cooling tower refer to the technical parameters provided by the potential manufactories. The detailed operation and maintenance costs for each tri-generation system are listed in Table 5.

Table 5. Operation and maintenance costs for each tri-generation system (HKD/a)*

System	Items	Absorption chiller	Cooling tower	Chilled/cooling water pump	Maintenance	Total cost
System 1	Power(kW)	0.21	0.372	5.59E+03	3.11E+04	4.77E+04
	Cost(HKD)	1.75E+03	9.29E+03			
System 2	Power(kW)	3.7	3.728	3.34E+04	2.60E+04	1.83E+05
	Cost(HKD)	3.08E+04	9.31E+04			
System 3	Power(kW)	3.25	3.728	4.14E+04	3.60E+04	1.66E+05
	Cost(HKD)	2.70E+04	6.20E+04			
System 4	Power(kW)	3.25	3.728	3.82E+04	3.27E+04	1.60E+05
	Cost(HKD)	2.70E+04	6.20E+04			

*The annual operation hours are 8320.

The annual benefit of the tri-generation system comes from the cooling electricity saving and boiler heating saving. Suppose the integrated COP of the chiller for the original central air-conditioning system is 4.5 and the market price for electricity is 1.0 HKD/kWh, the cooling electricity saving was calculated to be 8.69E+04 HKD/a (system 1), 9.55E+05 HKD/a (system 2), 1.11E+06 HKD/a (system 3) and 1.11E+06 HKD/a (system 4). Besides, the annual boiler heating saving of system 3 and 4 was calculated to be 1.14E+04 HKD/a.

Then, the dynamic payback period calculation approach considering the time value of money is adopted to measure how long it takes for the retrofitting scheme to 'pay for itself'. The cash flow diagram was used in economic analysis, where outflows are recorded as negative and inflows are positive. The net cash flow value (*NCF*) refers to the difference between cash inflow value (*CIV*) and cash outflow value (*COV*).

$$NCF = CIV - COV \quad (18)$$

Because of the time value of money, the net cash flow in each year is converted to net present value. The total net present value (*NPV*) can be calculated as:

$$NPV = \frac{NCF}{(1+i)^n} \quad (19)$$

Where *NCF* is the net cash flow value each year (HKD/year); *n* is the equipment lifetime (years) and *i* is the annual real interest rate (the discount rate), which can be calculated as:

$$i = \frac{i' - f}{1 + f} \quad (20)$$

Where *i* is the interest rate, *i'* nominal interest rate, and *f* is the annual inflation rate. Nominal interest rate is 6.5% and the inflation rate is 4.5%, giving an annual real interest rate of 1.9% in this study.

The payback time (*PBT*) can be expressed as follows:

$$PBT = (N-1) + \frac{\left|\sum_{0}^{n-1} NPV_n\right|}{NPV_n} \quad (21)$$

Where, N represents the year when net present value (NPV) becomes positive.

By adopting the payback period calculation methods above, the summary of economic analysis results is shown as Table 6. It shows that the payback period for system 4 is the shortest, which is between 2 to 3 years, showing a very good economic benefit of the retrofitting scheme.

Table 6 Economic analysis results of four tri-generation system

		System 1	System 2	System 3	System 4
Initial investment (HKD)	Absorption chiller	2.07E+05	1.04E+06	1.44E+06	1.31E+06
	Cooling tower	3.59E+04	3.88E+05	2.59E+05	2.59E+05
	Installation (low)	3.65E+04	2.43E+05	3.74E+05	3.45E+05
	Installation (high)	1.05E+05	6.43E+05	8.49E+05	7.84E+05
Maintenance fee (HKD/a)	Maintenance fee	3.11E+04	2.60E+04	3.60E+04	3.27E+04
	Operation fee	1.66E+04	1.57E+05	1.31E+05	1.27E+05
Annual benefits (HKD/a)	Cooling saving	8.69E+04	9.55E+05	1.10E+06	1.10E+06
	Boiler heating saving	0.00E+00	0.00E+00	1.14E+04	1.14E+04
Payback (year)	Payback (low)	7.74	2.23	2.26	2.07
	Payback (high)	9.81	2.78	2.79	2.56

5.4 Comparison of the four tri-generation systems

The comparisons among the four tri-generation systems are listed in Table 7. It can be seen that system 3 and 4 are superior in terms of thermal efficiency and payback period. However, the stability of system 3 is worse than system 4, because the heating demand of the digester may not be met once the absorption chiller fails. While system 4 achieves good reliability by regulating the flue gas distribution through a three-way valve. Once the absorption chiller fails, the digester heating would not break down. Therefore, from the view point of annual benefit, thermal efficiency, stability, control requirement and payback period, system 4 is the optimal tri-generation retrofitting scheme.

Table 7 Comparison of the four tri-generation systems

	System 1	System 2	System 3	System 4
Annual benefit (HKD/a)	8.69E+04	9.55E+05	1.11E+06	1.11E+06
Average thermal efficiency	22.2%	35.9%	38.3%	38.2%
Initial investment (HKD)	2.80E+05	1.67E+06	2.07E+06	1.91E+06
System stability	high	medium	low	medium
modification workload	small	small	large	large
Requirement of control	low	medium	high	medium

6. Conclusions

The paper presents a feasibility study on converting a combined heat and power system (CHP) to a tri-generation system for sewage treatment plant and thereby overcoming the problems associated with CHP application in subtropical regions. Four possible tri-generation systems designs were explored and the performances investigated by hourly dynamic simulations before the economic analysis was conducted. The highlight conclusions are as follows:

1) The thermal efficiency improves significantly by adding an absorption chiller for producing cooling by using massive excess heat from CHP, especially in summer. The annual average thermal efficiency improves from 20.8% to 35.9%, 38.3% and 38.3% for system 2, 3 and 4.

2) By maximizing the utilization of energy in the high temperature fuel gas, the heating load of the auxiliary boiler decreases by 1.14E+04 kWh for system 3 and 4.

3) The double effect absorption chiller driven by bypass of 450℃ flue gas was found to be the optimal retrofitting scheme for its highest thermal efficiency, shortest payback period, better stability and easier control. The payback time was calculated to be 2~3 years taking into account the initial investment, operational costs and maintenance cost.

The study results provide references for future designs of converting an existing CHP system to a tri-generation system in subtropical areas aimed at improving energy utilization.

7. Acknowledgments

This research was financially supported by the Shatin Sewage Treatment Works in Hong Kong and the Hong Kong polytechnic university research institute for sustainable urban development (RISUD).

References

[1] http://www.dsd.gov.hk/Documents/AnnualReports/0102/03e.html

[2]http://www.dsd.gov.hk/EN/Sewerage/Environmental_Consideration/Renewable_Energy/index.html

[3] http://www.epa.gov/chp/markets/wastewater.html

[4] Tassou, S. A. (1989). Combined heat and power generation at a sewage treatment plant—a feasibility study. Heat Recovery Systems and CHP, 9(2), 169-174.

[5] http://www.cogeneration.com.ua/img/zstored/J420V21_en.pdf

[6] De Sa, A., & Al Zubaidy, S. (2011). Gas turbine performance at varying ambient temperature. Applied Thermal Engineering, 31(14), 2735-2739.

[7] Al-Ibrahim, A. M., & Varnham, A. (2010). A review of inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia. Applied thermal engineering, 30(14), 1879-1888.

[8] Najjar, Y. S. (1996). Enhancement of performance of gas turbine engines by inlet air cooling and cogeneration system. Applied Thermal Engineering, 16(2), 163-173.

[9] Wu DW, Wang RZ. Combined cooling, heating and power: a review. Prog Energy Combust 2006;32:459-95.

[10] Jradi, M., & Riffat, S. (2014). Tri-generation systems: Energy policies, prime movers, cooling technologies, configurations and operation strategies. Renewable and Sustainable Energy Reviews, 32, 396-415.

[11] Chicco C, Mancarella P. Tri-generation primary energy saving evaluation for energy planning and policy development. Energy Policy 2007;35:6132-44.

[12] Cho, H., Mago, P. J., Luck, R., & Chamra, L. M. (2009). Evaluation of CCHP systems performance based on operational cost, primary energy consumption, and carbon dioxide emission by utilizing an optimal operation scheme. Applied Energy, 86(12), 2540-2549.

[13] Maraver, D., Sin, A., Royo, J., & Sebastián, F. (2013). Assessment of CCHP systems based on biomass combustion for small-scale applications through a review of the technology and analysis of energy efficiency parameters. Applied Energy, 102, 1303-1313.

[14] Al-Sulaiman FA, Hamdullahpur F, Dincer I. Greenhouse gas emission and exergy assessments of an integrated organic Rankine cycle with a biomass combustor for combined cooling, heating and power production. Applied Thermal Engineering 2011,31:439-46.

[15] Maraver, D., Sin, A., Sebastián, F., & Royo, J. (2013). Environmental assessment of CCHP (combined cooling heating and power) systems based on biomass combustion in comparison to conventional generation. Energy, 57, 17-23.

[16] Huang, Y., Wang, Y. D., Rezvani, S., McIlveen-Wright, D. R., Anderson, M., & Hewitt, N. J. (2011). Biomass fuelled trigeneration system in selected buildings. Energy Conversion and Management, 52(6), 2448-2454.

[17] Lian, Z. T., Chua, K. J., & Chou, S. K. (2010). A thermoeconomic analysis of biomass energy for trigeneration. Applied Energy, 87(1), 84-95.

[18] Al-Sulaiman, F. A., Dincer, I., & Hamdullahpur, F. (2012). Energy and exergy analyses of a biomass trigeneration system using an organic Rankine cycle. Energy, 45(1), 975-985.

[19] Huang, Y., Wang, Y. D., Rezvani, S., McIlveen-Wright, D. R., Anderson, M., Mondol, J. & Hewitt, N. J. (2013). A techno-economic assessment of biomass fuelled trigeneration system integrated with organic Rankine cycle. Applied Thermal Engineering, 53(2), 325-331.

[20] http://www.ge-energy.com/about/index.jsp

[21] http://adbioresources.org/wp-content/uploads/2013/06/59-80_chapter5_v41.pdf

[22] http://www.v3e.ch/assets/files/news/2011/The%20new%20J624_PGA_2011.pdf

[23] https://www.clarke-energy.com/service/upgrades-repair-and-overhaul/

[24]http://www.coned.com/steam/Steam_Seminar_May_2008/Joseph%20Rubino_Steam%20Cooling%20Systems%20Explained.pdf

[25] Eicker, U., & Pietruschka, D. (2009). Design and performance of solar powered absorption cooling systems in office buildings. Energy and Buildings, 41(1), 81-91.

[26] http://www.trane.com/commercial/uploads/pdf/865/ctv-trt-001-en.pdf

[27] http://www.overnitesupply.com/coolingtowers.aspx