Desalination and Water Treatment
www.deswater.com

51 (2013) 1915–1921
February

Taylor & Francis
Taylor & Francis Group

doi: 10.1080/19443994.2012.694232

The performance of a temperature cascaded cogeneration system producing steam, cooling and dehumidification

Aung Myat[a,*], Kyaw Thu[b], Young-Deuk Kim[b], Ng Kim Choon[a]

[a]*Department of Mechanical Engineering, National University of Singapore, 9, Engineering Drive 1, Singapore 117576, Singapore*
Email: aung_myat@ices.a-star.edu.sg
[b]*Water Desalination and Reuse Centre, King Abdullah University of Science and Technology, Thuwal 23955-6900, Saudi Arabia*

Received 27 February 2012; Accepted 10 May 2012

ABSTRACT

This paper discusses the performance of a temperature-cascaded cogeneration plant (TCCP), equipped with an efficient waste heat recovery system. The TCCP, also called a cogeneration system, produces four types of useful energy—namely, (i) electricity, (ii) steam, (iii) cooling and (iv) dehumidification—by utilizing single fuel source. The TCCP comprises a Capstone C-30 micro-turbine that generates nominal capacity of 26 kW of electricity, a compact and efficient waste heat recovery system and a host of waste-heat-activated devices, namely (i) a steam generator, (ii) an absorption chiller, (iii) an adsorption chiller and (iv) a multi-bed desiccant dehumidifier. The performance analysis was conducted under different operation conditions such as different exhaust gas temperatures. It was observed that energy utilization factor could be as high as 70% while fuel energy saving ratio was found to be 28%.

Keywords: Temperature cascaded; Co-generation; Waste heat recovery; Energy utilization factor

1. Introduction

The conventional method of generating electricity is through a centralized power station where a primary fuel source (solid, liquid or gaseous) is burned and the exhaust gases are purged into the ambient. Such power plants are usually designed with the economy of scale, and the boiler units have capacities up to 1,200 MW. Despite the advanced heat recovery with the working fluids and the high pressure technology of boilers, the best overall energy efficiency of these power plants is below the 50% margin. More than half of the fuel energy burned at power plants is exhausted to the ambient as flue gases.

The 50%-efficiency "barrier" of power stations can be overcome with the implementation of temperature-cascaded cogeneration concept: Using this concept, a quantum rise in the overall plant efficiency is expected, typically 70% or higher. Recent advances in the technologies of prime movers such as gas engines and micro-turbines, heat-activated thermodynamic cycles for the production of heating and cooling as well as the recent availability of a variety of fuel types have made distributed cogeneration a reality for the past decades [1–3]. A survey of the literature shows that the cogeneration concept has been implemented successfully since the 1960s [1–15], with many config-

*Corresponding author.

Presented at the International Conference on Desalination for the Environment, Clean Water and Energy, European Desalination Society, 23–26 April 2012, Barcelona, Spain

urations in terms of the heat-to-power ratios. Simulation and optimization of these power plants have been studied earlier [16–19]. Despite the inherent benefits of cogeneration, a suitable thermodynamic methodology or tool for evaluating the efficacy of a cogeneration plant other than the overall energy utilization factor (EUF) is yet to be evolved. In this paper, we proposed a configuration to optimize the waste heat recovery from exhaust gas emanating from prime movers.

2. System description and mathematical modelling

2.1. System description

A temperature-cascaded cogeneration plant (TCCP) comprises a Capstone C-30 micro-turbine, a set of waste heat recovery cross-flow heat exchanger, a steam generator (SG), an absorption chiller (AB), an adsorption chiller (AD) and a desiccant dehumidifier (DD). The TCCP produces electricity, steam, cooling and dehumidification. The schematic layout of the proposed TCCP is shown in Fig. 1. The exhaust gas emanating from the Capstone C-30 micro-turbine is diverted to a series of cross-flow heat exchangers comprising a copper tube and aluminium fins. The first two heat exchangers are employed as the steam-generation cycle, whilst the remaining three heat exchangers are utilized to produce hot water of different temperatures so as to drive the cooling facility and dehumidification facility. The exhaust gas at 285℃ drives the steam-generation cycle to produce superheated steam at 250℃ at 10 bars and leaves the steam generator at 235℃. The exhaust gas leaving the steam generator is then fed to the heat exchanger to produce hot water in the range of 70–95℃. The heat exchanger is designed to maintain the exhaust gas temperature at 185℃. The hot water produced serves as the main heat source to drive the Li–Br absorption chiller, which can produce 7 kW (2 refrigeration ton) of cooling. The exhaust gas is then subsequently fed to the next heat exchanger to produce hot water, which is the driving heat source for silica gel–water adsorption cooling system. The exhaust gas leaves the heat exchanger at 130℃ and is diverted to the heat exchanger to produce hot water in a temperature range of 65–85℃. The hot water produced is utilized as the main heat source for the multi-bed desiccant dehumidification system [20]. Thus, the proposed cogeneration system produces four types of useful energy, utilizing a single energy source.

2.2. Mathematical modelling

The mathematical model for this cogeneration system based on the second law of thermodynamics is presented in this section. The numerical model for each of the waste-heat-activated component in the TCCP is formulated to capture the transient

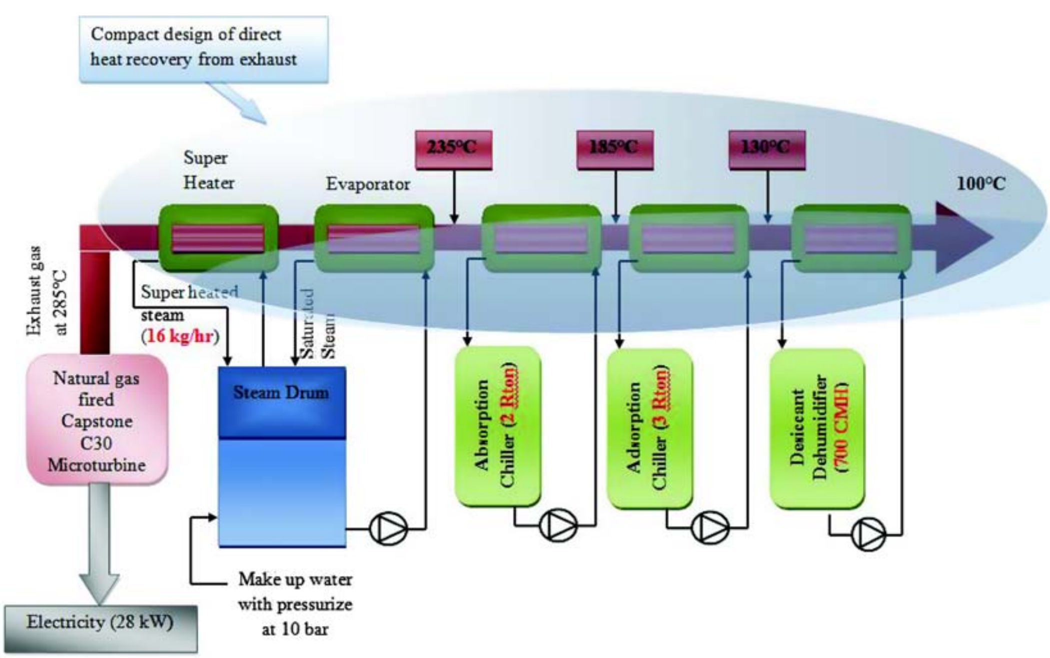

Fig. 1. The schematic layout diagram of temperature cascaded cogeneration system.

characteristics of the corresponding devices. The exhaust gas temperature and the electricity generated from the Capstone C-30 micro-turbines are experimentally investigated. With the available data of exhaust gas temperature and electricity generation capacity, the waste-heat-activated devices are sized to optimize the energy recovery. The generic mathematical modelling for the TCCP is developed to estimate the performance of the plant. The heat source or heat supply to micro-turbine can be expressed as

$$Q_{\mathrm{in,cogen}} = \mathring{m}_f x C_v \quad (1)$$

Energy gas recovered from the exhaust gas can be represented by the following expression:

$$Q_{\mathrm{rec}} = \mathring{m}_{\mathrm{exh}} x Cp_{\mathrm{exh}}(T_{\mathrm{exh,in}} - T_{\mathrm{exh,o}}) \quad (2)$$

Since the cogeneration system comprised a host of waste heat recovery devices, namely, (i) a steam generator, (ii) an absorption chiller, (iii) an adsorption chiller and (iv) a multi-bed desiccant dehumidifier, the useful energy produced from each of these devices can be expressed as follows:

From the steam generator:

$$Q_{\mathrm{steam}} = \mathring{m}_{\mathrm{steam}} x h_g(T, P) \quad (3)$$

From the absorption chiller:

$$Q_{\mathrm{cooling,AB}} = \frac{\mathrm{d}M_{\mathrm{chi}}}{\mathrm{d}t}, Cp_{\mathrm{chi}}(T_{\mathrm{chi,in}} - T_{\mathrm{chi,o}}) \quad (4)$$

From the adsorption chiller:

$$Q_{\mathrm{cooling,AD}} = \frac{\mathrm{d}M_{\mathrm{chi}}}{\mathrm{d}t} Cp_{\mathrm{chi}} \int_0^{t_{\mathrm{cyc}}} \frac{T_{\mathrm{chi,in}} - T_{\mathrm{chi,o}}}{t_{\mathrm{cyc}}} \mathrm{d}t \quad (5)$$

From the multi-bed desiccant dehumidifier:

$$Q_{\mathrm{deh}} = \frac{\mathrm{d}M_{air}}{\mathrm{d}t} h_{fg} \int_0^{t_{\mathrm{cyc}}} \frac{\omega_{\mathrm{in}} - \omega_{\mathrm{o}}}{t_{\mathrm{cyc}}} \mathrm{d}t \quad (6)$$

The performance of the TCCP can be determined by the overall efficiency of the plant—the energy utilization factor (EUF). EUF can be expressed as the ratio of total useful energy produced by the cogeneration plant to the energy supplied to the micro-turbine:

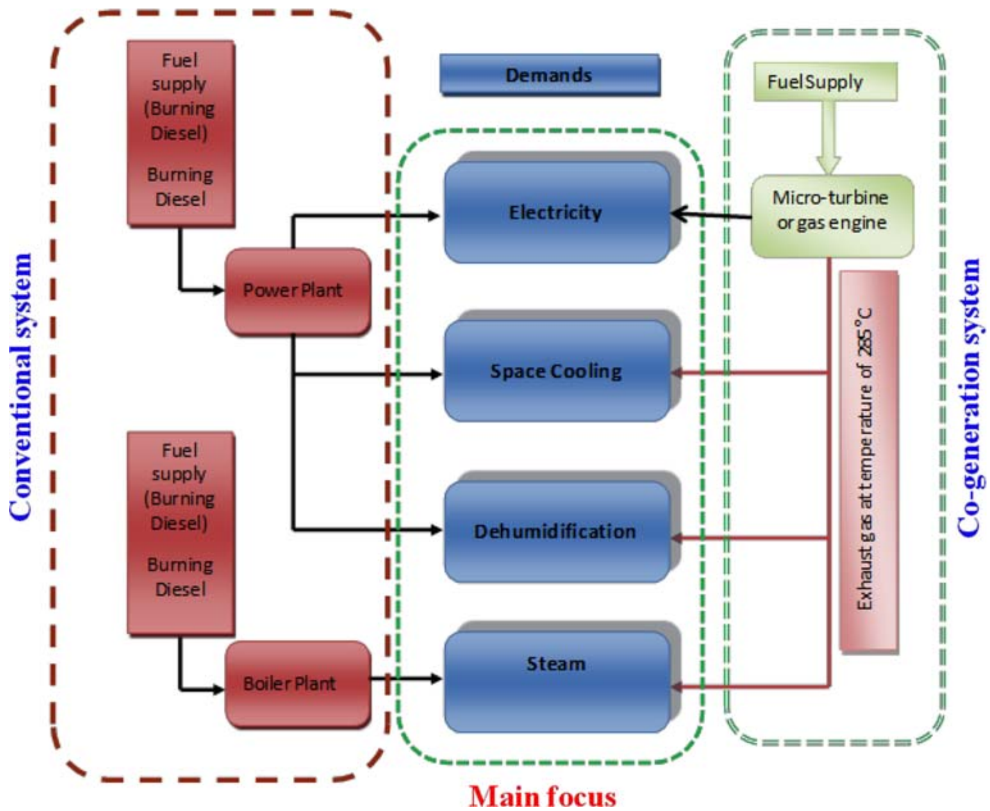

Fig. 2. The comparison between a conventional power plant and the temperature-cascaded cogeneration plant.

Table 1
The nominal capacity of each of the devices equipped in the TCCP system

Component	Type of energy	kW	Refrigeration ton
C-30 Micro-turbine	Electricity	28	NA
Waste heat recovery steam generator	Steam	15	NA
Absorption chiller	Cooling	7.04	2
Multi-bed adsorption chiller	Cooling	7.04	2
Multi-bed desiccant dehumidifier	Dehumidification	7	NA

$$\mathrm{EUF} = \frac{Q_{\mathrm{steam}} + Q_{\mathrm{cooling,AB}} + Q_{\mathrm{cooling,AD}} + Q_{\mathrm{deh}}}{Q_{\mathrm{in,cogen}}} \tag{7}$$

The fuel energy saving (FESR) ratio can be represented as follows:

$$\mathrm{FESR} = 1 - \frac{Q_{\mathrm{in,cogen}}}{Q_{\mathrm{in,conv}}} \tag{8}$$

where $Q_{\mathrm{in,conv}}$ is the energy required by a conventional plant that produces the same energy demand compared to the TCCP. Fig. 2 shows the comparison between a conventional plant and the cogeneration plant.

The energy supplied to the conventional plant, $Q_{in,\mathrm{conv}}$, can be represented as the sum of fuel supplied to the power plant and the boiler:

$$Q_{\mathrm{in,conv}} = \left(\mathring{m}_{f,\mathrm{power}} + \mathring{m}_{f,\mathrm{Boiler}}\right) x C_v \tag{9}$$

3. Result and discussion

The governing equations are solved by the fifth-order Gear's backward differential method of the DIVPAG subroutine of IMSL Fortran library subroutines. The iterative scheme uses a double precision format that has a tolerance of 1×10^{-6}. The nominal capacity of each of the waste-heat-activated systems is given in Table 1 and the numerical calculations were performed for the performance analysis and entropy generation for each system was also computed. The consistency of the simulation model was checked using a suitable range of parameters that is anticipated of the TCCP operation domain. The hot water source temperatures are varied from 65 to 85℃ whilst the cooling water and the chilled water temperatures are kept at about 29 ± 1℃ and 12 ± 1℃, and the cooling output and overall performance are monitored. Fig. 3 shows the experimental exhaust gas data at assorted electricity load demand and the fuel consumption of micro-turbine and the generated electricity at different exhaust gas temperatures. It is observed that both fuel consumption and electricity generation are linearly increased with the increase in the exhaust gas temperature. The predicted temperature profiles of the exhaust gas temperature emanating from each of the waste heat recovery heat exchanger units are illustrated in Fig. 4 while the micro-turbine outlet exhaust gas temperature is about 285℃.

Fig. 5 shows the performance parameters such as temperature profiles and the steam production of waste-heat-driven steam generator (WHSG) while the waste heat source (exhaust gas temperature emanating from the micro-turbine) is maintained at 285℃. It is observed that the superheated steam at 250℃ at 10 bars is produced from the WHSG whilst the steam production is found to be about 0.044 kg/s (15.84 kg/h). The exhaust gas emanating from the steam-generation cycle, at 240℃, is then channelled to the waste heat extraction heat exchanger WHR-HE-01 coupled with the absorption cycle to produce the hot water to fire the absorption cycle. The transient profiles of outlet water temperature for the absorber, the condenser, the evaporator and the generator are illustrated in Fig. 6. It is indicated from Fig. 6 that thermal mass heat or sensible heating of the system takes a few hundred seconds before it reaches the steady state. The performance of two-bed regenerative adsorption chiller is presented in Fig. 7. The outlet exhaust gas stream, at about 210 ± 2℃, is diverted to WHR-HE-02 coupled with the adsorption cooling cycle to produce the hot water to fire the AD cycle, which operates in a batch manner. The predicted performance of the AD cycle is shown in Fig. 7.

The total mass of the adsorbent has been fixed at 78 kg, and the half cycle time is fixed at 600 s while the switching time is 40 s. It is observed that the cycle steady state is reached within 4 cycles of operation. Fig. 7 illustrates the temporal temperature profiles of inlet hot water and outlet heat transfer fluid. The cycle average outlet coolant temperature for the adsorber is 3.5 to 4℃ higher, that of inlet temperature about 5℃ and that of chilled water is about 5℃ while the cooling capacity obtained is 7 kW. The temperature pro-

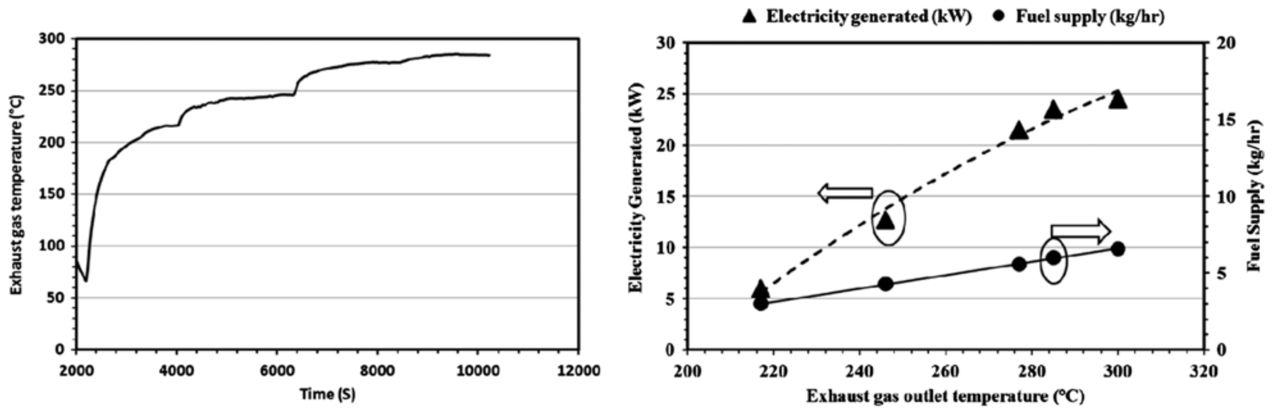

Fig. 3. The performance of the micro-turbine contained in the temperature cascaded cogeneration plant.

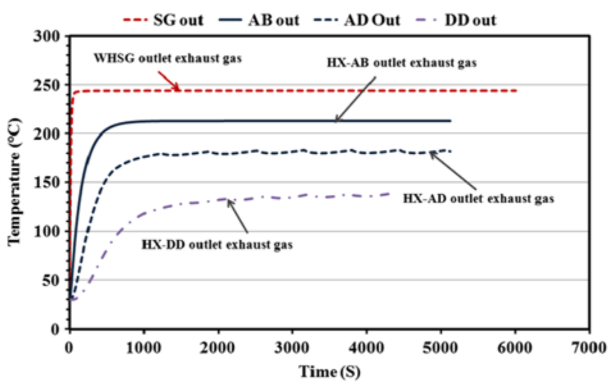

Fig. 4. Temperature–time history of exhaust gas from each of the waste heat recovery heat exchanger unit.

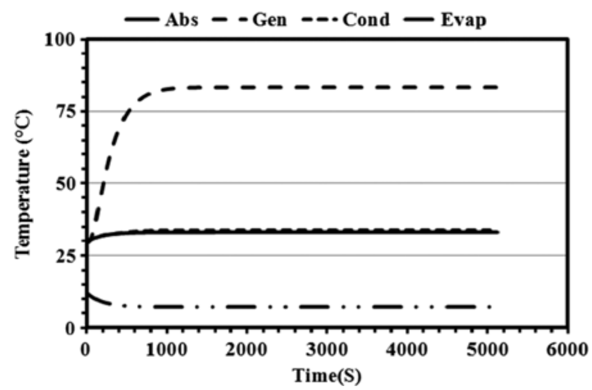

Fig. 6. The performance of LiBr–water absorption chiller.

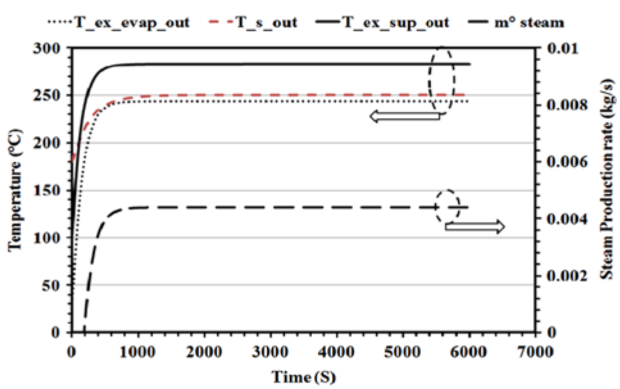

Fig. 5. The performance of the waste heat recovery steam generator and its steam production rate.

files of adsorber and desorber beds in the MBDD cycle are shown in Fig. 8. The total mass of adsorbent has been fixed at 20 kg for the corresponding air flow capacity of 750 CMH, and the half cycle time is fixed at 500 s while switching time is 40 s. The adsorber bed and the desorber bed are designed to create a laminar flow with a set velocity of 0.25 m/s. It is observed that the cyclic steady state is reached within 4 cycles of MBDD operation. The inlet and outlet temperature profiles of hot water along with cooling water outlet temperature profile for MBDD cycle are illustrated in Fig. 8. In this context, the ambient conditions are fixed at the maximum humidity condition (32℃ and 95% RH) for Singapore. Owing to exothermic nature of adsorption process, the sensible heat is induced to the outlet air stream with an average temperature increase of 2–3℃. Thus, a cooling coil is installed to reduce the sensible heat induced by the adsorption process. Fig. 9 shows the temporal history of outlet air stream before and after the installation of the cooling coil. It is indicated in Fig. 9 that a considerable amount of sensible heat is induced during hot-to-cold switching. The residual energy in the adsorbent provides the sensible heat needed for the outgoing air stream. The temperature of outlet air stream could rise up to 36℃ without further cooling. Nevertheless, a cooling coil is employed to reduce the

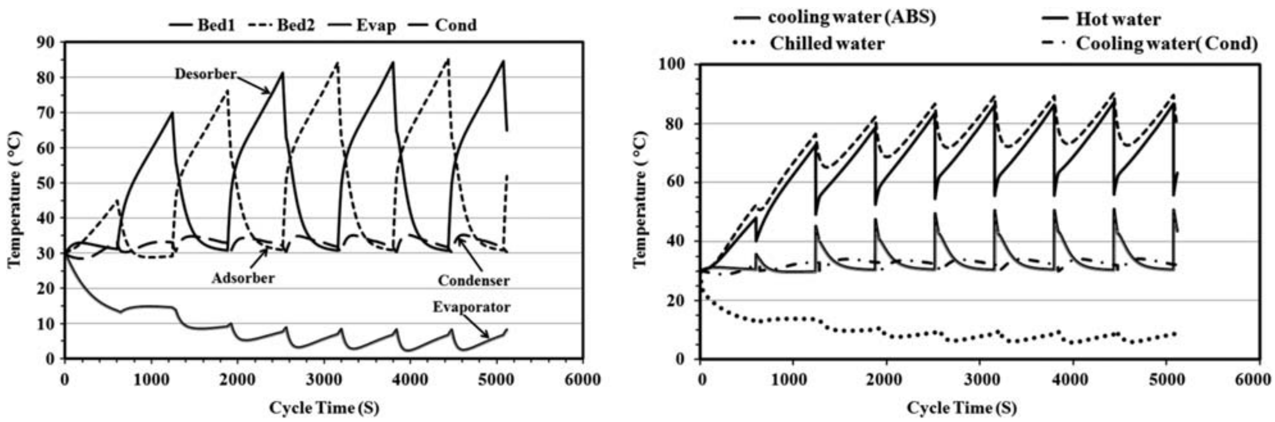

Fig. 7. The performance of two beds regenerative silica gel–water adsorption chiller.

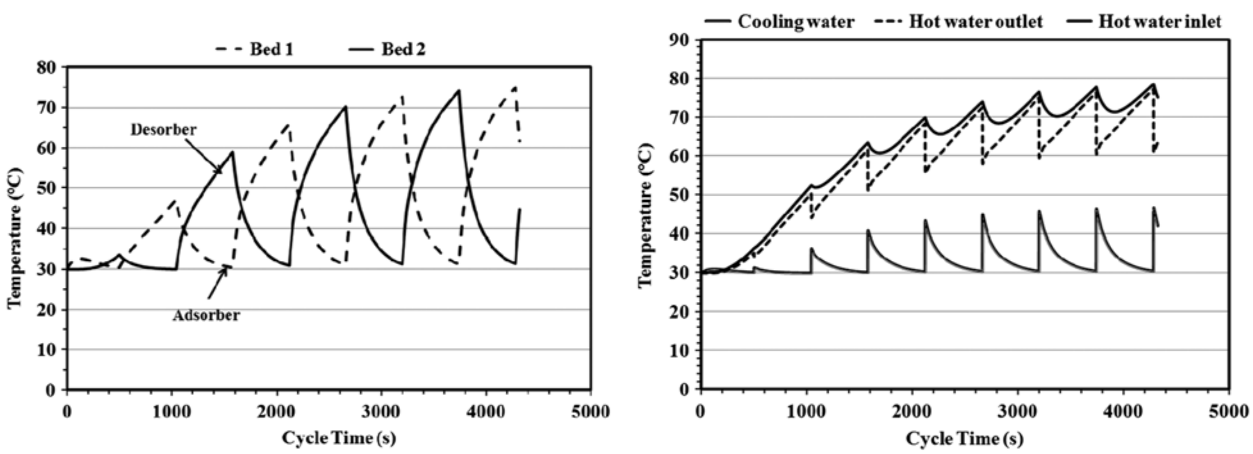

Fig. 8. The performance of multi-bed regenerative silica gel–water desiccant dehumidifier.

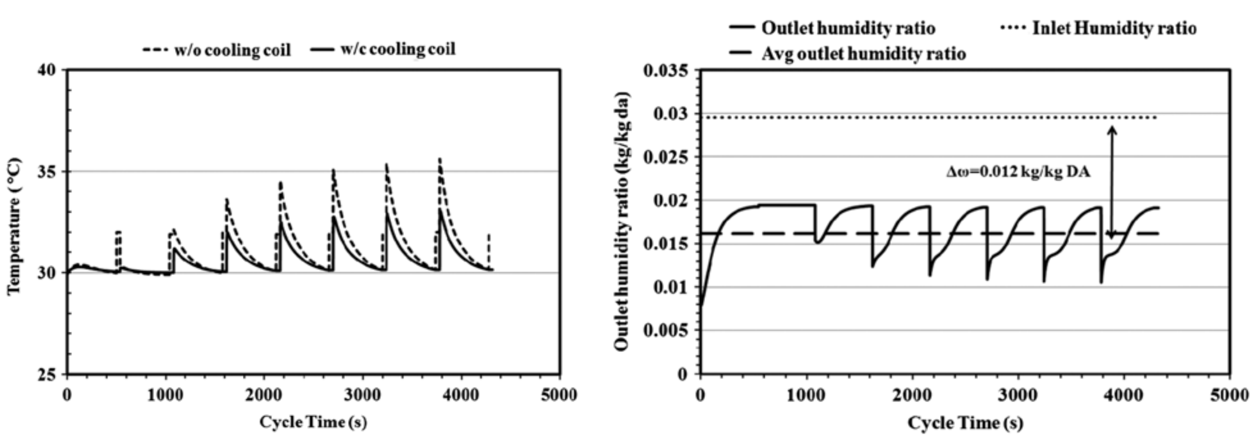

Fig. 9. The performance of multi-bed regenerative silica gel–water desiccant dehumidifier and its moisture removal.

sensible heat (temperature) of outlet air stream before it is sent to AHUs. Inlet humidity ratio, outlet humidity ratio and cycle average outlet humidity ratio are illustrated in Fig. 9. It is observed that the cycle average outlet humidity ratio is about 16 g/kg DA while cycle average moisture removal by MBDD unit is 12.5

Table 2
The nominal capacity of each of the devices equipped with the TCCP

Waste-heat-activated devices	Waste energy recovery (kW)	Useful effect produced (kW)	Coefficient of performance or COP of the system, EUF
Steam generator	18.5	12.9	0.7
Absorption chiller	11.7	6.76	0.58
Adsorption chiller	13.96	6.98	0.5
Desiccant Dehumidifier	14.24	7.12	0.5
Electricity produced	NA	28	NA
Total energy	NA	54.92	
energy input	82.74	NA	0.7

g/kg DA. Thus, the MBDD cycle could remove 40% of moisture present in the ambient air, resulting reduction in AHUs load. Table 2 summarizes the performance of each of the waste-heat-activated devices and the total useful energy demand.

4. Conclusion

The performance of a TCCP—which comprises (i) a steam-generation cycle, (ii) an absorption cooling cycle, (iii) an adsorption cooling cycle and (iv) a multi-bed desiccant dehumidification cycle—has been successfully demonstrated to capture the optimal operation heat source temperature. It is observed that the overall efficiency, or energy utilization factor (EUF), obtained is as high as 70% while the said host of waste-heat-activated devices are arranged in the cascaded configuration. Owing to the effective arrangement of devices, the fuel energy savaging ratio (FESR) attained is 28%, which will lead to a significant reduction in energy costs and carbon dioxide emission.

Acknowledgment

The authors gratefully express the gratitude to Agency of Science, Technology and Research (A*STAR) for their generous financial support for the project (Grant Number R265-000-287-305).

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