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The performance investigation of a temperature cascaded cogeneration system equipped with adsorption desalination unit

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ABSTRACT

This paper presents the performance investigation of a temperature cascaded cogeneration plant, shortly in TCCP, equipped with an efficient waste heat recovery system. The TCCP or cogeneration system produces four types of useful energy namely (i) electricity, (ii) steam, (iii) cooling, and (iv) dehumidification and distilled water by utilizing single energy source. The TCCP comprises a Capstone C30 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 desalination system, and (iv) a multi-bed desiccant dehumidifier. The analysis is performed under different operation conditions such as heat source temperatures, flow rates of heat transfer fluids and chilled water inlet temperatures. The only single heat source for TCCP is obtained from exhaust gas of micro-turbine and it is channeled to a series of waste heat recovery heat exchangers to steam and hot water at different temperatures. Hot water produced by such a compact heat exchangers is the driving heat source to produce steam of 15 kg/h, cooling of 2 Rton, dehumidification of 2 Rton, and distilled water of 0.7 m^3/day. A set of experiments, both part load and full load, of micro-turbine is conducted to examine the electricity generation and the exhaust gas temperature. It is observed that energy utilization factor could achieve as high as 70% while fuel energy saving ratio is found to be 28%.

Keywords: Temperature cascaded; Cogeneration; Waste heat recovery; Energy utilization factor; Adsorption desalination

1. Introduction

The conventional method of generating electricity is a centralized power station where a primary fuel source such as solid, liquid, or gaseous fuel 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 unit sizes up to 1,200 MW. Despite the advanced heat recovery with the working fluids and

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the high pressure technology of boilers, the best overall energetic efficiency of 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 breached with the implementation of temperature cascaded cogeneration concept. In such a concept, a quantum rise in the overall plant efficiency is expected, typically up to 70% or higher. Recent advances in both 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 availability of fuel types have made the distributed cogeneration a reality for the past decades [1–3]. A survey of the literature shows that cogeneration has been implemented successfully in industry since the 1960s [1–15] with many configurations in terms of the heat-to-power ratios. Although some literature regarding the simulation and optimization work have been presented earlier [16–19]. Despite the inherent benefits of cogeneration, there is yet a suitable thermodynamic methodology or tool for evaluating the efficacy of a cogeneration plant other than the overall energy utilization factor (EUF). In this paper, we proposed the configuration to optimize the waste heat recovery from exhaust gas emanating from prime movers.

2. System description and mathematical modeling

2.1. System description

The temperature cascaded cogeneration plant, shortly in 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 the electricity, the steam, cooling, and the dehumidification. The schematic lay out of the proposed TCCP is shown in Fig. 1. As shown in Fig. 1, the exhaust gas emanating from C-30 capstone micro-turbine is diverted to a series of cross flow heat exchangers comprising copper tube and aluminum fins. The first two heat exchangers are employed as the steam generation cycle while the rest of three heat exchangers are utilized to produce hot water with the different temperature ranges so as to drive the cooling, desalination, and dehumidification facilities. The exhaust gas at a temperature of 285°C drives the steam generation cycle to produce super heated steam at temperature of 250°C at 10 bars and leaves the steam generator at 235°C. The exhaust gas leaving from steam generator is then feed to the heat exchanger to produce the hot water at range of temperature 70 to 95°C. The heat exchanger is designed to maintain the exhaust gas temperature of 185°C. The produced hot water serves as the main heat source to drive the Li–Br absorption chiller that produces 7 kW (Two Refrigeration Ton) of cooling. Similarly, the exhaust gas is then subsequently feed to the next heat exchanger to produce hot water, which is the driving heat source for silica gel/water adsorption desalination cycle. The exhaust gas leaves the heat exchanger at the temperature of 130°C and diverted to the heat exchanger to produce hot water at a temperature range from 65 to 85°C. The

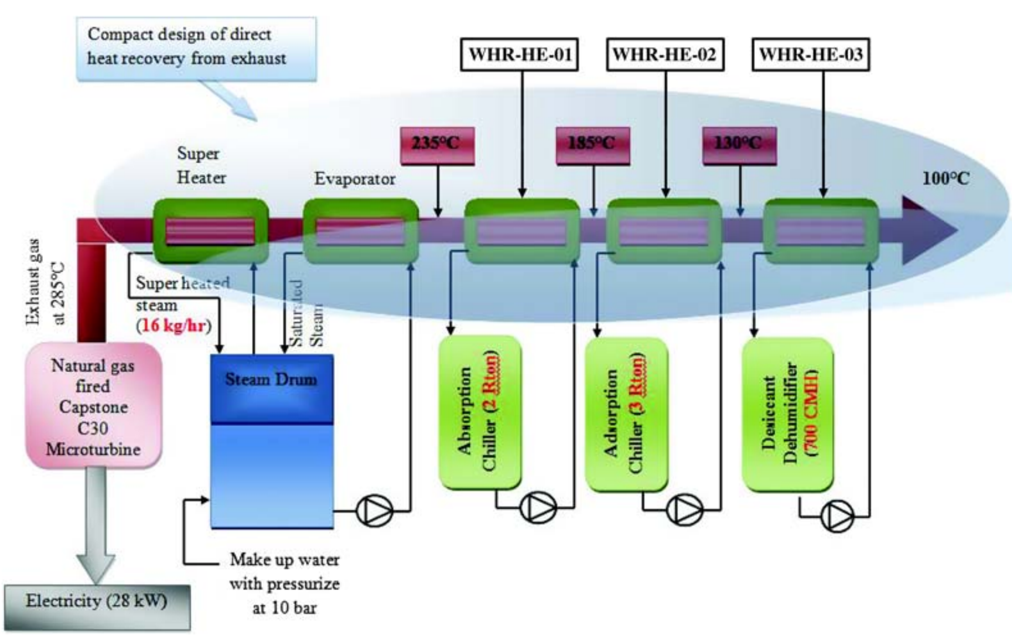

Fig. 1. The schematic lay out diagram of temperature cascaded cogeneration system.

hot water is utilized as the main heat source for the multi-bed desiccant dehumidification system [20]. Thus, the proposed cogeneration system produces four different types of useful energy with utilizing a single energy source.

2.2. Mathematical modeling

The mathematical model based on the second law of thermodynamics is presented in this section. As shown in Fig. 1, the numerical model for each of the waste heat-activated component in the TCCP is formulated to capture the transient characteristic of the corresponding devices. The exhaust gas temperature and the electricity generation 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 modeling for the temperature cascaded cogeneration plant is developed to observe the performance of the plant. The heat source or heat supply to micro-turbine can be expressed as

$$Q_{\text{in,cogen}} = \overset{\circ}{m}_{\text{f}} \; xC_{\text{v}} \tag{1}$$

The energy gas recovered from exhaust gas can be represented by the following expression.

$$Q_{\text{rec}} = \overset{\circ}{m}_{\text{exh}} \; xCp_{\text{exh}} \; (T_{\text{exh,in}} - T_{\text{exh,o}}) \tag{2}$$

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 the devices can be expressed as follows:

A steam generator

$$Q_{\text{steam}} = \overset{\circ}{m}_{\text{steam}} \, x \, h_{\text{g}}(T, P) \tag{3}$$

An absorption chiller

$$Q_{\text{cooling,AB}} = \frac{dM_{\text{chi}}}{dt} Cp_{\text{chi}} \; (T_{\text{chi,in}} - T_{\text{chi,o}}) \tag{4}$$

An adsorption chiller

$$Q_{\text{cooling,AD}} = \frac{dM_{\text{chi}}}{dt} Cp_{\text{chi}} \int_0^{t_{\text{cyc}}} \frac{T_{\text{chi,in}} - T_{\text{chi,o}}}{t_{\text{cyc}}} dt \tag{5}$$

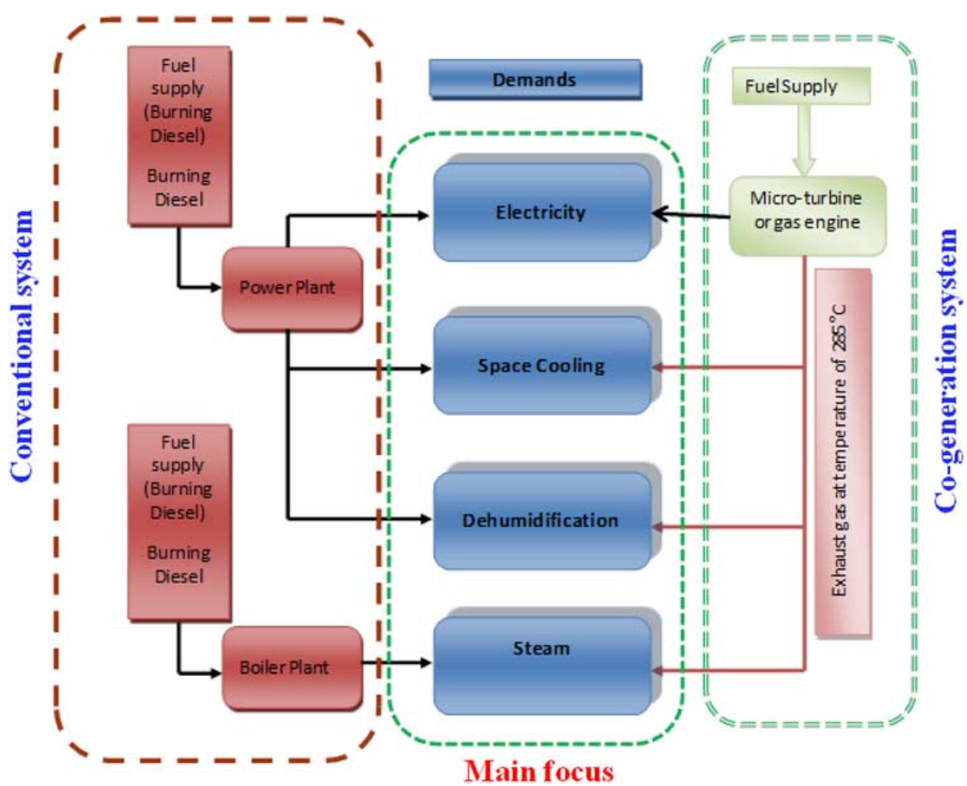

Fig. 2. The comparison between conventional power plant and the temperature cascaded cogeneration plant.

A multi-bed desiccant dehumidifier

$$Q_{\rm deh} = \frac{dM_{\rm air}}{dt} h_{\rm fg} \int_0^{t_{\rm cyc}} \frac{\omega_{\rm in} - \omega_{\rm o}}{t_{\rm cyc}} dt \tag{6}$$

The performance of the temperature cascaded cogeneration plant can be determined by the overall efficiency of the plant so-called EUF and it can be expressed by the ratio of total useful energy produced by the cogeneration plant and the energy supplied to the micro-turbine.

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

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

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

where $Q_{\rm in,conv}$ is the energy required by the conventional plant, which produces the same energy demand compared to the temperature cascaded cogeneration plant. Fig. 2 shows the comparison between conventional plant and the cogeneration plant.

The energy supplied to the conventional plant can be represented as the sum of fuel supplied to the power plant and the boiler and it can be written as

$$Q_{\rm in,conv} = (m^{\circ}_{\rm f,power} + m^{\circ}_{\rm f,Boier}) x\ C_{\rm v} \tag{9}$$

3. Result and discussion

The governing equations are solved by the fifth-order Gear's Backward Differential method of the double precision for initial value problem using Adam's Gear method subroutine of international mathematics and statistics library (IMSL) Fortran library subroutines. The iterative scheme uses a double precision format, which has a tolerance of 1×10^{-6}. The nominal capacity of each of waste heat-activated system and capacity of is tabulated in Table 1, and numerical calculations were performed for the performance analysis and entropy generation. The consistency of the simulation model was checked with suitable range of parameters that is anticipated of the TCCP operation domain. The hot water source temperatures are varied from 65 to 85°C, while the cooling water and the chilled water temperatures are kept at about 29 ± 1°C and 12 ± 1°C, and the cooling

Table 1
The nominal capacity of each of the devices equipped with 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-beds desiccant dehumidifier	Dehumidification	7	NA

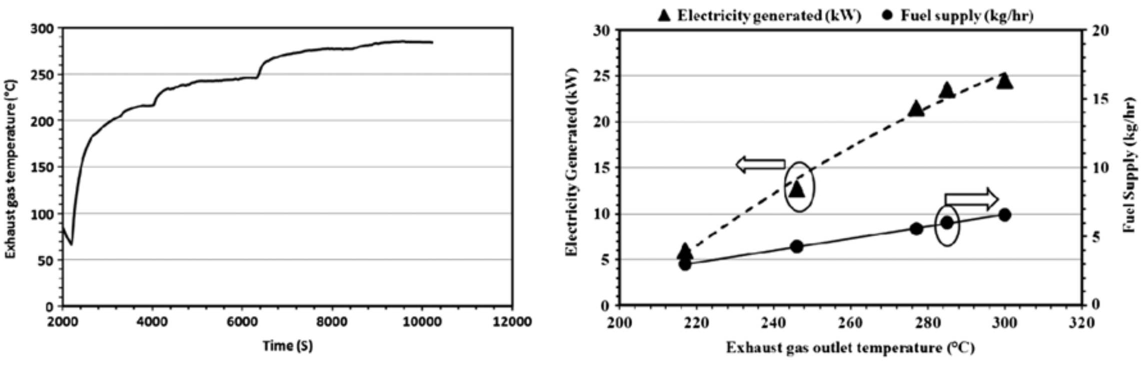

Fig. 3. The performance of micro-turbine contained in the temperature cascaded cogeneration plant.

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 temperature. It is observed that both of fuel consumption and electricity generation are linearly increased with the increase in the exhaust gas temperature. The predicted temperature profiles of exhaust gas temperature emanating from each of the waste heat recovery heat exchanger unit are illustrated in Fig. 4, while micro-turbine outlet exhaust gas temperature is about 285°C.

The performance parameters such as temperature profiles and the steam production of waste heat-driven steam generator (WHSG) are shown in Fig. 5, while the waste heat source (exhaust gas temperature emanating from micro-turbine) is maintained at 285°C. It is observed that the super heated steam at a temperature of 250°C with 10 bars is produced from the WHSG while 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 a temperature of 240°C, is then channeled to the waste heat extraction heat exchanger WHR-HE-01 coupled with the absorption cycle to produce the hot water to fire to 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 hundreds second before it reaches to the steady state. The performance of two beds regenerative adsorption chiller is presented in Fig. 7. The outlet exhaust gas stream, at a temperature about 210 ± 2°C, is diverted to WHR-HE-02 coupled with the adsorption cooling cycle to produce the hot water to fire to the AD cycle, which operates in a batch manner. The predicted performance of the AD cycle is shown in Fig. 7.

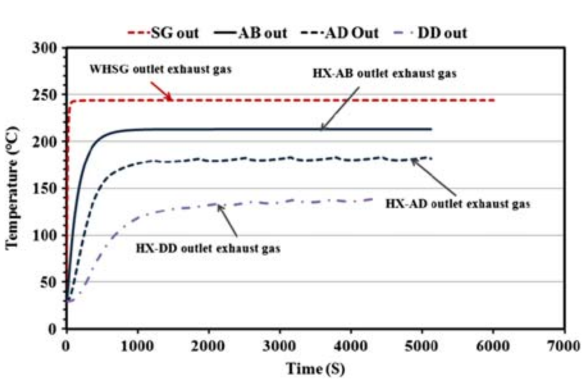

Fig. 4. Temperature time history of exhaust gas from each of the waste heat recovery heat exchanger unit.

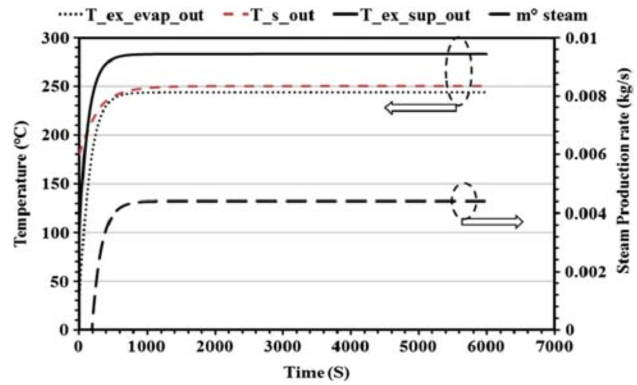

Fig. 5. The performance of the waste heat recovery steam generator and its steam production rate.

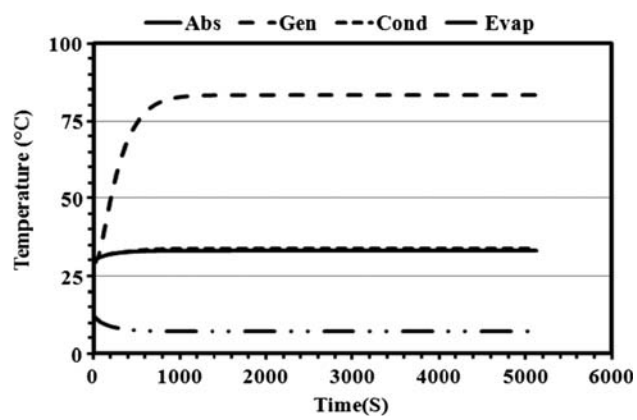

Fig. 6. The performance of Li–Br water absorption chiller.

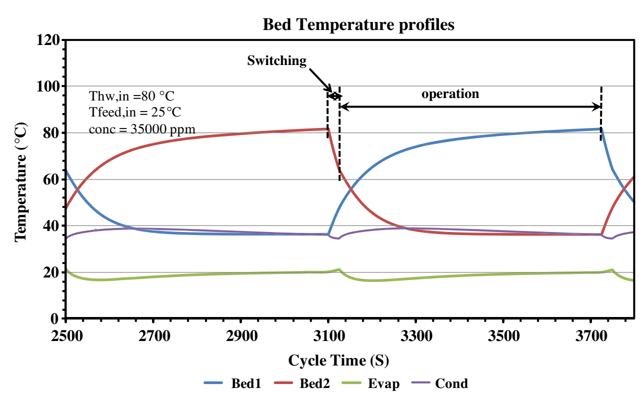

Fig. 7A. The performance of two beds regenerative silica gel–water adsorption desalination unit.

The total mass of adsorbent has been fixed at 78 kg and the half cycle time is fixed at 600 s while switching time is 40 s. It is observed that the cyclic steady state reaches within 4 cycles of operation. Fig. 7(A and B) illustrates the temporal temperature profiles major components in the AD cycle and its heat trans-

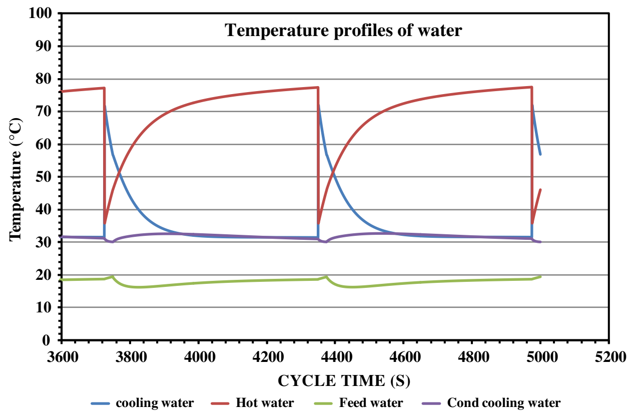

Fig. 7B. The outlet temperatures of heat transfer fluids and feed water from silica gel–water adsorption desalination unit.

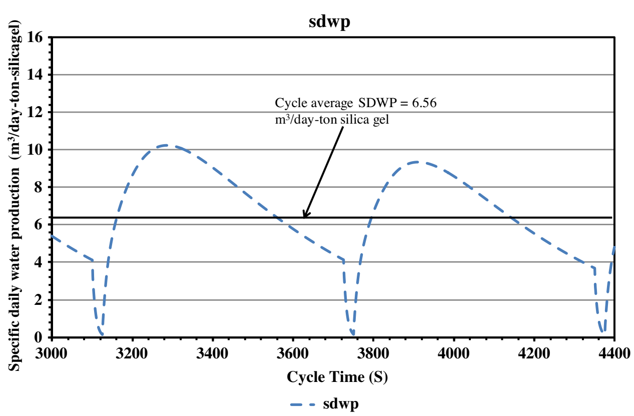

Fig. 7C. Specific daily water production of AD cycle.

fer fluids outlet heat transfer fluid. The cycle average outlet coolant temperature for adsorber is 3.5–4℃ higher than that of inlet temperature about 5℃ and that of chilled water is about 5℃ while cooling capacity is obtained 7 kW. The specific daily water production (SDWP) is illustrated in Fig. 7C and it is achieved that 6.56 m^3 of distilled water can be produced daily with one ton of silica gel. Temperature profiles of adsorber and desorber beds in the multi-bed desiccant dehumidification system (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 the laminar flow with a set velocity of 0.25 m/s. It is observed that the cyclic steady state reaches 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 humid 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 by 2–3℃. Thus, a cooling coil is installed to reduce the sensible heat induced by adsorption process. Fig. 9 shows the temporal history of outlet air stream before cooling coil and after cooling coil. It is indicated in Fig. 9 that the considerable amount of sensible heat is induced during hot to cold switching. The residual energy in the adsorbent provides the sensible heat to 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 sensible heat (temperature) of outlet air stream before it is sent to air handling units (AHUs). Inlet humidity ratio, outlet humidity ratio, and cycle average outlet humidity ratio are illustrated in Fig. 9. It is observed that cycle average outlet humidity ratio is about 16 g/kg DA while cycle average moisture removal by MBDD unit is 12.5 g/kg DA. Thus, the MBDD cycle

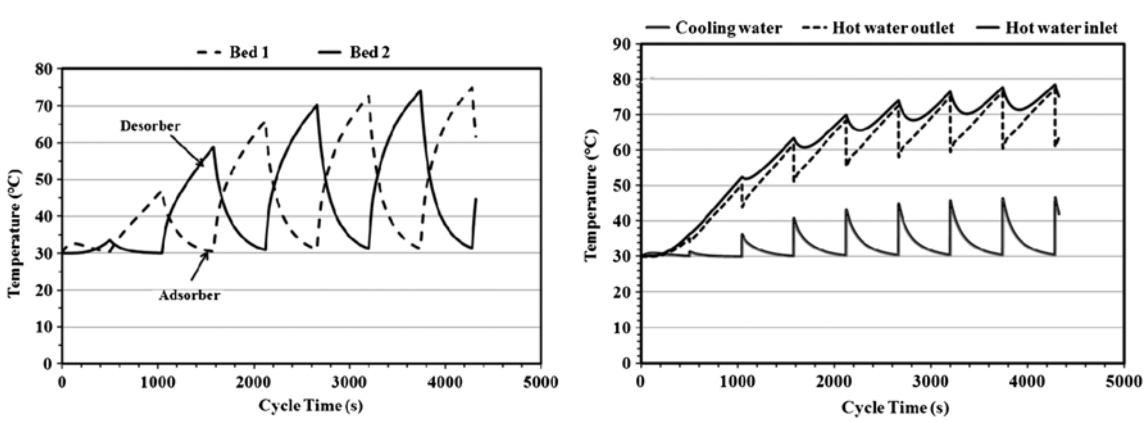

Fig. 8. The performance of multi-bed regenerative silica gel–water desiccant dehumidifier.

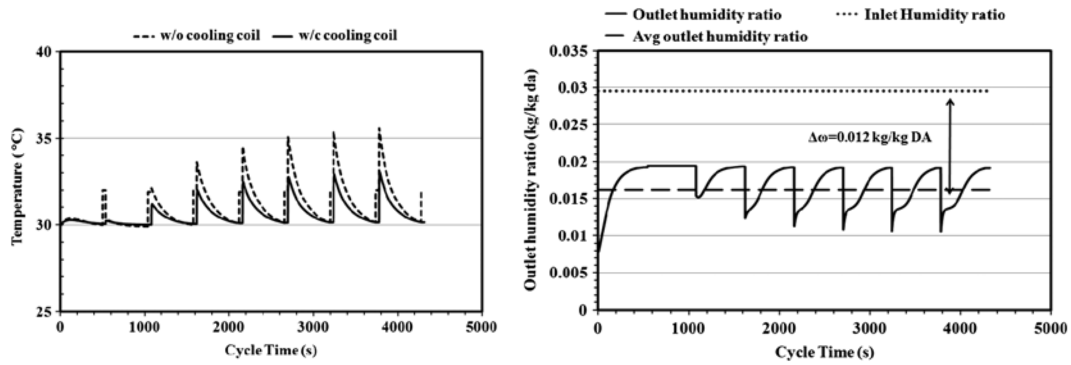

Fig. 9. The performance of multi-bed regenerative silica gel–water desiccant dehumidifier and its moisture removal.

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 system, EUF
Steam generator	18.5	12.9	0.7
Absorption chiller	11.7	6.76	0.58
Adsorption desalination	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

could remove 40% of moisture containing in the ambient air resulting reduction in AHUs load. Table 2 summarizes the performance of each of the waste heat-activated devices and total useful energy demand.

4. Conclusion

Performance analysis of TCCP comprises, namely (i) a steam generation cycle, (ii) an absorption cooling cycle, (iii) an adsorption desalination 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 EUF is obtained as high as 70% while the said host of waste heat-activated devices is arranged in the cascaded configuration. Owing to the effective arrangement of devices, the FESR is attained 28%, which will lead to the significant reduction in energy cost and carbon dioxide emission.

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