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Impact of Numerous Beds Operating Conditions for Enhancing the Performance of a Solar Heat-Supported Adsorption Chiller

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Abstract

Thermal adsorption cooling systems have gained significant attention due to their potential for energy savings and eco-environmental impact. An analytic investigation of the heat transfer inside an adsorption chiller with various bed silica gel-water pairs is presented. A comprehensive model has been designed to accurately predict the correlation between the overall performance of the proposed chiller system and the functional and structural condition of the building. This model takes into account various factors such as temperature, humidity, and air quality to provide a detailed analysis of the system's efficiency. At least 20 collectors consisting of a 34.4 m area (each) with a full-cycle time of 480 seconds are essential to improper run conditions. It is necessary to adjust the optimum cycle time for optimal performance. During the investigation, the base condition shows that the cooling capacity is 14 kw, 0.6 COPcycle, and 0.35 COPsolar at noon. Also, conduct a thorough investigation into the chiller's performance under varying cooling water supply temperatures and various chilled water flow rates.

Keywords

Adsorption Chillers, Adsorption Cooling, Multiple Adsorption Beds, Solar Heat

1. Introduction

Adsorption refrigeration is a well-known technology that exploits waste heat to antiquated cooling for reducing fossil fuel depletion and CO_2 radiations. Multi-effect adsorption technology is an innovative approach that has proven useful in desalination and water treatment [1]. Solar heat is an effective and suitable source of energy for low-grade heat adsorption cooling units. According to the study, the performance of the silica gel-water pair in terms of coefficient of performance (COP) is highly dependent on the regeneration temperature. Specifically, the study found that at lower regeneration temperatures, the COP-values for the pair were notably higher. This suggests that the silica gel-water pair is a promising choice for areas where low-temperature regeneration is preferred. Nevertheless, the main difficulty of this system is its large footprint. Therefore, this study is an effort to reduce the installation cost and size of the cooling unit.

The cascading adsorption scheme improves the COP of 2-bed irregular cycles [2] [3] [4]. This cooling system can enhance efficiency by using a combined evaporator and condenser in two cycles with multiple beds. It ensures uninterrupted cooling, which improves both COP and SCP. Habib *et al.* [5] conducted in-depth research on a highly efficient four-bed adsorption refrigeration system. By utilizing activated carbon/R507A and activated carbon/R134a as its working pairs, this system was designed to operate efficiently and effectively. This work investigates optimal operating conditions for 70˚C low recovery temperature and −10˚C refrigeration load.

The study revealed that the advantage and development of the adsorption phase Meunier [6] and current wave cycle Shelton *et al.* [7] to expand cooling capacities Wang [8] and Akahira *et al.* [9] and reheat two-stage cycle Alam *et al.* [10]. The advanced cycle introduces better exploitation of heat sources, for example, multi-bed systems [11] for exploitation of low-temperature heat sources and a three-stage cycle [12] and two-stage cycle [13].

The study investigated the performance of two different adsorbent/adsorbate pairs at various generation and evaporation temperatures, achieving an evaporator temperature of −10˚C. The pairs were zeolite water and activated carbon-methanol. Meunier [14] theoretically considered cascading adsorption of four absorbers with four configurations, two condensers, and two evaporators. Dawoud [15] developed a hybrid solar-adsorption refrigeration system that isunited double adsorption cycles to attain low-temperature cooling for vaccine storage. Oliveira [16] an experiment was done on a four-bed adsorption ice-making system with a single condenser and evaporator and an activated carbon/ammonia working pair. The desorption temperature ranged from 850˚C - 115˚C, while the evaporator temperature was as low as −27˚C. The research also involved the experimental investigation of double-stage adsorption freezing unit by Wang [17] experiment utilizing three distinct heat source temperatures at 75˚C, 80˚C, and 85˚C, and achieved corresponding evaporator outlet temperatures of −5˚C, −10˚C, and −15˚C, respectively. For this research, $CaCl_2$ and $BaCl_2$ chemical adsorbents were used together with ammonia. Wang created this system by combining Organic Ranking Cycle with the system that Jiang [18] introduced and using the cascading method, which is the most efficient way to make use of waste heat. After that, we moved on to the advanced cycle which included multiple beds for a more effective cooling process by Wirajati [19]. Lately, Dakkama [20] diverse working pairs were used to inspect the performance of a combined evaporator-condenser adsorption chiller for base-temperature cooling. Alam *et al.* [21] found that a four-bed advanced mass recovery adsorption refrigeration system has a higher coefficient of performance (COP) than a two-stage system when the heat source temperature is below 60˚C. Rifat *et al.* [22] investigated an adsorption cooling system based on Dhaka's climatic conditions and also studied energy management and heat storage for solar adsorption cooling systems [23]. Recently, multiple-bed systems were considered for low heat input by Rifat *et al.* [24]. The best approach to enhance adsorber bed performance is to divide them into multiple smaller beds. This system strongly recommends implementing this strategy to achieve better results with less solar heat. Additionally, the operating conditions for a three-bed chiller were thoroughly discussed by Rouf *et al.* [25].

Solar heat-driven adsorption chillers are indeed gaining popularity due to their ability to harness low-grade heat, and installing a multiple-bed reduced-sized cooling unit only further confirms their effectiveness. The working conditions on the acting of the two-bed system with 13 collectors each of area 2.415 m^2 with optimum cycle time 1000 sce has been considered by Rifat *et al.* [26] previously. The study investigates the impact of multiple-bed systems on solar heat-driven adsorption cycles, exploring their effects. Due to the reduction in container size in accordance with the amount of adsorbent for the chillers with three and four beds, heating time for the bed has significantly decreased. A short cycle time was chosen. On the other hand, since the multiple bed systems

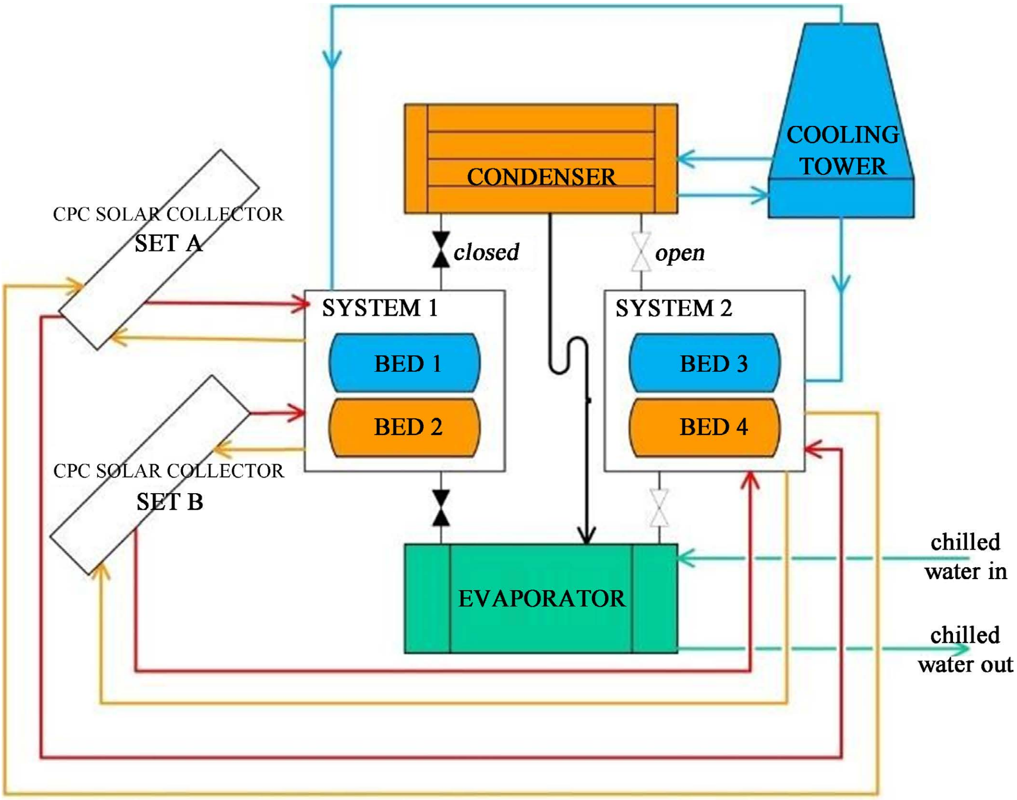

Figure 1. Graphic illustration of the 4-bed parallel system.

have preferred longer precooling time, the adsorption process is improved by a significant temperature gradient. Multiple adsorbent beds and shorter cycles produce better daytime cooling with available solar radiation.

2. System Description

The operating procedure of a solar-assisted chiller with two beds is described in the available literature [26]. Rouf *et al.* [25] have previously deliberated on the operational parameters of the three-bed chiller. A new four-bed cooling unit has been proposed to replace two separate two-bed conventional cooling units. The new unit will alternate between cooling the beds by utilizing a single condenser and an evaporator. As a result, a continuous process of evaporation and condensation occurs in this system, resulting in improved cooling production for this four-bed setup. The change in the operating conditions, the advantages and disadvantages of these changes, and a comparison in the performance has been discussed. After analysis, the system determined the optimal number of adsorption beds required for two conventional 2-bed solar heat-adsorption cooling systems, each featuring a single evaporator and condenser.The graphic representation of the new system is given in **Figure 1**. System one includes beds 1 and 2, while system two includes beds 3 and 4. Both systems are connected to the same condenser, evaporator, cooling tower, and two different CPC solar thermal collectors. System one closes valves during preheat/pre-cool. System two connects to the condenser and evaporator during desorption/adsorption.

Mathematical Formulations:

The solar thermal collector is collected from the heat collector as a heat source for the cooling unit manufactured and the collector efficiency equation is:

$$\eta = 0.64 - 0.89\left(\frac{T_f - T_{am}}{I}\right) - 0.001\left(\frac{T_f - T_{am}}{I}\right)^2 \tag{1}$$

A sine function as:

$$I = I_{\max} \sin\left(\frac{\pi\left(daytime - sunsettime\right)}{sunsettime - sunrisetime}\right) \tag{2}$$

Every pipe of the system independently calculated the temperature of the heat transporter fluid for all the solar thermal collectors. Hence for every single collector energy balance equation is:

$$\begin{aligned} M_{cp,k}\frac{\mathrm{d}T_{cr,k}}{\mathrm{d}t} &= \gamma\left\{\eta_k A_{cr,k} I + \tilde{m}_{f,cr} c_{p,f}\left(T_{cr,k,in} - T_{cr,k,out}\right)\right\} \\ &\quad + \left(1-\gamma\right) U_{tu\,M\,loss} A_{cr,k}\left(T_{am} - T_{cr,k}\right) \end{aligned} \tag{3}$$

$$T_{cr,k,out} = T_{cr,k} + \left(T_{cr,k,in} - T_{cr,k}\right)\exp\left(U_{cp,k} A_{cp,k} / \tilde{m}_{f,cr} c_{p,f}\right) \tag{4}$$

where $k = 1,\cdots,9$. γ is 1 during the daytime and 0 at nighttime, respectively.

The heat transfer units such as the pressure and temperature have been chosen as uniform. The mathematical model is the same as that of Rifat *et al.* [23].

The Average cooling capacity (*CACC*) is calculated by the equation:

$$CACC = TotalQ_{coolout} / t_{cycle} \tag{5}$$

The Specific cooling capacity (SCC) per cycle is calculated as:

$$SCC = TotalQ_{coolout} / \left(t_{cycle} W_s \right) \tag{6}$$

The solar coefficient of performance is calculated as:

$$COP_{solar} = TotalQ_{coolout/cycle} / TotalQ_{hotin/cycle} \tag{7}$$

The cycle COP (COP_{cycle}) and solar COP in a cycle (COP_{sc}) are calculated respectively by the equations:

$$COP_{cycle} = \frac{\int_{begin\ of\ cycletime}^{end\ of\ cycletime} \dot{m}_{cw} c_{p,cw} \left(T_{cw,in} - T_{cw,out} \right) \mathrm{d}t}{\int_{begin\ of\ cycletime}^{end\ of\ cycletime} \dot{m}_{HW} c_{P,HW} \left(T_{HW,in} - T_{HW,out} \right) \mathrm{d}t} \tag{8}$$

$$COP_{sc} = \frac{\int_{begin\ of\ cycletime}^{end\ of\ cycletime} \dot{m}_{cw} c_{p,cw} \left(T_{cw,in} - T_{cw,out} \right) \mathrm{d}t}{\int_{begin\ of\ cycletime}^{end\ of\ cycletime} n \cdot A_{cr} \mathrm{d}t} \tag{9}$$

3. Methodology

This study explores based on the maximum solar radiation 988 W/m^2 in the month of April in Dhaka. The study investigates the operating conditions for performance enhancement of a solar heat-supported adsorption chiller. Here, 5.5 h and 18.5 h are considered as the sunrise and sunset times of a particular month, and also considered a maximum temperature of 34˚C and minimum temperature of 24˚C. To solve the differential equations a finite difference method has been used and tolerance for all the criteria is 10^{-4}.

In the literature [27] the energy balance equations of all the heat transfer units can be found. Logical programming language FORTRAN with Compaq Visual Fortran compiler has been used to calculate the numerical results of this simulation.

4. Results and Explanations

Based on Rifat *et al.* [26], we examine the effect of operating conditions on the performance of a two-bed adsorption chiller. Accordingly, at least 13 collectors with an optimum cycle time of 1000 sec are considered. However, lately Rifat *et al.* [24] exploiting multiple adsorption beds can increase cooling from limited solar heat. This paper is specifically personalized to ensure efficient operation across multiple beds with varying numbers of collectors and cycle times. Furthermore, to observe the impact of different cooling water inlet temperatures on evaporator performance and adjust chilled water supply accordingly.

The temperature range achieved was between 85˚C to 95˚C using 23.5 kg of RD-type silica gel. It was observed that the evaporator outlet maintained a steady temperature of 8˚C with the 4-bed system. The study examines the bed and collector outlet temperature histories for different numbers of collectors and varying cycle times. The presented temperatures for collector outlet and bed at different cycle times in Figures 2(a)-(d).

Thus, it is achieved that the collector outlet temperature is 81˚C when 80˚C reaches the bed temperature shown in Figure 2(a) for ten collectors with a cycle time of 500 sec. In Figure 2(b), for cycle time, 600 sec bed temperature is 82˚C while the collector outlet is 83˚C, for the base run condition, achieve driving source temperature level using cycle time 1000 sec bed temperature increases 88˚C while the collector outlet is 89˚C.

In order to attain the desired temperature level with cycle times of 500 s, 600 s, and 800 s, 12 collectors are required in Figure 3(a) bed temperature reaches 87.6˚C while the collector outlet temperature is 88.7˚C. In Figure 3(b) for cycle time, 600sec bed temperature is 91.71˚C and the collector temperature 93˚C and Figure 3(c) shows that the bed temperature and the collector outlet increase by 104˚C and 105˚C accordingly. The temperature exceeds 100˚C for 14 collector numbers when cycle time is increased beyond 600s in Figure 4.

Again, the temperature history of 16 collectors with a cycle time of 600 s is extant in Figure 5. The collector outlet reaches 120˚C, greater than 100˚C. However, the bed temperature increases in the cycle time for all cases, which is not enhanced for the silica gel water adsorption bed. And also decrease in driving source temperature by decreasing the cycle time.

For 20 collectors the optimum cycle time of 120s is considered as driving heat source temperature is attainable with this cycle time depicted in Figure 6. Due to the limited amount of adsorbent in each small bed and high heat input, the system

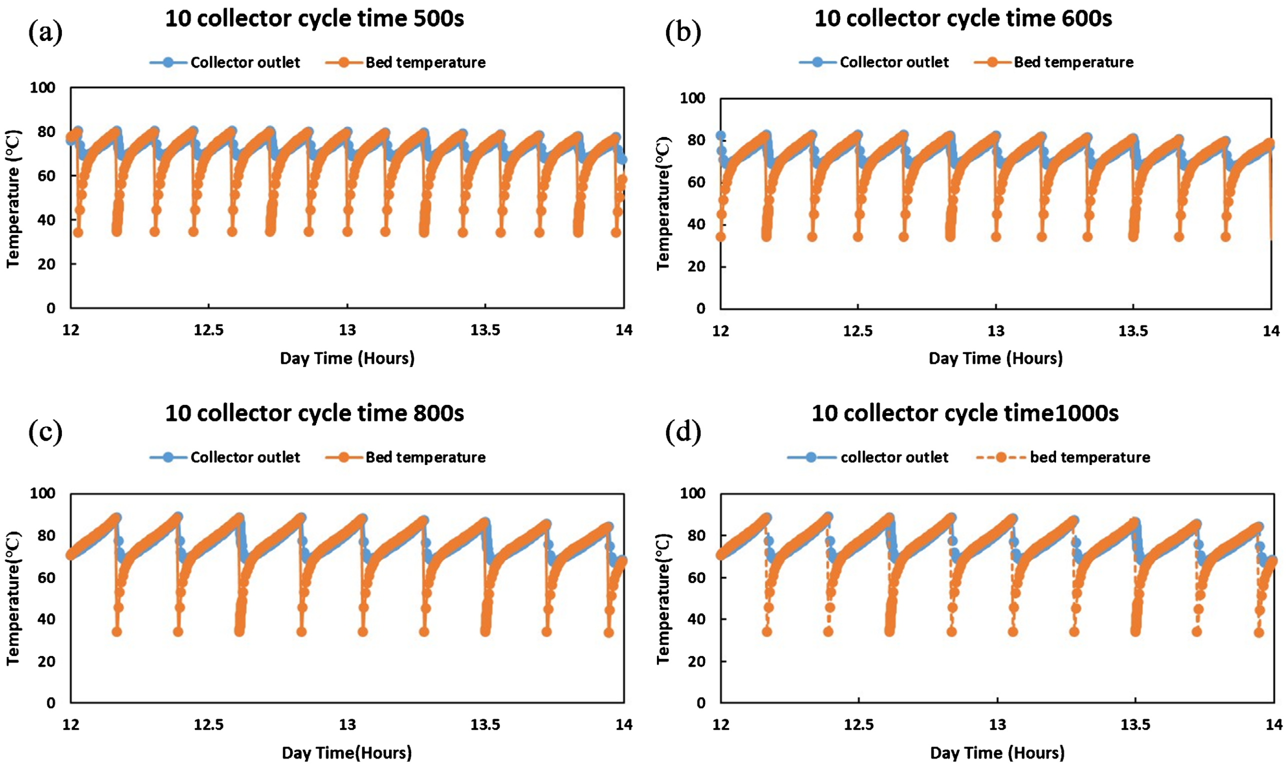

Figure 2. Temperature profiles for (a) 10 collectors with cycle time 500 s; (b) 10 collectors with cycle time 600 s; (c) 10 collectors with cycle time 800 s; (d) 10 collectors with cycle time 1000 s.

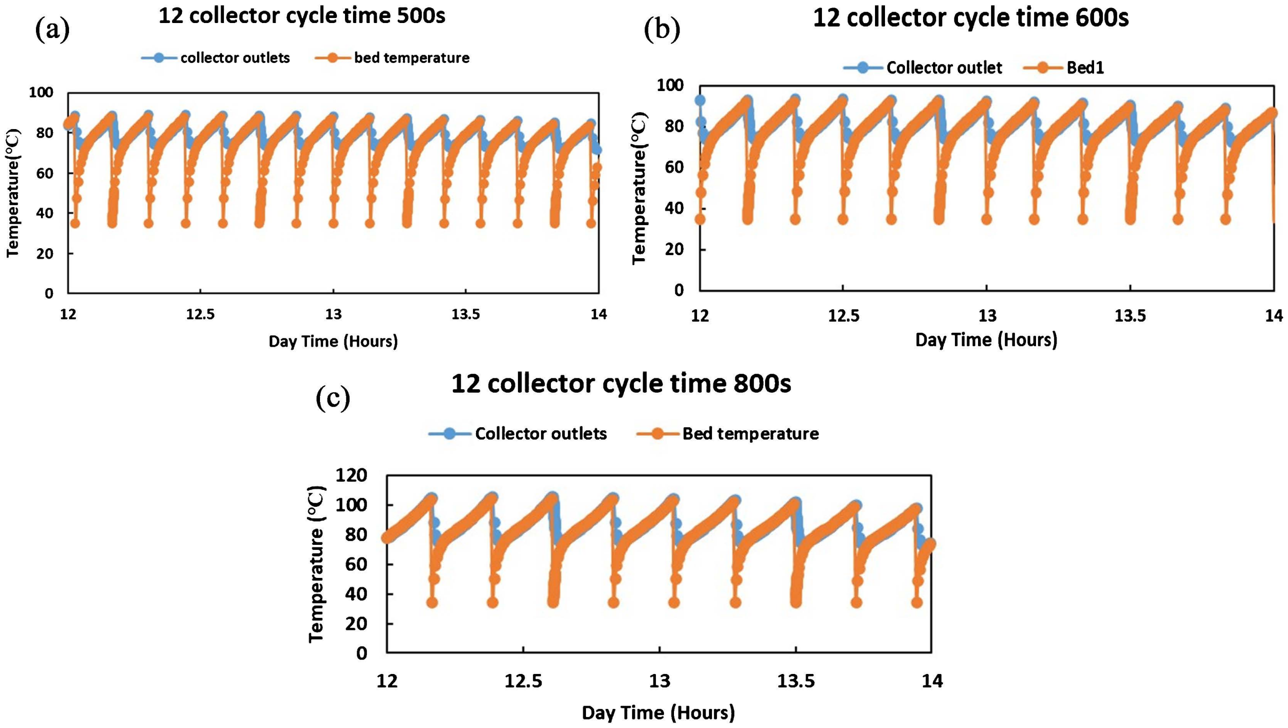

Figure 3. Temperature profiles for (a) 12 collectors with cycle time 500 s; (b) 12 collectors with cycle time 600 s and (c) 12 collectors with cycle time 800 s.

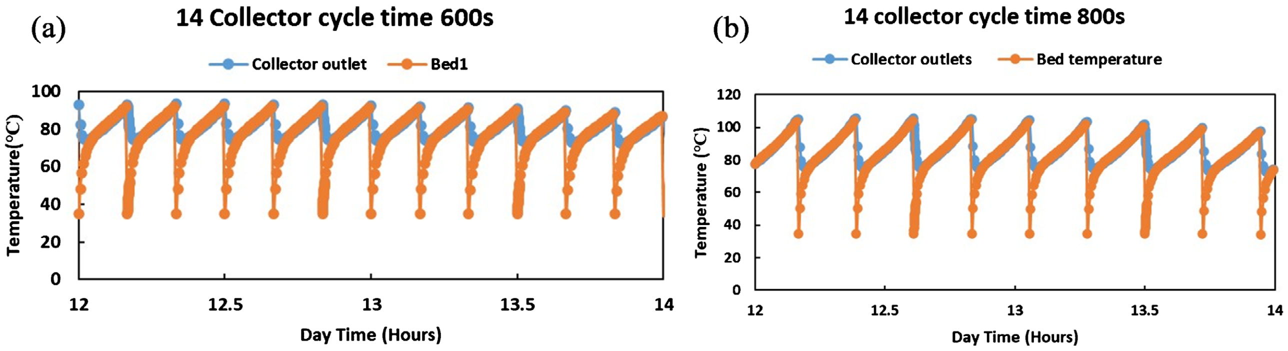

Figure 4. Temperature profiles for (a) 14 collectors with cycle time 600 s; (b) 14 collectors with cycle time 800 s.

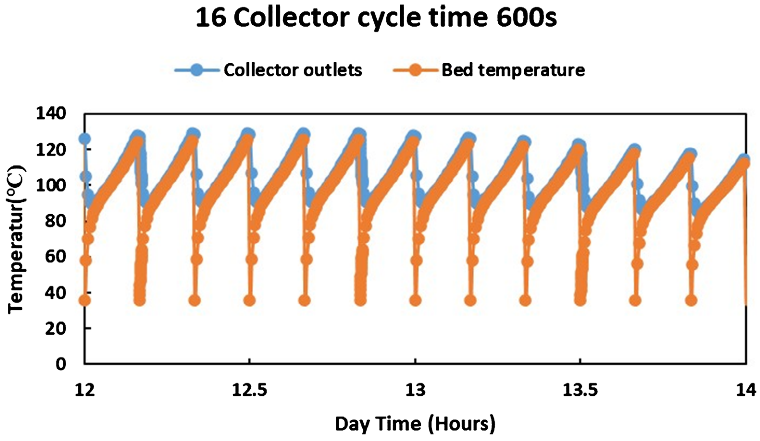

Figure 5. Temperature profiles for 16 collectors with 600 s.

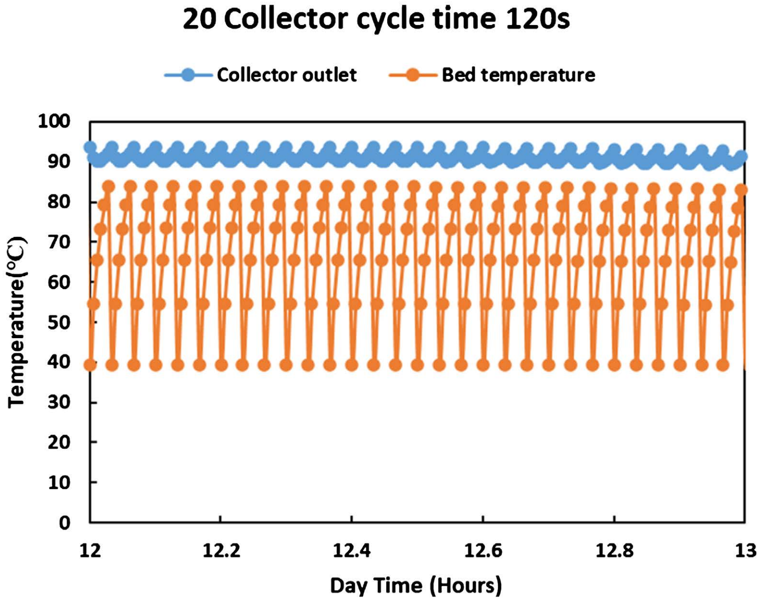

Figure 6. Temperature profiles for 20 collectors with 120 s.

cannot have a longer cycle time. Therefore, the bed temperature cannot decrease much, and the adsorber's entropy remains low. Furthermore, the output is the opposite when there is less heat input (34.4 m^2 collector area). By distributing the total amount of adsorbents into multiple adsorption beds and increasing precooling time with a small amount of heat input, a better output can be achieved.

Figure 6 The temperature of the collector outlet reaches 95˚C while the bed temperature reaches 85˚C for 20 collectors with a cycle time of 120 seconds. To investigate the effect of reducing heat input, we will be using four adsorption beds in our study. Two parallel systems are being considered. For optimal chiller performance, use a cycle time of 120 seconds when 20 collectors are in use. The ambient temperature of 31˚C is considered the cooling source temperature, with a chilled water inlet temperature of 14˚C. The chilled water flowing into the evaporator has a volumetric flow rate of 0.7 kilograms per second.

Figure 7 represents the cooling capacity disproportion to the cooling source temperature. For the time being, it decreases the cooling source temperature while increasing the cooling capacity. As the temperature difference between the hot and cooling source increases, the adsorption capacity of the silica gel bed increases, resulting in an increase in cooling; furthermore, before the late afternoon, the efficiency of the chiller improves. I made sure to correct any spelling, grammar and punctuation errors. When the cooling source temperature is 31˚C, then the chiller produces a 14 kW cooling temperature at noon. But suddenly, in the late afternoon starts to decline, which is 28˚C cooling source temperature.

Now, investigating the performance of the chiller to determine how different chilled water flow rates affect its operation in **Figure 8** shows that the chiller's arrangements ($CACC$, COP_{cycle}, and COP_{sc}) are proportional to chilled water mass flow rates of at least 5 pm. Opposite trends are observed after 5 pm. If we

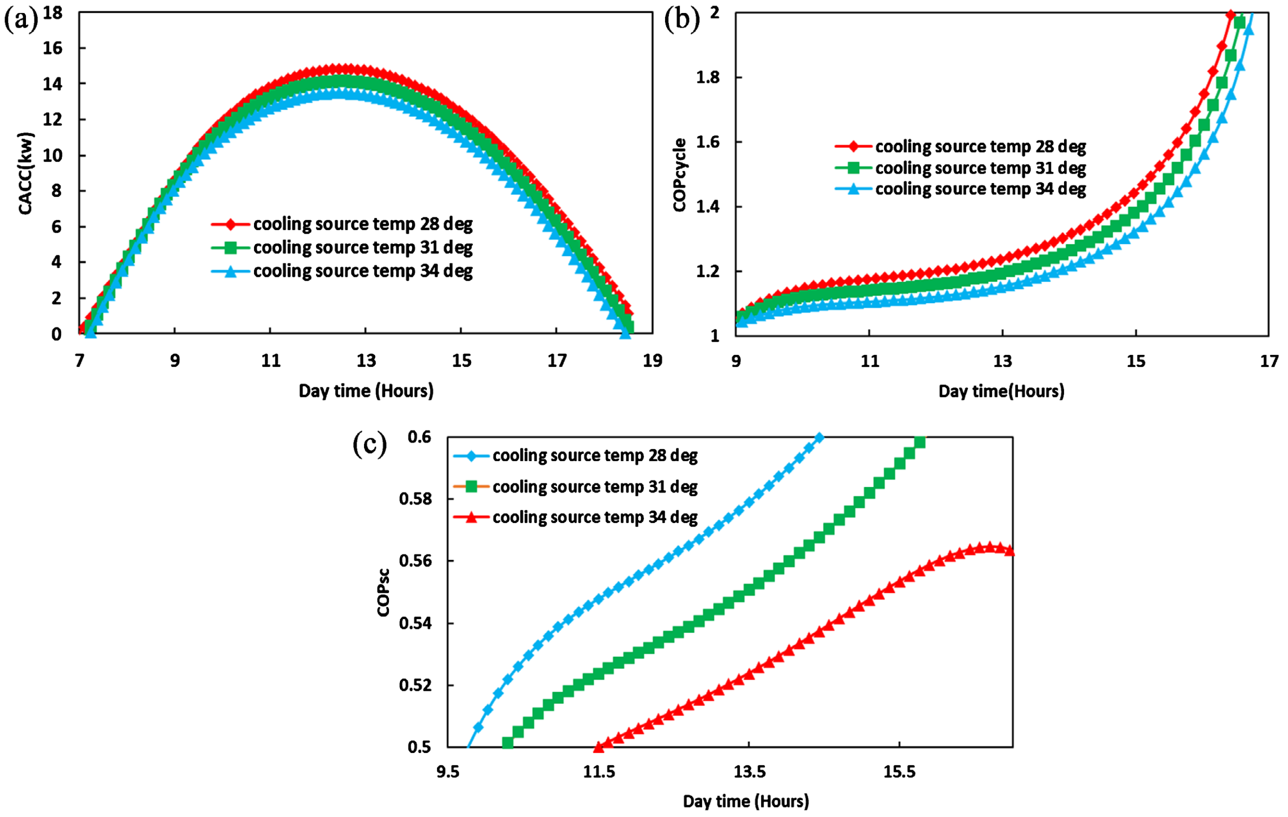

Figure 7. Performance for different temperature cooling water supply to the adsorber (a) *CACC*; (b) COP_{cycle}; (c) COP_{sc}.

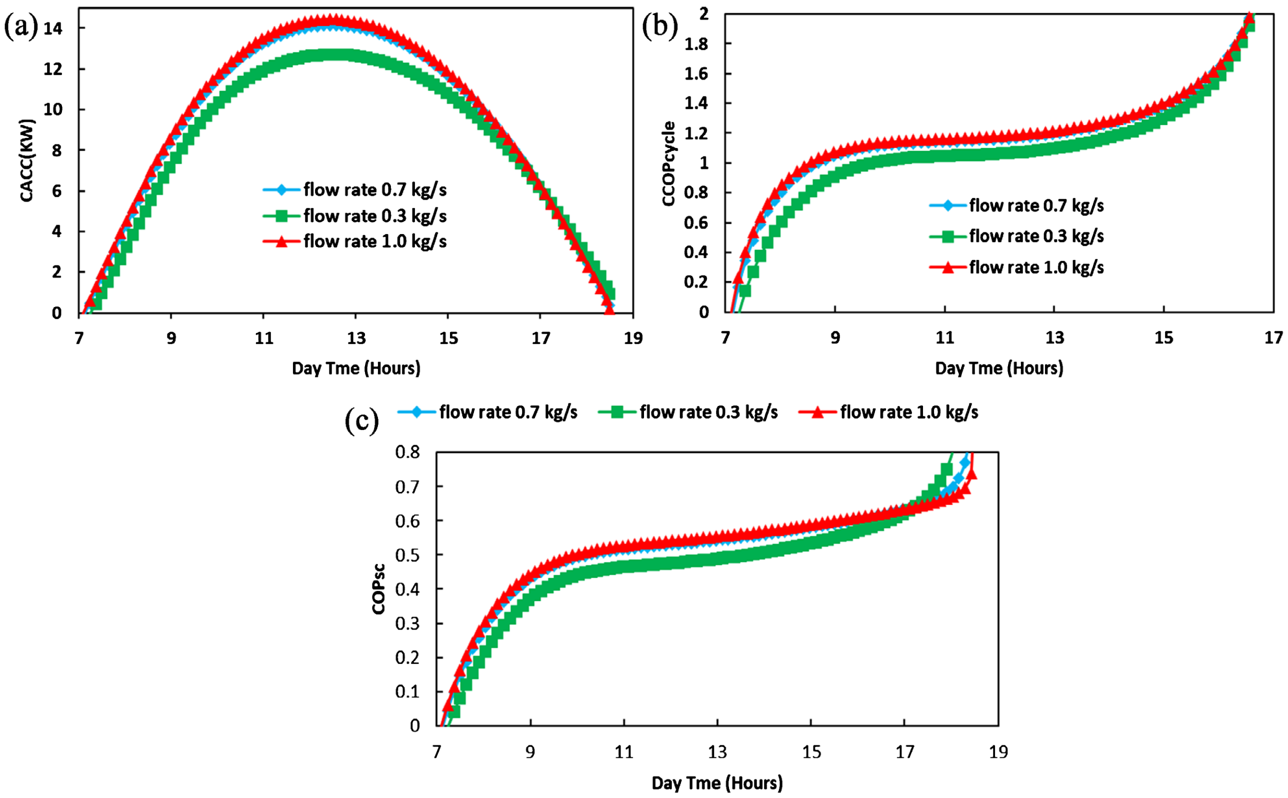

Figure 8. Performance for different chilled water flow rate of 20 collectors with cycle time 120 s (a) *CACC*; (b) COP_{cycle}; (c) COP_{sc}.

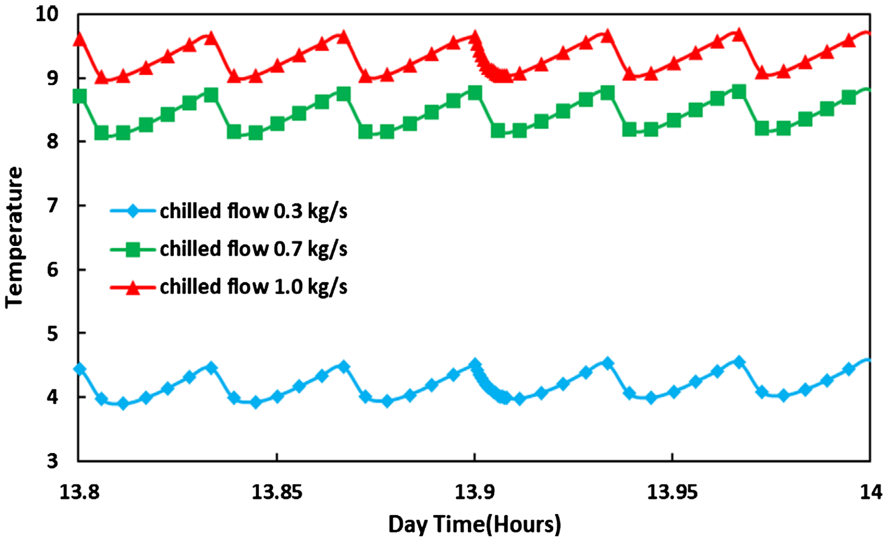

Figure 9. Different chilled water flow rates of chilled water outlet temperature.

examine the measurement equations of the system, it becomes apparent that the performance of the chiller is directly proportional to the flow rate of chilled water.Meanwhile, the chilled water flow rates are strongly inclined to the chilled water temperature outlet, presented in Figure 9. The study examines the temperature records of the chilled water outlets from 20 collectors with varying flow rates in this figure. It has been observed that decreasing the flow rates of chilled water results in a lower temperature of the chilled water outlet. Having a chilled water outlet is crucial as it provides the necessary cooling for the space. The cooling capacity and COP are directly proportional to the chilled water mass flow rate. However, increasing the chilled water mass flow rate can lead to an increase in the chilled water outlet temperature, which is consumed by the user. Hence, it is important to select an appropriate chilled water outlet temperature and flow rate to achieve optimal performance.

5. Conclusions

This study examines the impact of operating conditions on the efficiency of a water-driven adsorption AC system with multiple beds containing silica gel. The following conclusions can be drawn:

- It is necessary to have a minimum of 20 collectors, each with an area of 34.4 m^2, and a full-cycle time of 480 seconds for the base run conditions.
- When the flow rate of chilled water through the evaporator increased, the coefficient of performance (COP) of the chiller improved and the outlet temperature of the chilled water increased as well.
- The flow rates of chilled water are greatly affected by the temperature of the outlet.
- For optimal performance, 20 collectors require a minimum cooling water flow of 1 kg/sec at 31˚C and chilled water flow of 1 kg/sec at 14˚C. The full-cycle time is 480 seconds, with 14 kW cooling capacity, 0.6 COPcycle, and 0.35

COPsolar.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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Nomenclature

A	area (m^2)
$CACC$	cyclic average cooling capacity (kW)
COP	coefficient of performance
cp	specific heat energy (J/kgK)
Ds	surface diffusivity (m^2/s)
$Ds0$	coefficient of surface diffusivity (m^2/s)
Ea	energy to activate (J/kg)
I	solar radiation (W/m^2)
$\kappa_s a_p$	total mass transfer coefficient (m^2/s mm)
L	latent heat of vaporization (J/kg)
M	mass (kg)
$\tilde{m}$	rate of mass flow (kg/s)
n	number of collector
P	pressure (kPa)
q	adsorption capacity (kg/kg silica gel)
q^*	concentration at equilibrium state (kg/kg silica gel)
Qst	heat of adsorption (J/kg)
R	water gas constant (J/kgK)
S_A	outer surface area (m^2)
Rp	diameter of a particle (silica gel) (mm)
SCC	specific cooling capacity (W/kg)
t	time (s)
T	temperature (K)
U	coefficient of heat transfer (W/m^2 K)
COP	coefficient of performance (−)
c_p	specific heat capacity (J·kg^{-1}·K^{-1})
P	power (W)
$\dot{E}$	exergy rate (W)
hs	specific enthalpy (J·kg^{-1})
s_0	specific enthalpy of the dead state (J·kg^{-1}·K^{-1})

Subscripts

A	area (m^2)
$CACC$	cyclic average cooling capacity (kW)
COP	coefficient of performance
cp	specific heat energy (J/kgK)
Ds	surface diffusivity (m^2/s)
$Ds0$	coefficient of surface diffusivity (m^2/s)
Ea	energy to activate (J/kg)
I	solar radiation (W/m^2)
$\kappa_s a_p$	total mass transfer coefficient (m^2/s mm)
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SCC	specific cooling capacity (W/kg)
t	time (s)
T	temperature (K)
U	coefficient of heat transfer (W/m^2 K)

Greek Symbols

γ	logical parameter
δ	logical parameter
η	efficiency of compound parabolic concentrator (CPC) solar collector