Simulation Study of Continuous Solar Adsorption Refrigeration System Driven by Compound Parabolic Concentrator

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Abstract: Solar adsorption refrigeration is promising technology especially in the developing countries and remote areas because the system can be driven by solar energy without involving electricity. Therefore, it is technologically possible and socially feasible in the areas where electricity is not enough but solar energy is rather easy to obtain. The conventional system works intermittently and produces cooling only during night. In this article, a solar adsorption refrigerator is presented that produces cooling continuously using only solar energy. The system consists of two beds with CPC collectors, a condenser and evaporator. The one cycle complete in two days. A simulation model was developed to evaluate the system performance. The objectives of this study were to investigate the dynamic behaviour of the system, and to compare the system behaviour with the conventional system. The results show that system produces continuous cooling. Cooling rate decreased in the middle of the day because of relatively high ambient temperature. The performance of the system increased with high value of heat transfer coefficient between receiver and bed, and decreased by increasing the adsorbent mass. The performance of the continuous system was compared with that of intermittent system. It was found that the evaporator temperature in the intermittent system increased during the day because cooling is not produced during day in that system, while proposed system kept the evaporator at low temperature continuously. This study will facilitate the future researches in solar adsorption continuous refrigeration technology because achieving continuous cooling only with solar energy is very attractive.

Keywords: Adsorbent bed, adsorption refrigeration, CPC, solar cooling, solar adsorption.

1. INTRODUCTION

The technology advancement has changed the life style in the remote areas especially rural areas in the developing countries. The cooling demand is also increased for various purposes, such as storing foods and medicines, cold water for drinking, and for space cooling. Normally, electric driven systems are used for refrigeration or cooling purposes. But the energy shortfall and insufficient electric supply are the main constrains to use the electric driven cooling system in remote areas. Therefore, it is needed to produce cooling from renewable energy resources that are naturally available and environmental friendly. Many researches have been conducted to use the renewable energy resources for refrigeration purposes. Solar cooling can be promising and attractive technique in future because of near coincidence of peak cooling loads with the available solar power. There are many solar cooling techniques such as absorption, adsorption, desiccant, and ejector system. Solar adsorption refrigeration has advantages over other systems because it can bedriven with low heat [1] and does not require electric power. These systems are easy operated without high skills and required less maintenance, moreover, the working pairs used in the systems are environmentally friendly [2]. On the other hand there are some remote areas, especially local areas in developing countries where electricity is not enough but solar energy is rather easy to obtain. From this point of view, it can be said that solar driven adsorption refrigeration is technologically possible and socially feasible in such areas.

Despite the potential advantages, the solar adsorption refrigeration technology is not competent to replace the electric driven refrigerator because of low efficiency, intermittency, and high initial cost. A number of researches has been conducted to improve the performance of conventional one bed solar adsorption refrigeration system by increasing the heat and mass transfer capability of adsorbent bed using fins [3, 4], adding metal pieces into adsorbent bed [5], using high conductive adsorbent [6] and consolidated bed [7]. In most of the studies, flat plate collectors were used [3, 4] and some attempts were also carried out by using concentrator collectors [8] especially compound parabolic concentrator (CPC) [9].

The conventional system worked intermittently, it collects solar energy in the day time and produce cooling at night, because there is only one bed that acts as desorber in the day and adsorber in the night. Various studies have been conducted to produce continuous cooling with multi bed and advance cycle operation in adsorption cooling system technology. The performance of these systems were enhanced by using the advanced operations like internal mass recovery cycle [10, 11], heat and mass recovery cycles [12], multistage and cascade

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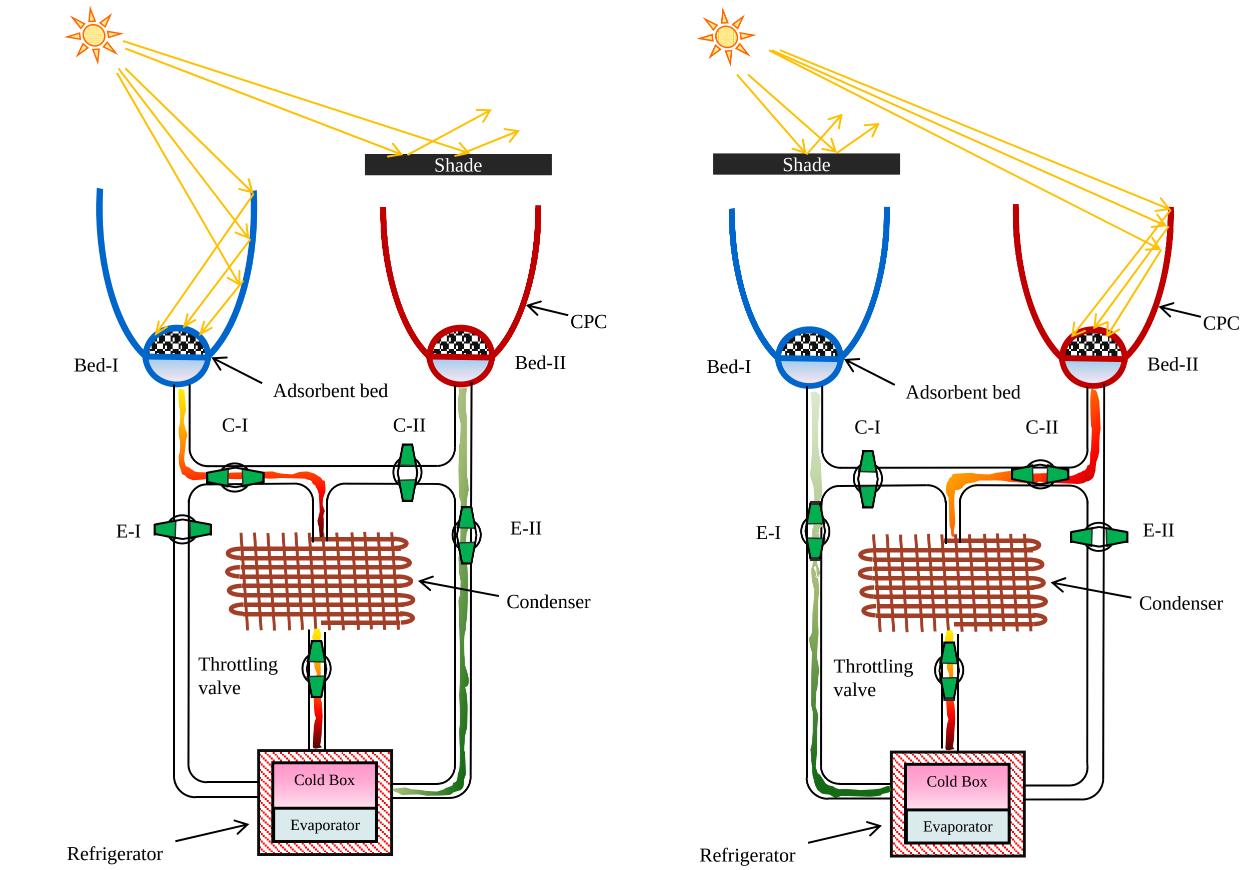


Fig. (1). Schematic diagram of the system.

cycles [13, 14]. In all these multi bed advance cycle systems, a continuous heating source is necessary to produce cooling continuously. However the solar energy is intermittent in nature and can be used only in day time, for night time, some backup heating system is required for continuous operation.

In order to produce cooling continuously only with solar energy, there should be at least two beds so that if one bed is desorbing the refrigerant then other should be in adsorbing mode. The simple and possible way is to complete the cycle in two days by shading one bed and connecting it with evaporator in first day and heating for desorption in the second day. There is negligible work on producing cooling continuously with solar driven adsorption system using only solar energy without involving electricity or any other backup system. The only review can be found by Hassan *et al*. [15]. They also suggested the two day cycle operation to produce continuous cooling assuming that the adsorption and rejection of the adsorption heat in the second bed is done at constant temperature. They used the assumed ambient and heat exchanger temperatures and conduct the thermodynamic analysis. But the solar driven adsorption cooling system is totally dependent on ambient conditions. The rate of desorption and adsorption changes with the ambient temperature and there is no such study according to our best knowledge that describe the realistic behaviour of such continuous system.

In this study, a solar adsorption refrigeration system is proposed that produce cooling continuously using only solar energy. A detailed simulation model was developed based on the heat and mass balance equations. The actual measured data of solar energy (beam and diffuse radiation), ambient temperature and wind velocity was used in the simulation. The objectives of the study were to investigate the real dynamic behaviour of the system, especially the adsorption rate during the day in the shaded bed, and to compare the performance of the proposed continuous system with that of conventional intermittent system. Solar adsorption cooling is already attractive technology because it can bedriven only by solar energy without using electricity, but intermittent operation (produce cooling only at night period) is one of its draw back. This study will further facilitate and motivate the researchers in the field of solar adsorption cooling systems, because achieving cooling continuously with only solar energy looks very attractive and promising technology in remote areas where electricity is not available but solar energy is in abandon to use.

2. SYSTEM DESCRIPTION

The solar adsorption cooling system is based on the adsorption desorption property of the working pair. The activated carbon fibber (adsorber) and ethanol (refrigerant) was used as the working pair. The schematic diagram of solar adsorption continuous cooling system is shown in Fig. (**1**). The system consists of two generators with compound parabolic concentrator (CPC) in which the adsorbent bed was directly packed in the receiver of each CPC. There was one condenser and one evaporator. The CPC is a non-imaging concentrator. It consists of two parabolas and one receiver. In current study a two dimensional (2D) CPC with partially exposed tubular absorber was used. CPC can achieve higher temperature and

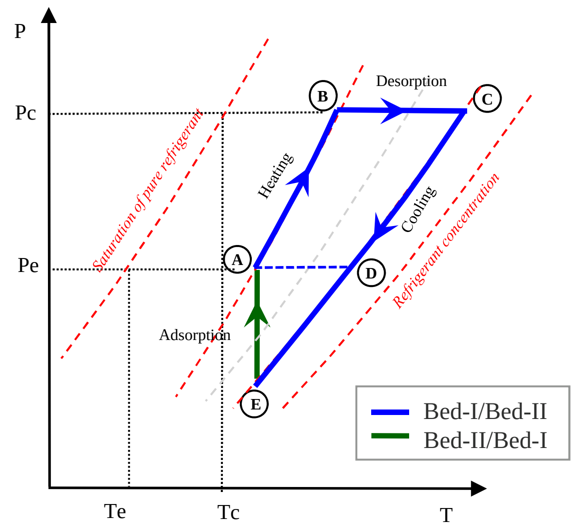


Fig. (2). Ideal adsorption cycle (Duhring diagram).

perform well in overcast situation compare to commonly used flat plate collectors [16]. Partially exposed receiver was reported to be favourable for solar adsorption cooling application as it give the advantage of heat rejection from back side during desorption and increases the amount of adsorbent per unit area [9]. Normally, CPC does not require tracking and it can accept incoming radiation over a relatively wide range of angles by using multiple reflections.

The system operation was controlled by the valves between the heat exchangers. The conventional solar adsorption cooling system consists of one bed that acts as desorber during day time and adsorber at night period. In this way one cycle is completed in one day and cooling is produced only during night.

In the present continuous system, there were two beds with separate CPC. One bed was shaded so that it cannot receive solar energy and connected with evaporator while the other bed was heated with solar energy and desorbs the refrigerant. The refrigerant comes to the condenser where it condensed and enters the evaporator through throttling valve. In evaporator the refrigerant evaporates and adsorbed in the bed that was shaded, due to evaporation cooling is produced in the refrigerator. Next day, the shaded bed was open and other bed was shaded. The same process was repeated but in opposite beds. In simple words, one bed desorbs the refrigerant in the first day and adsorbs in the second day, so one cycle is completed in two days but continuous cooling is produced in the evaporator, because evaporator was always connected with one of the bed. In the refrigerator the evaporator was dipped into the water that converts into ice when evaporator temperature became 0 °C or below. Theices acts as cold storage and keep the cold box at low temperature.

The working cycle of the system can be explained by ideal Duhring (P-T-q) diagram (Fig. **2**) and schematic of the system (Fig. **1**). The cycle starts from ***A,*** when the sun rises and the collector heated the Bed-I (A-B). The process is called pre-heating (PH). During this process valve C-I and E-I are closed. At certain point ***B***, when the pressure of Bed-I becomes equal to the condensation pressure of the refrigerant, the refrigerant starts desorbing from the Bed-I in vapour form. The Bed-I is connected to the condenser (valve C-I opens) where the refrigerant condensed and enter the evaporator, while Bed-I continuously heating at constant pressure. This process (B-C) is called desorption (DS). In the late afternoon (point C) when the solar radiation are not enough to continue further desorption process, the Bed-I is disconnected from the condenser (C-I closed) and allowed to cool down until next morning(C-E). This process is called precooling (PC).The next day, in the morning, when Bed-II is at ***A***, the Bed-I is connected with evaporator and concentration of refrigerant increasing in Bed-I(E-A), that process is adsorption (AD).

In one day operation, Bed-I follow the processes from A-B-C-E and Bed-II complete only adsorption process from E-A, the next day processes are interchanges in beds that is Bed-II follow A-B-C-E and Bed-I follow E-A.

The conventional intermittent system follows the cycle ABCDA and adsorption is done at constant pressure while the proposed continuous system follows the cycle ABCEA and adsorption is considered at constant temperature. The different operating processes in conventional and proposed system are shown in Fig. (**3**) with process timing.

3. MATHEMATICAL MODELS

A dynamic simulation model was developed for CPC collectors [9, 17, 18], adsorption system and for the performance evaluation [19-22] based on mass and energy balance equations.

Assumptions

The following main assumptions were made to simplify the model:

- The CPC is ideal and free from the fabrication errors.
- The heat transfer from receiver surface to adsorbent bed is uniform and given constant value.
- The adsorbent particles have uniform size, shape and distribution.

Time [h]	6	9	16	20	6
Bed	PH	DS	PC	AD	
	One cycle in one day				

a) Conventional one bed intermittent system

Time [h]	6	9	16	20	6	9	16	20	6
Bed-I	PH	DS	PC		AD				
Bed-II	AD				PH	DS	PC		
	First Day [1st half cycle]				Second Day [2nd half cycle]				

b) Present two bed continuous system

Fig. (3). Operating processes and timing.

- All specific heats of the components and coefficient of heat transfer are assumed to be constant.
- The pressure is uniform in refrigerant flowing channel.
- The bottom side of the receiver of CPC is well-insulated, and there is no heat loss to the surroundings.

Governing Equations

Adsorption Isotherms

$$W = W_0 exp\left\{-D\left[Tln\left(\frac{P_s}{P}\right)\right]^2\right\} \tag{1}$$

Adsorption Kinetics

$$\frac{dq}{dt} = k_s a_v (W - q) \tag{2}$$

CPC Cover Temperature

$$\left(M_c C_{p,cov}\right)\frac{dT_{cov}}{dt} = \alpha Q_{cov} + \left(h_{Rr} + h_{rc}\right)\left(T_r - T_{cov}\right) - h_{Rs}\left(T_{cov} - T_{sky}\right) - h_{ca}\left(T_{cov} - T_{air}\right) \tag{3a}$$

Q_c is the energy absorbed by cover and calculated as;

$$Q_c = A_{cov}\left(I_{diff+}I_{bm}cos\theta_i\right)\left(\alpha_{cov} + \alpha_{cov}\tau_{cov}\rho_r\rho_{cpc}^{nr}\right)\frac{A_{cov}}{A_r} \tag{3b}$$

CPC Receiver Temperature

$$\left(M_r C_{p,Cu}\right)\frac{dT_r}{dt} = \alpha Q_r - (h_{Rr} + h_{rc})(T_r - T_{cov}) - U_{bed}A_r(T_r - T_{bed}) \tag{4a}$$

Q_r is the energy absorbed by the receiver and is calculated as;

$$Q_r = A_{ap}\left(F I_{bm} cos\theta_i \tau_{cov}\tau_{cpc}\alpha_r + I_{diff}\tau_{diff}\tau_{cpc}\alpha_r\right) \tag{4b}$$

F and *α* are the control parameters. α is 0 when CPC is shaded and 1 when CPC is open to receive solar energy. *F* is 1 when the incident angles of beam radiation are within the acceptance range of CPC otherwise *F* is 0. The conditions can be found from reference [18].

Adsorbent Beds Temperature

$$\left(M_{ACF}\, C_{p,ACF} + q^d M_{ACF}\, C_{p,re}\right)\frac{T_{bed}^d}{dt}$$

$$= U_{bed}A_{bed}\left(T_r^d - T_{bed}^d\right) + \beta\left(\Delta H_{st}\, M_{ACF}\frac{dq^d}{dt} + M_{ACF}\right.$$

$$\left. C_{p,re}\left(T_{bed}^d - T_c\right)\frac{dq^d}{dt}\right) \tag{5}$$

$$\left(M_{ACF}C_{p,ACF} + q^a M_{ACF}C_{p,re}\right)\frac{T_{bed}^a}{dt}$$

$$= U_{bed}A_{bed}\left(T_r^a - T_{bed}^a\right) + \beta\left(\Delta H_{st}\, M_{ACF}\frac{dq^a}{dt} - M_{ACF}\right.$$

$$\left. C_{p,re}\left(T_{bed}^a - T_e\right)\frac{dq^a}{dt}\right) \tag{6}$$

β is 0 when beds are in pre-cooling or pre-heating mode and 1 when there is desorption or adsorption in the beds. The temperature of the cover, receiver, and adsorbent bed of each part of CPC is calculated simultaneously using above mentioned equations with respective parameters.

Condenser Temperature

$$\left(M_c\, C_{p,Cu} + M_{re,c}\, C_{p,re}\right)\frac{T_c}{dt} = -U_c A_c (T_c - T_{air}) +$$

$$\beta\left(-L_{H,re}\, M_{ACF}\frac{dq^d}{dt} - M_{ACF}\, C_{p,re}\left(T_{bed}^d - T_c\right)\frac{dq^d}{dt}\right) \tag{7}$$

Evaporator Temperature

$$\left(M_e C_{p,Cu} + M_{re,e}C_{p,re}\right)\frac{T_e}{dt} =$$

$$-U_e A_e (T_e - T_{air}) - U_w A_{wt}(T_e - T_w) - L_{H,re}\, M_{ACF}\frac{dq^a}{dt} +$$

$$M_{ACF}\, C_{p,re}\left(T_{bed}^a - T_e\right)\frac{dq^a}{dt} \tag{8}$$

Water Temperature in Refrigerator

$$\left(M_w C_{p,w}\right)\frac{T_w}{dt} = \psi\left(U_w A_{wt}(T_e - T_w) - U_\beta A_{wt}(T_w - T_{air})\right) \tag{9}$$

ψ is 0 when evaporator temperature is 0 °C or below otherwise it is 1.

Ice Production

$$L_{H,w}\frac{dM_{ice}}{dt} = \lambda L_{H,re}M_{ACF}\frac{dq^a}{dt} \tag{10}$$

Table 1. Numerical values used in simulations.

Symbol	Value	Unit	Symbol	Value	Unit
Adsorption Isotherm			**Adsorbent Beds**		
W_o	0.707	kg kg^{-1}	M_{ACF}	changing	kg
D	1.716 ×10^{-6}	K^{-2}	Cp_{ACF}	941	J kg^{-1} K^{-1}
CPC Characteristics			Cp_{re}	2400	J kg^{-1} K^{-1}
A_{ap}	0.25/per CPC	m^2	ΔH_{st}	1.2×10^6	J kg^{-1}
A_r	0.098	m^2	**Condenser**		
L_{cpc}	2/per CPC	m	M_c	2	kg
nr	0.68	-	A_c	0.4	m^2
ρ_{cpc}	0.92	-	h_{cc}	15	W m^{-2} K^{-1}
τ_{cpc}	$(\rho_{cpc})^{nr}$	-	$L_{H,re}$	8.46×10^5	J kg^{-1}
Cover			**Refrigerator**		
M_{cov}	0.9	kg	M_e	4	kg
Cp_{cov}	840	J kg^{-1}	A_e	0.86	m^2
A_{cov}	0.372/per CPC	m^2	A_{wt}	0.37	m^2
α_{cov}	0.05	-	U_e	0.6	W m^{-2} K^{-1}
τ_{cov}	0.89	-	U_w	5	W m^{-2} K^{-1}
Receiver			U_β	0.2	W m^{-2} K^{-1}
M_r	5	kg	$L_{H,,w}$	2.5 ×10^6	J kg^{-1}
Cp_{Cu}	386	J kg^{-1} K^{-1}	$Cp_{,w}$	4200	J kg^{-1} K^{-1}
U_{bed}	changing	W m^{-2} K^{-1}	-	-	-
A_r	0.95	-	-	-	-
ρ_r	0.15	-	-	-	-

λ is 1 when evaporator temperature is 0 °C or below otherwise it is 0.

Mass Balance of Refigerant

$$\frac{dM_{ree}}{dt} = -M_{ACF}\left(\frac{dq^a}{dt} + \frac{dq^a}{dt}\right) \tag{11}$$

Performance Index

The performance of the system is evaluated with specific cooling effect (SCE) and coeffcient of performance (COP) that are calculated as;

$$SCE = \frac{Q_e}{M_{ACF},total} \tag{12a}$$

where Q_e is the cooling energy achieved by the evaporator and calculated by multiplying the total amount of adsorbed refrigerant with the latent heat of refrigerant;

$$Q_e = M_{adsorb} \times L_{H,re} \tag{12b}$$

and

$$M_{adsorb} = M_{ACF}(q_{max} - q_{min}) \tag{12c}$$

$$COP = \frac{Qe}{Aap\int_{end\ of\ desorption}^{sunrise}(I_{bm}cos\theta_i + I_{diff})(t)\,dt} \tag{13}$$

The supplementary equations to calculate the h_{Rr}, h_{Rs}, h_{rc} and h_{ca} used in the mathematical models are given in reference [16, 21]. The numerical values used in the simulation are given in Table **1**.

Initial Conditions

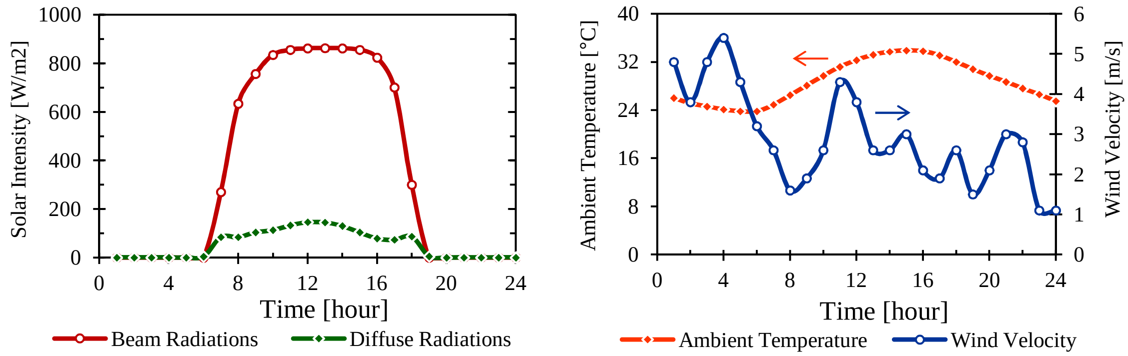
$$T_{cov}(0) = T_r(0) = T_{bed}(0) = T_c(0) = T_{air} \tag{14}$$

and,

$$T_e(0) = T_w(0) = 285\ K \tag{15}$$

Boundary and Operating Conditions

The actual measured data for solar radiation, ambient temperature and wind velocity was used in the simulation for



(a) Solar radiations

(b) Wind velocity and Ambient temperature

Fig. (4). Weather data used in simulation (for Tokyo, August 28).

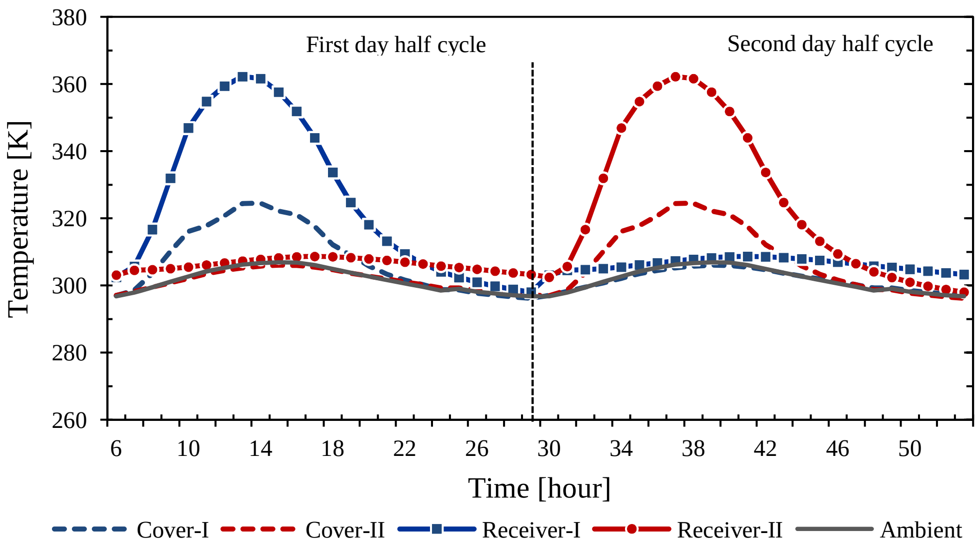


Fig. (5). Cover and Receiver temperature profile of CPCs.

Tokyo for a sunny day on August 28 (Fig. **4**). The data was obtained from the commercial software Meteonorm v 6.1. The system was operated according to the fixed time schedule shown in Fig. (**3**).

Numerical Solution of the Mathematical Models

The mathematical models developed for different components of the system were used to build a computer algorithm in the commercial computer software MATLAB R2010b. Ordinary differential solver (ode45) tool was used to incorporate the differential equations in the simulation model. Graphs were prepares in Microsoft Excel using simulation result data from MATLAB.

4. RESULTS AND DISCUSSIONS

The simulation model was run using the actual measured data to get the realistic behaviour of the system. First day CPC-I collected the solar energy while CPC-II was shaded, next day they were interchanged. The Fig. (**5**) shows the temperature of cover and receiver of CPC. It can be noted that the temperature of cover in shaded CPC is following the ambient temperature, while for the same shaded CPC, receiver temperature is higher than the ambient temperature. This is because the shaded bed is connected with the evaporator and adsorption is occurring in that bed. Adsorption is exothermal process and heat is released during the adsorption. Due to adsorption heat, temperature of the receiver also increases because the bed is directly packed into the receiver tube.

Fig. (**6**) show the temperatures of the main heat exchangers. It can be noted that the temperature of desorbing Bed-I in the first day (Bed-II in the second day) is lower than its receiver temperature (Fig. **5**). Desorption is endodermal process and heat is absorbed from the soundings, also the adsorbent is the poor heat conductive; therefore, the bed temperature is lower than the receiver temperature. The condenser temperature increase upto 40 °C, this high tempera-

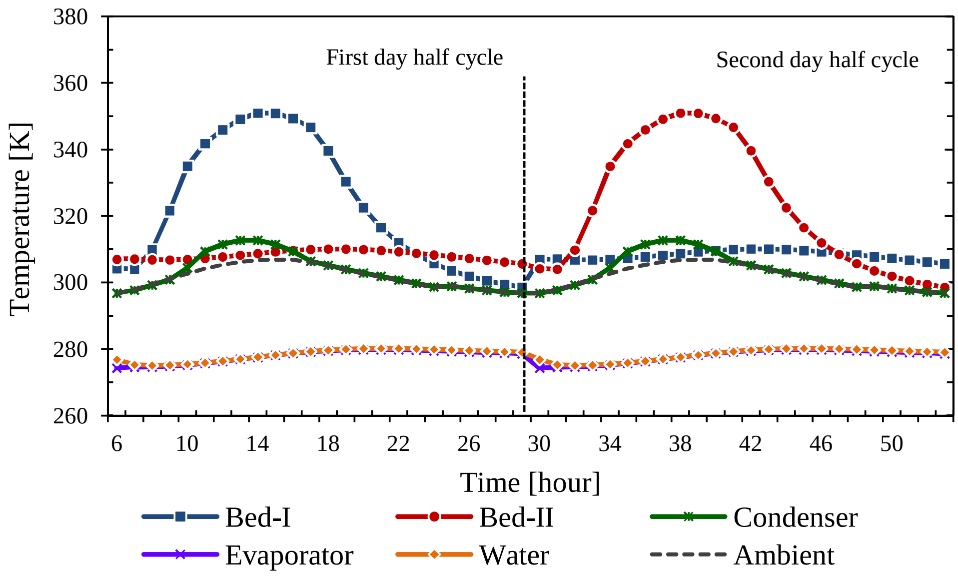


Fig. (6). Heat exchanger temperature profile of system.

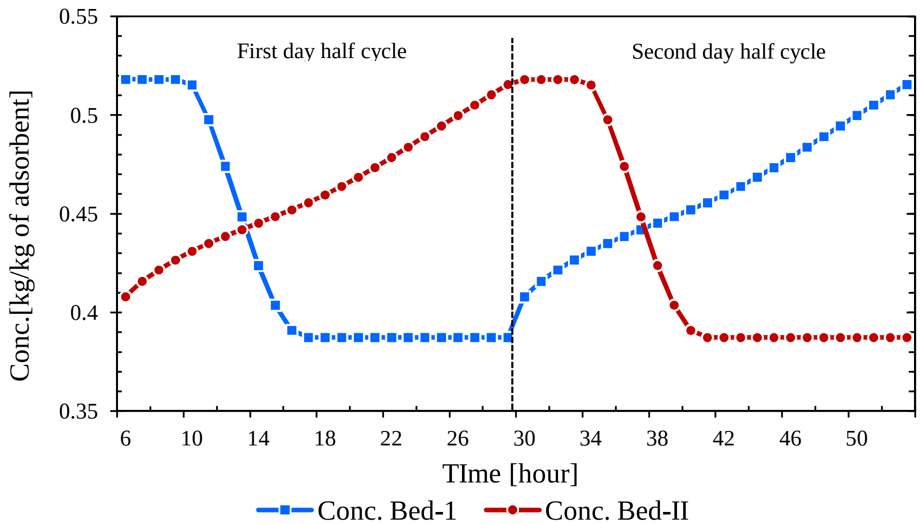


Fig. (7). Concentration of the refrigerant in the beds.

ture is because the condenser was naturally air cooled. The important was the evaporator temperature during cycle. The results show that system kept low evaporator temperature indicating continuous cool production. It can be observed that the evaporator temperature is changing with time. This variation is because of the corresponding adsorption rate (Fig. **7**). It can be seen from Fig. (**7**) that adsorption rate in the shaded bed is high in the start of the day, and decreases in the middle of the day.

During the day time when there is desorption in first bed and simultaneous adsorption in the shaded bed, the temperature of desorbed refrigerant entering to the evaporator is higher. On the other hand, the temperature of shaded bed is also high due to high ambient temperature. As a result the adsorption rate during the day period, when there is desorption in other bed, becomes slow and ultimately the evaporator temperature increase. In the late afternoon when desorption process stopped, the temperature of the shaded bed decreases. As a result adsorption rate again increases and evaporator temperature decrease.

Similar behaviour can be seen more clearly (Fig. **8**) in the specific cooling effect (SCE) of the system for one day operation. Fig. (**8**), show the change in SCE per unit time interval. It can be observed that the SCE is highest in the starting of the day when bed is connected with evaporator. The cooling rate decreases in the peak hot period of the day and again increases from the evening to the next morning, the reason of decreasing the cooling rate in the middle of the day is according to the adsorption rate as discussed above. In the simulation model it was set that if the evaporator temperature become 0 °C or below, ice produced and water temperature remain constant at 0 °C. It can be observed from Fig. (**6**) that water temperature is following the evaporator temperature and does not lower down below 0 °C. This indicates that no ice was produced through out cycle.

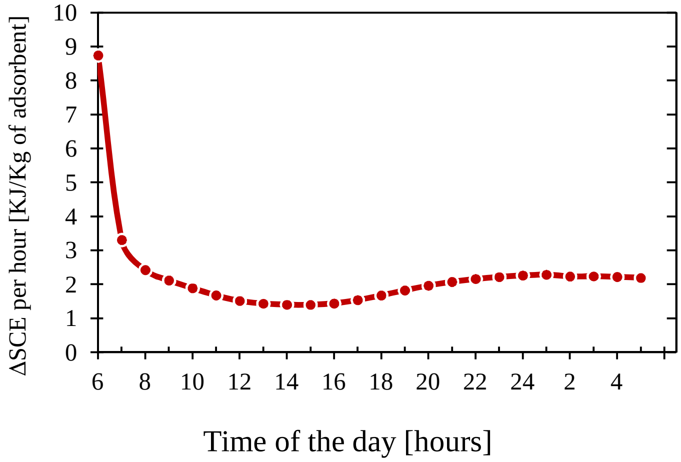


Fig. (8). Variation in SCE per unit hour of the operation time.

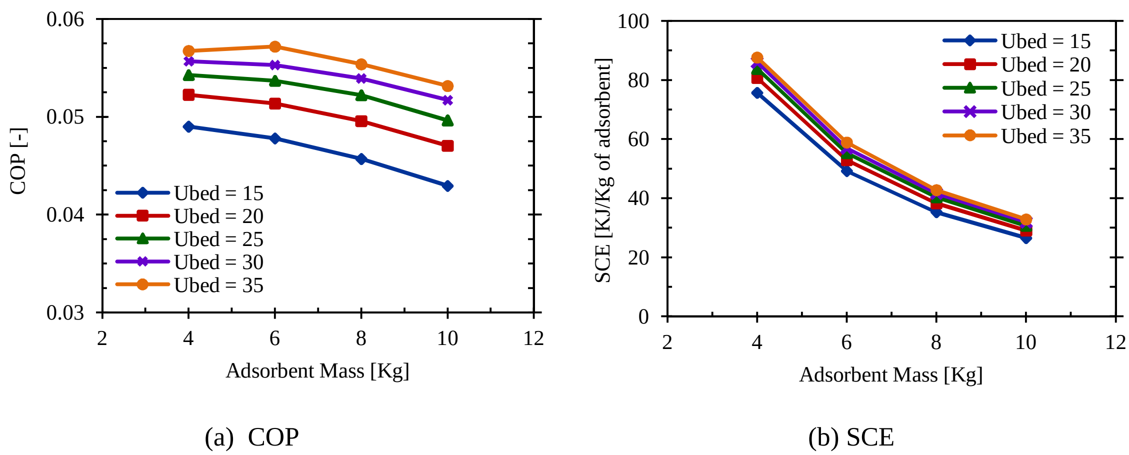


Fig. (9). Effect of adsorbent mass and U_{bed} on the system performance.

The performance of the system was determined in term of coefficient of performance (COP) and specific cooling effect (SCE) for one day half cycle. Fig. (**9**) shows the effect of adsorbent mass (M_{ACF}) and the heat transfer coefficient between receiver of CPC and adsorbent bed (U_{bed}) on the performance of the system. Both the COP and SCE decreases when the mass of adsorbent bed increases. The adsorbent is poor heat conductive in nature, when the mass of adsorbent increased, the thickness of the bed also increases and heat transfer to the end of the bed becomes slow that results in poor desorption adsorption process. Ultimately, the performance of the system decreases. The performance of the system increases with high value of heat transfer coefficient (U_{bed}). The heat transfer coefficient can be increases practically by using the high conductive adsorbent or using heat transferring medium like fins inside the adsorbent bed.

In sum, the solar adsorption refrigeration system with two beds and completing cycle in two days can produce cooling continuously but the cooling rate decreases in the day period because of high ambient temperature. This can be improved by using some heat rejecting medium to the shaded bed like opening the CPC cover or attaching fins to external surface of the receiver of shaded CPC.

5. COMPARISON BETWEEN INTERMITTENT AND PRESENT CONTINUOUS SYSTEM

The performance of the presented continuous system was compared with the performance of conventional intermittent system. It should be noted that there is clear difference between the two systems. The intermittent system produces cooling only during night while proposed system produced cooling continuously. The objectives of this comparison were to investigate the behaviour of evaporator temperature and cooling capacity of the system when operated in continuous and intermittent mode. The cooling rate highly depends on the amount of adsorbent and size of the bed connected to the evaporator. In continuous system there were two beds but only one bed was connected with evaporator at a time. Therefore, two criteria were set to compare the performance of continuous system with intermittent cycle. In case-a, the size of the bed in intermittent system was kept equal to the sum of the two beds in continuous system. In

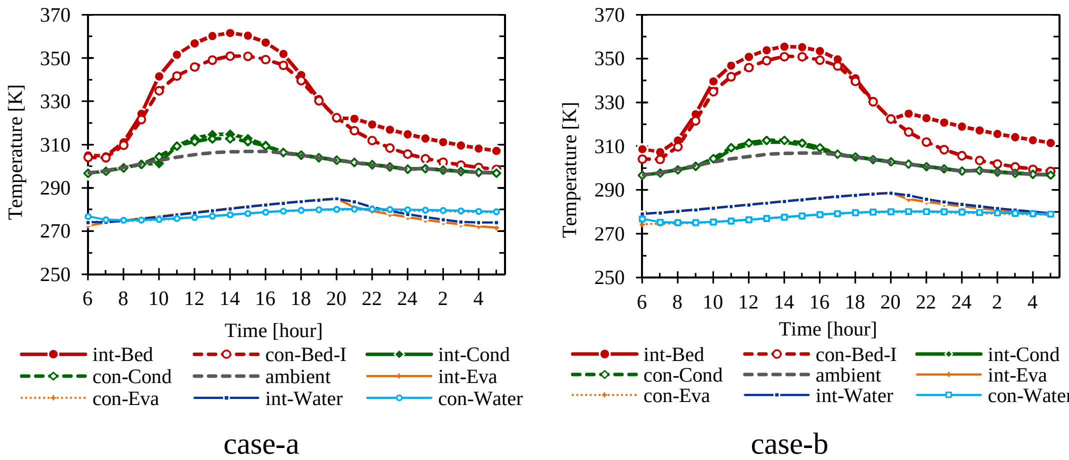


Fig. (10). Comparison between system temperature of intermittent and present continuous system.

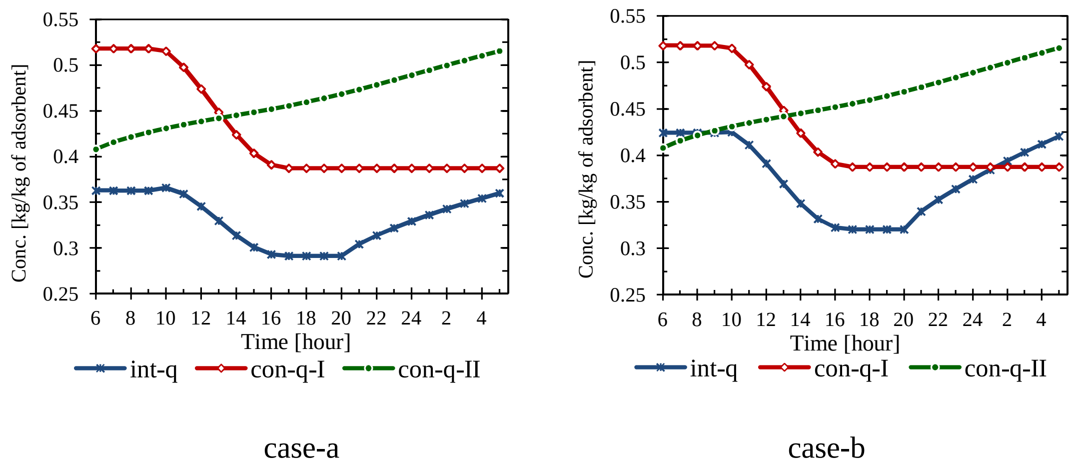


Fig. (11). Comparison betweenrefrigerant concentration in the beds of intermittent and present continuous system.

this way both system has same size but the bed size connected to the evaporator in intermittent system become double compare to the bed size connected to evaporator in continuous system. Therefore, in case-b, the size of the bed in intermittent system was kept equal to the one bed in continuous system. In this way, bed size connected with evaporator was same in both systems but total size of beds become twice in the continuous system compare to the intermittent system.

Fig. (**10**) shows the comparison between system temperatures of intermittent and continuous system for one day operation. It can be noted that the bed temperature for continuous system (res dotted line) is lower than the bed temperature of intermittent system (red complete line) in both cases. This is because the more amount of refrigerant is desorbing at faster rate from the continuous bed compare to the intermittent bed as shown from Fig. (**11**). Desorption is endothermic process, therefore, the bed temperature decreases according to desorption rate. There is no significant difference in condenser temperature of intermittent and continuous system in both cases. The evaporator temperature for intermittent system is different for two cases compare with continuous system. For the case-a, where the bed size is same in both systems, the difference in evaporator temperature for intermittent and continuous system is small. But still the continuous system maintains the low temperature throughout the day compare to the intermittent system where the evaporator temperature increases during day. It should be remember that in intermittent system cooling is not produced during the day time because evaporator is connected only at night time, while in continuous system the evaporator is connected all the time to shaded bed and cooling is produced continuously as shown in Fig. (**12**).

On the other hand, for the case-b, where the same bed sizes were connected with evaporator, there is significant difference in evaporator temperatures. The continuous system always keeps the evaporator temperature lower than evaporator temperature in intermittent system, and maximum difference of 10 °C was found between two temperatures.

It can be summarized from the above comparison that if the same amount and size of the system is considered, con-

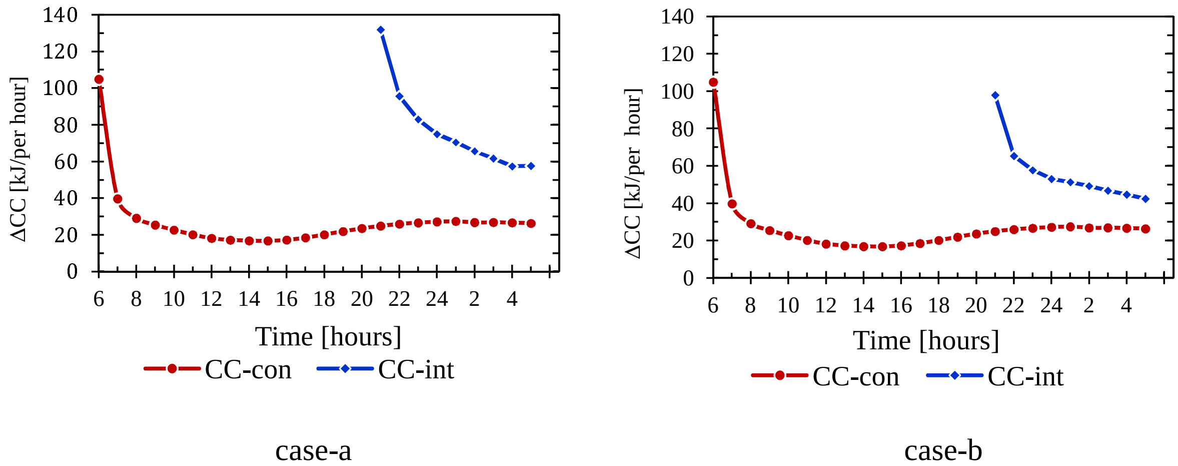


Fig. (12). Comparison between change in cooling capacity of intermittent and present continuous system.

tinuous cooling can be achieved but effect is small. If continuous cooling is the primary objective than double size of the bed is required for significant effect.

6. CONCLUSION

In this article, simulation study of a solar adsorption refrigeration system was presented that produce cooling continuously using only solar energy as driving force. The conventional one bed system work intermittently, day time it desorbs refrigerant and produced cooling only at night time. But basically cooling demand increases during the day time when outside is hot. The proposed system consists of two beds with CPC collectors, a condenser and evaporator. In one day operation one bed was shaded and connected with evaporator while other works normally. Next day the beds were interchanged and same operation was repeated. In this way one cycle completed in two days and continuous cooling produces because evaporator was always connected with one of the beds. There are very fewer studies related to such continuous system, but no work was found that investigate the actual behaviour of the solar adsorption refrigeration system producing cooling continuously using only solar energy. The objectives of the study were to investigate the actual behaviour of the system especially the adsorption behaviour during day period when ambient temperature is high, and to compare the performance of the system with conventional intermittent system. A detailed simulation model was developed that uses the actual measured weather data for Tokyo, Japan. The simulation results can be summarised as follow;

- The system achieves low temperature in the evaporator throughout the day and night, although the cooling capacity is low but continuous cooling is attractive.
- The cooling rate changes according to the adsorption rate of refrigerant. In the peak day hours when the ambient temperature was relatively higher, the adsorption rate became slower because of slow heat rejection to ambient. The temperature of the shaded adsorbent bed increases when the adsorption rate increases.
- The ice was not produce during the process indicating the amount of desorbed refrigerant that were coming to evaporator was not enough to bring the water temperature below 0 °C.
- The estimated specific cooling effect of the system was found to be110.5 kJ/kg of adsorbent in one bed that is equivalent 7.67 W/day or 1.27 W/Kg of adsorbent or 30.7 W/m^2 of the each CPC aperture area for L_{CPC} = 2m/CPC, M_{ACF} = 3kg/L_{CPC} and U_{bed} = 25W m^{-2} K^{-1}
- The performance of the system increases with high value of coefficient of heat transfer between CPC receiver and adsorbent bed (U_{bed}). This means the use of heat transferring medium into bed and adsorbent with high thermal conductivity are preferred. On the other hand the performance decreases by increasing the adsorbent mass (M_{ACF}).
- The performance of the continuous system was compared with performance of intermittent system. The significant difference is the intermittent system produce cooling only during night while the proposed system produced continuous cooling. For the case when both system has same bed size the difference in evaporator temperatures was small compare to the case when evaporator was connected with same size of beds in both system. It was found that continuous system keeps the evaporator temperature maximum 10 °C lower than the intermittent system.

It can be concluded that the continuous cooling can be achieved with such system. But still more investigation is needed to evaluate the system performance, specially the experimental investigation of adsorption rate in the shaded bed during day time. Some heat rejecting mechanism is needed for the shaded bed to increase the adsorption rate that can be done by opening the cover or attaching the fins to the shaded CPC. System can be used for refrigeration or cooling purposes but difficult for continuous ice making until some high efficient working pair with optimized system design is used. This study will facilitate and motivate the future researches in solar adsorption continuous refrigeration technology because achieving continuous cooling even at low rate; only with solar energy is very attractive, especially in the developing and remote areas.

ABBREVIATIONS AND SYMBOLS

A	area (m^2)	Subscripts	
C_P	specific heat (J kg^{-1} K^{-1})	*ACF*	activated carbon fibber
CC	cooling capacity (kJ)	*air*	air
con	continuous system	*ap*	aperture
D	exponential constant (K^{-2})	*bed*	adsorbent bed
F	control factor (1 or 0)	*bm*	beam
h_{Rr}	radiation heat transfer coefficient between receiver and cover (W m^{-2} K^{-1})	*c*	condenser
h_{Rs}	radiation heat transfer coefficient between cover and sky (W m^{-2} K^{-1})	*cov*	CPC cover
h_{rc}	convective heat transfer coefficient between cover and receiver (W m^{-2} K^{-1})	*cpc*	CPC collector
h_{ca}	convective heat transfer coefficient from cover due to wind (W m^{-2} K^{-1})	*Cu*	Copper
ΔH_{st}	heat of adsorption/desorption (J kg^{-1})	*diff*	diffuse
I	solar radiation (W m^{-2})	*e*	evaporator
int	intermittent system	*ice*	ice
$k_s a_v$	overall mass transfer coefficient (S^{-1})	*re-e*	refrigerant in evaporator
L_H	latent heat of vaporization (J kg^{-1})	*re*	refrigerant ethanol
M	mass (kg)	*w*	water
P	pressure [Pa]	*wt*	water tank
Ps	saturation pressure [Pa]	Superscripts	
q	instantaneous uptake (kg kg^{-1})	*a*	adsorption
SCE	specific cooling effect [k J kg^{-1}]	*d*	desorption
T	temperature [K]	*nr*	average number of reflections
U_{bed}	overall heat transfer coefficient between receiver and adsorbent bed (W m^{-2} K^{-1})		
U_e	overall heat transfer coefficient between evaporator and ambient (W m^{-2} K^{-1})	Greek letters	
U_w	overall heat transfer coefficient between evaporator and water tank (W m^{-2} K^{-1})	α	absorptance
U_β	overall heat transfer coefficient between water tank and ambient (W m^{-2} K^{-1})	θ_i	angle of incident
W	uptake of refrigerant (kg kg^{-1})	ρ	reflectance
W_o	maximum uptake (kg kg^{-1})	τ	transmittance

CONFLICT OF INTEREST

The authors confirm that this article content has no conflicts of interest.

ACKNOWLEDGEMENTS

Declared none.

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Received: January 12, 2014 Revised: February 15, 2014 Accepted: February 20, 2014