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Energy Management and Heat Storage for Solar Adsorption Cooling

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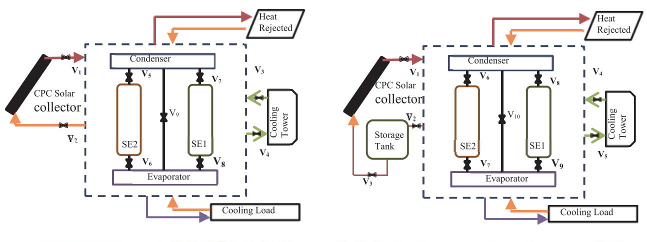
A solar heat driven cooling system utilizes abandoned solar radiation, plays an important role in preservation of primary energy, prevents sound pollution and chlorofluorocarbon (CFC) /hydro-chlorofluorocarbon (HCFC) free environment. Other than the installation and commissioning cost, the running cost of this system can be reduced to its bare minimum by maximum exploitation of the collected solar energy. This article points out a way how collected heat energy can be properly utilized to run a conventional two bed solar adsorption cooling system to produce optimum cooling energy ensuring prolonged coverage. Increased solar thermal units can enhance heat collection up to 864.1006 MJ/75.68 m^2 of collector area. Enlarged storage tank provides backup for longer working hours, 14 hours a day. On the other hand, a smart choice of operating conditions can ensure a higher cooling capacity of 16.1 kW/2.197m^3 heat storage tank at peak hours and a comfortable, steady cooling effect over a relatively longer duration. Proper management of collected energy can turn out to be an economic gain factor in a developing country like Bangladesh. It could save roughly BDT 9324 ≈ \$116.55/year for only 1 RT space cooling purpose. Moreover, it lowers CO_2 emission and preserves primary energy and electricity.

Keywords: solar heat, adsorption chiller, energy management, green energy.

1. Introduction

At present, in the field of energy systems the study in various energy conversion systems mainly heat pumps, sorption systems, energy conversion and storage devices are in the top priority. Adsorption refrigeration and air conditioning cycles have earned considerable attention due to its ability to utilize low temperature heat source and for the environmental aspects as it uses environment friendly refrigerants. The advantage and development of adsorption cycle have been widely studied by Meunier [1]. Later, researchers have made development to adsorption technology. In this respect, some have considered the improvement of the coefficient of performance (COP values) while the others focused on the system cooling capacity. Adsorption technology had also been utilized for desalination and water treatment purposes [2-5]. Advanced cascaded cycle [6], thermal wave cycles [7] have been introduced for the enhancement of COP values. While mass recovery cycle [8-9] is for improvement of system cooling capacity. Advanced multiple-bed system [10], such as three-stage [11] and two-stage [12] cycles could be effective for utilization of low temperature heat source. Uddin et al.



a) b)

Fig. 1: Schematic diagram of solar adsorption cooling system a) with direct solar coupling and b) system with heat storage

[13] published thermodynamic analysis of adsorption cooling cycle for Ethanol surface treated Maxsorb III pairs.

Adsorption technology with solar coupling could be one of the attractive and alternative energy source to produce necessary cooling instead of conventional energy source. Sakuda and Suzuki [14], Leite and Daguenet [15], Boubarkri [16] studied solar ice making with adsorption technology coupled with solar heat collectors. Yang and Sumanthy [17] first exploited the lumped parameter model for two beds adsorption cycle driven by solar heat. Later, Clausse et al. [18] investigated the performances of a small adsorption unit for residential air conditioning in summer and heating during the winter period for the climatic condition of Orly, France. And Zhang et al. [19] investigated the operating characteristics of silica gel-water pair as adsorbent/ adsorbate utilizing solar powered adsorption cooling system. Recently Alam et al [20] investigated the performances of solar collector driven adsorption cooling system under the climatic condition of Tokyo, Japan. A similar study has been carried out by Rouf et al [21] for the climatic condition of Dhaka, Bangladesh. Later, effect of the operating conditions for a two bed basic adsorption cycle with silica gel-water pair powered by solar heat has been investigated [22].

However, this system has a vital setback. Intermittent solar energy cannot provide tangible support to run such a system unhampered. Also even if solar radiation is available for few sunny days, the system can work only for a limited time, as long as there is enough radiation available to provide sufficient thermal heat to run the chiller. Various options have been investigated to work out this problem such as natural gas or electric vapor compression chiller, thermal storage (cold or hot); a mix between these options. F. Meunier [23] recommend, a conventional electric vapor compression chiller backed by sorption unit when solar energy is present. Ammar et. al. [24] investigated analytically the performance of different adsorbent for tubular adsorber for solar powered adsorption refrigeration system in sub-Sahara region of Algeria. This paper addresses the option of adsorption solar air conditioning with hot thermal storage. This problem is same as liquid absorption.

Alam et al. [25] discussed utilization of heat storage as a backup heat supplier after sunset. Rout et al. [26] compared the longer working capacity of the chiller with heat storage and that of a chiller with direct solar coupling. Kim et al. [27] studied economic aspects of a solar hot water plant. Present study investigates a standard size of the heat storage in need of maximum heat collection and preservation. And thus calculates optimum cooling capacity with base run conditions. For intermittent heat source like solar radiation, choice of erratic cycle time can play an interesting role and increase cooling capacity of the system. Additionally, a controlled flow of chilled water can ensure a steady cooling effect to the end user. The investigation is conducted on a two bed adsorption cooling system which is run by solar heat, with silica gel-water pair as adsorbent/ adsorbate under the climatic condition of Dhaka. The place is located in the northern hemisphere at $23^{\circ}46'$ N (latitude), and $90^{\circ}23'$ E (longitude).

2. System description

A conventional basic adsorption chiller consisting two adsorbent beds, one condenser and one evaporator has been considered. The chiller configuration is same as Saha et al. [28] where silica gel-water pair has been utilized as adsorbent/adsorbate pair. The principle of basic adsorption cycle is available in this literature. The operating conditions are presented in Table 1. Solar collector data, compound parabolic concentrator CPC1509 manufactured by Ritter Solar, are utilized as heat source for the chiller. Solar radiation data has been supported by renewable energy research center (RERC) of University of Dhaka. The working principle of the present chiller is available in [25]. The schematic diagram of the chiller is given in Fig. 1.

The position of SE1 and SE2 (adsorption beds) during the different mode in a full cycle is represented in table 2. For the system with heat storage, the heat transfer fluid (water) is heated in the solar collector and transported to the desorber. Desorber gains heat and the outflow of this hot water from the desorber is collected in the storage tank. Storage tank supplies water to the collector where it gains heat and complete the cycle. In this article different dimension of the reserve tank has been investigated. Specification of the reserve tank is given in table 3.

Table 1. Design and the operating conditions used in the simulation

Symbol	Description	Value
A_{bed}	Adsorbent bed heat transfer area	2.46 m^2
A_{con}	Condenser heat transfer area	3.73 m^2
A_{cr}	Each collector area	1.72 m^2
A_{eva}	Evaporator heat transfer area	1.91 m^2
$C_{p,M(Al)}$	Specific heat of aluminum (Al)	905 J/kg.K
$C_{p,M(Cu)}$	Specific heat of copper (Cu)	386 J/kg.K
$C_{p,si}$	Specific heat of silica gel	924 J/kg.K
$C_{p,w,l}$	Specific heat of water (liquid phase)	4180 kJ/kg.K

$C_{p,w,v}$	Specific heat of water (vapor phase)	1890 J/kg.K
D_{s0}	Diffusion coefficient	2.54 cm^2/s
Ea	Activation energy	2330 kJ/kg
i	Number of pipe in each collector	9
L	Latent heat of vaporization (water)	2600 kJ/kg
$\dot{m}_{f,cool}$	Cooling water flow rate to adsorber	1.3 kg/s
$\dot{m}_{f,con}$	Cold water flow rate to condenser	1.3 kg/s
$\dot{m}_{f,hot}$	Total mass flow rate to CPC panel or to desorber	1.3 kg/s
Qst	Heat of adsorption (silica gel bed)	2810 kJ/kg
R	Water gas constant	46.2 kJ/kg.K
R_p	Particle diameter (Silica gel)	0.035 cm
U_{bed}	Heat transfer coefficient of each bed	1724.14 W/m^{2}K
U_{con}	Condenser heat transfer coefficient	4115.23 W/m^{2}K
U_{eva}	Evaporator heat transfer coefficient	2557.54 W/m^{2}K
$W_{con,w}$	Condenser refrigerant (water) inside condenser	0.0 kg
$W_{eva,w}$	Liquid refrigerant (water) inside evaporator initially	50 kg
W_{si}	Weight of silica gel in each bed	47 kg

3. Mathematical modeling

A lumped parameter model is exploited to investigate the performance of the cycle. It is assumed that the temperature, pressure and concentration throughout the adsorbent bed are uniform. Based on these assumptions the energy balance equation of the adsorbent bed, working as desorber or adsorber is as follows:

$$\frac{d}{dt}\left\{\left(W_M C_{pM} + W_s C_s + W_s q C_w\right)T_{bed}\right\} = Qst \cdot W_s \frac{dq}{dt} + \delta \cdot W_s C_{p,w,v}\frac{dq}{dt}\left(T_{eva} - T_{bed}\right) + \dot{m}_f C_f\left(T_{bed,in} - T_{bed,out}\right) + UlossAS_{bed}\left(T_{am} - T_{bed}\right) \quad (1)$$

$$T_{bed,out} = T_{bed} + (T_{bed,in} - T_{bed})EXP(-UA_{bed}/\dot{m}_f C_f) \quad (2)$$

where, δ equals to zero or one depending on whether adsorbent bed is working as desorber or adsorber. The energy balance for the condenser is represented by:

$$\frac{d}{dt}\left\{\left(W_{con,M} C_{con,M} + W_{con,w} C_w\right)T_{con}\right\} = -L \cdot W_s \frac{dq_d}{dt} + W_s C_{w,v}\frac{dq_d}{dt}\left(T_{con} - T_{bed}\right) + \dot{m}_{f,con} C_f\left(T_{con,in} - T_{con,out}\right) \quad (3)$$

$$T_{con,out} = T_{con} + (T_{con,in} - T_{con})EXP(-UA_{con}/\dot{m}_{f,con} C_f) \quad (4)$$

Energy balance for the evaporator is represented by:

$$\frac{d}{dt}\left\{\left(W_{eva,M} C_{eva,M} + W_{eva,w} C_{ml}\right)T_{eva}\right\} = -L \cdot W_s \frac{dq_a}{dt} + W_s C_{w,l}\frac{dq_d}{dt}\left(T_{eva} - T_{con}\right) + \dot{m}_{f,chill} C_f\left(T_{chill,in} - T_{chill,out}\right) \quad (5)$$

$$T_{chill,out} = T_{eva} + (T_{chill,in} - T_{eva})EXP(-UA_{eva}/\dot{m}_{f,chill} C_f) \quad (6)$$

Table 2: Performance of SE1 and SE2 during different mode of a cycle

cycle	Mode ->	Pre-cool	Adsorption/ evaporation	Pre-heat	Desorption/ condensation
1st half	A	SE1	---	SE2	---
1st half	B	---	SE1	---	SE2
2nd half	C	SE2	---	SE1	---
2nd half	D	---	SE2	---	SE1

Table3. Design and operating conditions of the reserve tank used in the simulation

Symbol	Description	Value
L	Dimension of the tank	0.7/1/1.3 m
W_{tv}	Volume of the tank	L^3 m^3
W_{wt}	Weight of water in reserve tank	W_{tv}x1000-10 kg
U_{tloss}	Reserve tank heat transfer loss coefficient	0.5 W/m^2K
A_{wt}	Reserve tank outer surface area	6xL^2 m^2
W_{tm}	Reserve tank metal weight	A_{wt}x0.005x2700kg

The mass balance of the refrigerant inside the Evaporator is expressed as:

$$\frac{dW_{eva,w}}{dt} = -W_s\left(\frac{dq_a}{dt} + \frac{dq_d}{dt}\right) \tag{7}$$

Adsorption rate of RD type silica gel-water pair is estimated by LDF model as;

$$\frac{dq}{dt} = k_s a_p\left(q^* - q\right) \tag{8}$$

Where, $k_s a_p$ is the overall mass transfer coefficient of the adsorbent/adsorbate pair and can be represented as;

$$k_s a_p = (15 * D_s)/\left(R_p\right)^2 \tag{9}$$

where, R_p is the adsorbent particle radius, D_s is the surface diffusivity and can be expressed by Arrhenius equation as

$$D_s = D_{s0} * exp\left(-E_a / RT\right) \tag{10}$$

the modified Freundlich (S-B-K) equation is used to present the adsorption isotherms of RD type silica gel-water pair as

$$q^* = AA.\left(P_s\left(T_v\right)/P_s\left(T_b\right)\right)^{BB} \tag{11}$$

where, $AA = A_0 + A_1T + A_2T^2 + A_3T^3$
and $BB = B_0 + B_1T + B_2T^2 + B_3T^3$.

The numerical values A_i's and B_i's are given in table 4.

Different numbers of collectors are combined in a panel. The heat transfer fluid is equally distributed to all the collectors. Each collector has nine pipes; water enters through the first pipe and the outlet of the first pipe

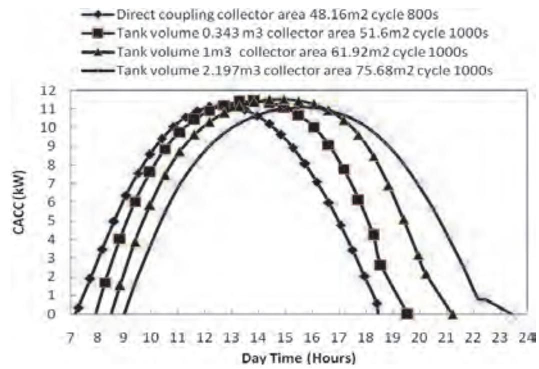


Fig. 2. Comparative cyclic average cooling capacity of the chiller with direct solar coupling and different dimension of storage tank with cycle time

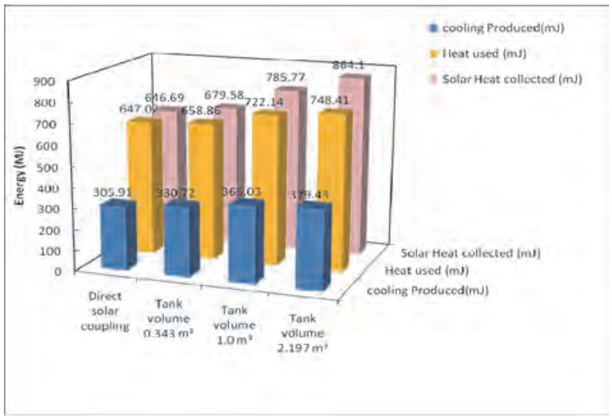


Fig. 3. Cooling load, heat in used and net heat collected of the chiller for different cases

flows into the next pipe. Thus, the outlet of the ninth pipe of each collector combines together and enters into the desorber. Hence, the temperature of the heat transfer fluid in each pipe is calculated separately for all the collectors. The energy balance of each collector can be expressed as:

$$W_{cp,i}C_{cr}\frac{dT_{cr,i}}{dt} = \gamma\left\{\eta_i A_{cr,i} I + \dot{m}_{f,cr}C_f\left(T_{cr,i,in} - T_{cr,i,out}\right)\right\} + (1-\gamma)U_{loss}A_{cr,i}\left(T_{am} - T_{cr,i}\right) \tag{12}$$

$$T_{cr,i,out} = T_{cr,i} + (T_{cr,i,in} - T_{cr,i})EXP(U_{cp,i}A_{cp,i}/\dot{m}_{f,cr}C_f) \tag{13}$$

where, i=1,…,9 $\cdot\gamma$ is either 1 or 0 depending on whether it is daytime or nighttime.

The energy balance for the reserve tank can be ex pressed as:

$$\frac{d}{dt}\left\{\left(W_{tm}C_{tm} + W_{wt}C_w\right)T_{wt}\right\} = \dot{m}_w C_w\left(T_{bed,out} - T_{wt}\right) + U_{tloss}AS_{rt}\left(T_{am} - T_{wt}\right) \tag{14}$$

where, $T_{cr,i,out} = T_{cr,i+1,in}$; $T_{cr,9,out} = T_{bed,in}$
and $T_{bed,out} = T_{wt,in}$.

The bed, evaporator and condenser energy balances and concentration in beds are calculated according to Saha et al. [29].

Table 4. Coefficients A_i and B_i

Coefficient (i)	0	1	2	3
A_i	-6.5314	0.72452×10^{-1}	-0.23951×10^{-3}	0.25493×10^{-6}
B_i	-15.587	0.15915	-0.50612×10^{-3}	0.5329×10^{-1}

The collector efficiency equation is:

$$\eta = 0.64 - 0.89\left(\frac{T_f - T_{am}}{I}\right) - 0.001\left(\frac{T_f - T_{am}}{I}\right)^2 \tag{15}$$

And solar radiation equation is considered to be same as Alam et al. [20]. The cyclic average cooling capacity (CACC) is calculated by the equation:

$$CACC = \dot{m}_{chill} C_{chill,f}\left(\int_{beginofcycletime}^{endofcycletime} \left(T_{chill,in} - T_{chill,out}\right)dt\right) \Big/ t_{cycle} \tag{16}$$

The cycle COP (coefficient of performance) and net solar COP ($COP_{solar,net}$) are calculated respectively by the equations:

$$COP_{cycle} = \frac{\int_{beginofcycletime}^{endofcycletime} \dot{m}_{chill} C_{chill,f}\left(T_{chill,in} - T_{chill,out}\right)dt}{\int_{beginofcycletime}^{endofcycletime} \dot{m}_f C_f \left(T_{d,in} - T_{d,out}\right)dt} \tag{17}$$

$$COP_{solar,net} = \frac{\int_{Sunrisetime}^{chillerstoptime} \dot{m}_{chill} C_{chill} (T_{chill,in} - T_{chill,out})dt}{\int_{Sunrisetime}^{chillerstoptime} n.A_{cr} I\, dt} \tag{18}$$

where, I is the solar irradiance, A_{cr} is each collector area and n is number of collectors. Therefore, the total cooling energy produced in a full day by the chiller can be expressed as:

$$Q_r = \int_{chillerstartingtime}^{chillerstopingtime} \dot{m}_{chill} C_{chill} \left(T_{chill,in} - T_{chill,out}\right)dt. \tag{19}$$

Similarly, total heat energy used in cooling production Q_h and total heat collected by the solar collector Q_{scrh} can be expressed by the equations

$$Q_h = \int_{chillerstartingtime}^{chillerstopingtime} \dot{m}_f C_f \left(T_{d,in} - T_{d,out}\right)dt \tag{20}$$

$$Q_{scrh} = \int_{Sunrisetime}^{Sunsettime} nA_{cr} I dt \tag{21}$$

respectively. The simulation procedure is available elsewhere, in Alam et al [20].

The mean of the average maximum monthly radiation data of Dhaka for seven years (2003-2010) (Latitude $23^\circ 46'$ N, Longitude $90^\circ 23'$ E) has been used. Results are calculated based on solar data of Dhaka on the month of April. During April, in Dhaka, the sun rises at 5.5h and sets at 18.5h, where the maximum temperature is 34°C and minimum temperature is 24°C. The average maximum solar radiation in this month is about 771 W/m^2. A sine function has been considered to simulate solar data. This equation is available in [25]. The tolerance for all the convergence criteria is 10^{-4}.

The ambient temperature is calculated using the following equation:

$$T_{am} = \left(T_{\max} + T_{\min}\right)/2 + \frac{T_{\max} - T_{\min}}{2} * Sin\left(\frac{\pi * \left(Daytime - Sunrisetime - i\right)}{Daylength}\right) \tag{22}$$

where, i equals to the time difference between the maximum radiation and maximum temperature of the day.

4. Result and discussions

A conventional single stage basic adsorption chiller run by silica gel-water pair has been considered. Similar chillers with two adsorption beds have been first discussed by Dauss et al. [30]. An autonomous adsorption chiller run by solar heat, supported by a storage tank had been discussed by Alam et al. [25]. The performance of such a chiller has been compared with a chiller run by direct solar coupling by Rouf et al. [26]. Solar heat supported system performance is dependent on chiller configuration, collector number (also on available climatic conditions) and cycle time. For the climatic condition of Dhaka, 28 collectors (48.16 m^2 collector area) are needed with 800s cycle time to run this chiller with direct solar coupling for the base run conditions.When collector area is increased, the temperature of the heat transfer fluid (water) increases very rapidly and exceeds 100℃ for the present chiller. It causes high pressure on the heat transfer pipe inside the collector as well as on the aluminium pipes used in the chiller. Thus, with 48.16 m^2 collector area and 800s cycle time, the adsorption bed temperature reaches 83℃ while the collector temperature is 90.5℃. With a collector area of 51.6 m^2 (30 collectors) and 1000s cycle time, the collector temperature reaches 93.52℃ which can increase the bed temperature to 92.07℃ at the peak hours at steady state with a storage tank volume of 0.343m^3. If the tank volume is increased to 2.197 m^3(dimension: 1.3 meter) and holds 2187 kg water, it needs 75.68m^2 collector area (44 collectors) to increase collector temperature to 88.9°C. It can raise adsorption bed temperature to 88.55°C. It should be noted that for silica gel-water pair, driving temperature is around 80℃. Hence, if a smaller number of collectors are considered, the collector temperature as well as the bed temperature will decrease. However, it affects the performance of the chiller.

Increase in the cycle time does not have much effect on the cooling capacity but it increases the operating time of the chiller. On the other hand, in the middle of the day, when solar radiation is at its maximum and the collector outlet temperature is at its highest, a smaller cycle time could be a better option. The maximum cooling capacity is depicted as being 11.5 kW for 61.92 m^2 collector area (36 collectors) with a storage tank of volume $1m^3$. Whereas, for direct solar coupling, maximum cooling capacity is 11.1 kW with the optimum collector area and cycle time discussed earlier. For an adsorption chiller assisted by heat storage, a huge amount of water in the storage tank needs to be heated. Therefore, a large collector area is needed. The collector area can be reduced if a smaller tank is utilized.

As the system is at the steady state on the third day, for both cases, a comparative CACC is depicted in Fig. 2. At the end of the day, as the tank water is still hot and this hot water is supplied to the desorber, it is working as the basic cycle after sunset. The system continues working until the temperature difference between the heat source (tank water) and heat sink (ambient temperature) is 25℃. At this time, the heat supply from tank water, being used up gradually looses temperature. Hence, at this time, a longer cycle time is preferable. The chiller can function for a longer time after sunset if a larger tank is considered. Thus, when the volume of the tank is increased from 0.343 cubic meters to 1 cubic meter, it needs 6 more collectors (10.32 m^2 additional collector area). But for the next 0.3 meter increase in the dimension, it needs to increase 8 more collectors (13.76 m^2 additional collector area). Hence, a tank volume of 1 cubic meter produces optimum cooling capacity 11.5 kW and it works for almost 12.5 hours, namely from 8.5h in the morning till 21.0h at night. However, with a tank of volume 2.197 cubic meters, cooling capacity is 11 kW and it functions for almost 14 hours, namely from 9.0h in the morning till 23.0h, late at night.

Table 5. Energy distribution for different tank volume

Tank volume m^3	Total cooling Produced (MJ)	Total Heat used (MJ)	Total Heat collected (MJ)
N/A	305.9139152	647.0234632	646.6949381
0.343	330.7191619	658.8672466	679.5850481
1.0	365.0319832	722.1462433	785.7779722
2.197	379.4378875	748.4155547	864.1006393

For a larger storage tank, the system needs more heat to raise the temperature of the tank water. Hence, the solar heat collected through the additional collectors is mostly used up to heat the tank water rather than to be used in the chiller. The total energy collected and the total cooling production in one day for different cases is represented in Table 5.

That is, heat storage with solar heat driven cooling system produces 73.5 MJ more cooling than a chiller with direct solar coupling. The installation of a storage tank enables the system to run for almost 4 more hours after sunset. It also enhances the overall cooling production by approximately 24%. However, it needs to enhance 57% of solar collector area allocated to the change from direct solar coupling to the system with storage tank of volume 2.197 m^3.

The bar diagram (Fig. 3.) demonstrates the total cooling energy produced compared with the total heat energy utilized for all the cases. It is observed that it uses comparatively less amount of heat energy to produce cooling energy in case of the system with storage tank compared with direct solar coupling and tank volume 2.197 m^3. Which indicate energy leakage in case of system with direct solar coupling. On the other hand, in case of system with heat storage, residual heat collected

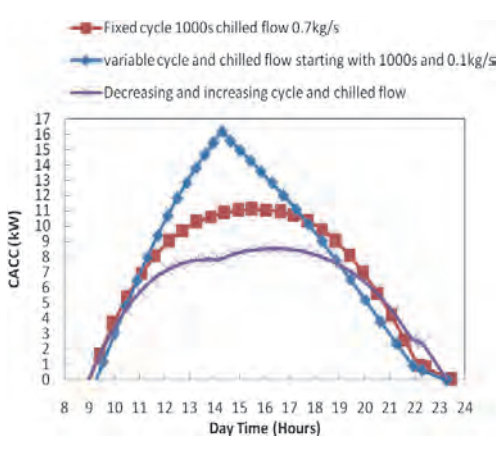


Fig. 4. Comparative CACC for different choice of cycle and Chilled flow for chiller with tank volume 2.197 m^3

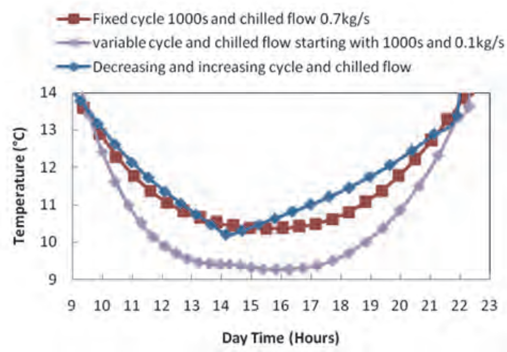


Fig. 5. Comparetive evaporator outlet for chiller with storage tank volume 2.197 m^3

by the collector which is not used in the cooling production, is utilized to heat up the tank water.

An increased collector area and tank volume increases the working hour of the chiller, but the cooling capacity decreases. However, there remains the effect of the operating conditions on the performance of the chiller (Rouf et al. [22]). In the beginning and at the end of the day, heat input is very low, so it is preferable to use a longer cycle time to increase the temperature of the collector as well as the desorber. On the contrary, at midday, radiation is high and collector temperature is already above 80°C. Hence, at this time, a shorter cycle is appropriate. A very high temperature does not help in enhancing the performance, but instead has an impact on the heat transfer fluid (water). Increased chilled flow increases the cooling capacity and the temperature of the evaporator out flow. In the intention to get a steady cooling effect for a longer time, controlled chilled water flow rate provides a better result. In this respect, a variable cycle time and variable chilled water supply to the evaporator throughout the chiller operating hours has been considered. Table 6 shows two different chosen variations of the two parameters respectively.

Figure 4 shows that the cooling capacity increases due to the variation in cycle time and chilled flow rate from 12.0h to 18.5 h. Figure 6 indicates a steady evaporator outlet temperature for choice i) (Fig. 5). The temperature is between 10°C to 9°C from 12.0h to almost 18.0h. Due to these changes in the operating conditions, there are small variations in energy collection and cooling production. The adsorbent, once saturated or exhausted due to desorption, increased heat input or longer cycle time does not help in cooling production. Rather, when this cycle of adsorption-desorption is accelerated, better cooling capacity and lower cooling effect can be acclaimed. The choice of variable cycle time and chilled water flow rate for optimum performance is depicted in Fig. 6. Consequently, the system design should depend on how much cooling production and maximum cooling capacity is required. The increase in heat energy collection and cooling production is represented in Table 7.

A ton of refrigeration is 3.517 kW [31]. Now, let one find out the number of units consumed per hour by a conventional 1.5 Ton Air Conditioner. The indoor unit is constantly on, but the compressor doesn't run the whole time. It only starts when the indoor temperature begins to rise and stops once the required temperature is achieved. So, in an hour, the compressor runs only for about half the time. So, the total no. of units consumed per hour would be [(1kW x 0.5hrs) + (0.2kW x 1 hr)] = 0.7 kWh [32]. Now, one can multiply this factor with the number of hours the AC runs in a month. Suppose it runs for about 6 hours every day. Then the total units consumed would

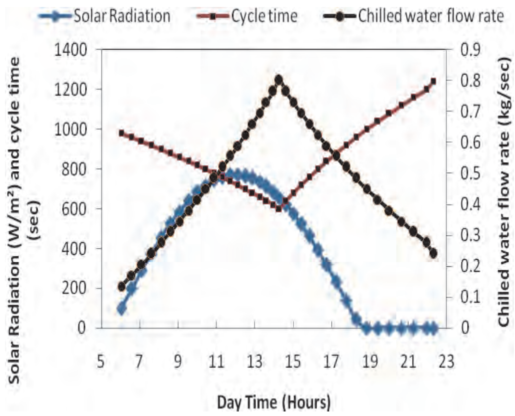


Fig. 6. Choice of variable cycle time and chilled water flow rate for optimum performance with storage tank volume 2.197m^3

Table 6. Choice of operating conditions

Operating parameter	Sun rise hour till 14.0h	14.0h till chiller stop time
Cycle time	i) Starting with 1000s decreasing in every consecutive cycle at the rate of 20s/cycle ii) Starting with 1000s decreasing in every consecutive cycle at the rate of 20s/cycle	i)increasing in every consecutive cycle at the rate of 40s/cycle ii) increasing in every consecutive cycle at the rate of 40s/cycle
Chilled flow rate	i)Starting with 0.1kg/s increasing in every consecutive cycle at the rate of 0.035kg/cycle ii) Starting with 1kg/s decreasing in every consecutive cycle at the rate of 0.035kg/cycle	i)decreasing in every consecutive cycle at the rate of 0.035kg/cycle ii) increasing in every consecutive cycle at the rate of 0.035kg/cycle

Table 7. Energy management tank volume 2.197m^3

Case	Total cooling Produced (MJ)	Total Heat used (MJ)	Total Heat collected (MJ)
Cycle 1000s chilled flow 0.7kg/s	379.437	748.415	864.100
Variable cycle time and chilled flow rate	411.101	756.922	880.969

be 0.7 x 180 = 126 units per month. At current rates, on average 7.4 BDT per unit energy in Bangladesh, one would be paying 126 x 7.4 = 932.4 BDT/month. For the climatic condition of Bangladesh, two months-May and June- are considered to be the full monsoon period. Therefore, if the solar adsorption chiller is considered to be running for 10 months a year, it could roughly save 9324 BDT/year only for 1RT space cooling purpose. Furthermore, it could guarantee a safe environment, and preserve electricity/fossil fuel. Moreover, it contributes in the preservation of food in rural areas.

5. Conclusion

A solar heat driven adsorption cooling system has been investigated for better energy utilization. In this regard, a direct solar coupling, a chiller with heat storage and a smart choice of operating conditions has been compared. In this study a conventional single stage, two bed, basic adsorption chiller has been considered which is driven by silica gel-water pair. Based on the above discussion following conclusions can be drawn;

i) When the chiller is connected with a storage tank of volume 2.197 m^3, the overall cooling production is increased by approximately 24%. However, it needs to enhance 57% of solar collector area compared with direct solar coupling.

ii) The maximum cooling capacity reported for storage tank of volume 1 m^3 with cycle time 1000sec and chilled water flow rate 0.7 kg/sec is 11.5 kW.

iii) The cooling capacity can be increased to a maximum value of 16.1 kW for a storage tank of volume 2.197m^3, for which overall cooling production increases by 8% /75.68 m^2 collector area in one day, when variable cycle time and chilled flow rate is considered based on particular time of the day.

Performance of such a chiller, considered in this paper, can be improved by undertaking a backup heat source. There remain other options to be taken as backup, these comparisons will be discussed in the future.

Nomenclature

A	Area (m^2)
c_p	specific heat (J/kgK)
$c_{w,v}$	specific heat of water vapor (J/kgK)
I	solar radiation (W/m^2)
L	latent heat of vaporization (J/kg)
$\dot{m}$	mass flow rate (kg/s)
q	adsorption capacity (kg/kg_s)
Q	energy
Qst	heat of adsorption (J/kg)
t	time (s)
T	temperature (K)
U	heat transfer coefficient (W/m^2K)
$Uloss$	heat loss coefficient (W/m^2K)
$Utloss$	heat loss coefficient of tank (W/m^2K)
W	Mass(kg)

Subscripts

a	adsorber
am	ambient
bed	adsorbent bed
chill	chilled water
con	condenser
cp	collector pipe
cr	collector
d	desorber
eva	evaporator
f	heat transfer fluid (water)
h	heat
l	liquid
M	metal
r	refrigeration
s	silica gel
s_{crh}	solar heat consumed by collector
t	tube
tm	tank metal
v	vapor
w	water
wt	tank water

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