

Solar Adsorption Cooling: A Case Study on the Climatic Condition of Dhaka

Rifat. A. Rouf

School of Engineering and Computer Science, Independent University, Bangladesh, Dhaka, Bangladesh
Email: rifat_rouf@yahoo.com

K. C. A. Alam¹, Md. Abdul. H. Khan², Tasnuva Ashrafee³ and Mohammed Anwer³

¹ Department of Electronics and Communication Engineering, East-West University, Dhaka, Bangladesh

² Department of Mathematics, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh

³ School of Engineering and Computer Science, Independent University, Bangladesh, Dhaka, Bangladesh
Email: kcaalam@ewubd.edu

Abstract—An analytical investigation has been conducted on the performance of an adsorption chiller driven by solar collector panel for the climatic condition of Dhaka. A set of mathematical model and simulation technique are exploited to investigate the system performances of solar driven basic adsorption chiller with silica gel-water as adsorbent/adsorbate pair. For base run condition, at least 13 collectors (each of area $2.415 m^2$) are essential to achieve the required heat source temperature (around $85^\circ C$) to provide 10 kW cooling capacity. It is seen that the chiller provide 10kw cooling capacity at noon, while, the system provides solar COP around 0.35. It is also observed that the collector size can be reduced by optimizing cycle time and chilled water outlet temperature can be controlled by setting an appropriate chilled water flow rate.

Index Terms—Adsorption, solar heat, renewable energy, air-conditioning, heat source temperature

I. INTRODUCTION

Global warming, change in the seasonal cycle, limitation of primary energy compared to the need of the growing global population, are the prime concern of the twenty first century. Cooling and refrigeration are essentials for modern day's society to provide the human comfort. Most of the technologies at present for providing cooling or refrigeration are vapor compressor technology. However, the vapor compressor refrigeration device is one of the technologies responsible for ozone layer destruction as most of these use HCFCs and HFCs. Moreover, it also increases the dependency on the primary energy resources. As a result, it is imperative to reduce the primary energy consumption and to introduce renewable energy for the sustainable development in the global energy sector.

At present, absorption (liquid vapor absorption) is most promising technology and is commercially available alternative to vapor compressor refrigeration devices. Although, solid vapor absorption technology termed as adsorption technology have an additional advantage over other systems, ability to be driven by heat source of

relatively low temperature heat source studied by Kashiwagi "et al" [1], such that waste heat or solar heat. For the last three decades investigations have been carried out both mathematically and experimentally about different features of adsorption technology.

It is well known that the performance of adsorption cooling / heating system is lower than that of other heat driven heating / cooling systems. Meunier [2] and Saha "et al" [3] studied the advantage and development of adsorption cycle. Many adsorbent/adsorbate pairs have been investigated ([4], [5] and [6]) for adsorption refrigeration/heat pump purposes. A very few of these adsorbent/adsorbate pairs were found to be suitable to utilize low grade waste heat (especially below $100^\circ C$ temperature) as driving source. Silica gel-water pair is well suited to this temperature range. Saha "et al." [7], Boelman "et al." [8] and Chua "et al." ([9] and [10]) studied adsorption refrigeration cycle using the silica gel-water pair which can be driven by at least $80^\circ C$ heat source with $30^\circ C$ cooling source. To utilize even lower temperature waste heat (around $50^\circ C$ temperatures), researchers investigated advanced adsorption refrigeration cycles. Saha "et al." [11] proposed and analyzed a two stage and a three stage [12] adsorption chiller. A parametric study on two stage cycle was carried on by Alam "et al." [13] in the same context. Hamamoto "et al." [14] studied two-stage cycle with different mass ratio, while Alam "et al." [15] and Khan "et al." [16] studied the reheat two stage cycle. Though the multistage regeneration techniques enable the chiller to operate with lower waste heat temperature, however, the performance of these systems are very low.

The major intricacy for commercialization of adsorption chiller is their bigger machine size and lower performance. Motivated by this reason Pons and Poyelle [17] proposed internal mass recovery cycle to achieve relatively high cooling power utilizing lower heat source temperature suitable to run a conventional two bed adsorption refrigeration cycle. Alam "et al." [18]

investigated a four bed mass recovery cycle utilizing the pressure differences among the beds. Afterwards, Wang [19] and Akahira “et al.” [20] showed that mass recovery process in conventional two bed adsorption cycle improves the cooling capacity significantly for the relatively low regenerative temperature.

Using solar radiation, Sakoda and Suzuki [21] studied the simultaneous transport of heat and adsorbate in closed type adsorption cooling system. Li and Wang [22] investigated the effect of collector parameters on the performance of solar driven adsorption refrigeration cycle. Yong and Sumathy [23] applied lumped parameter model for solar driven two bed adsorption refrigeration cycle. Clause “et al.” [24] 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. Later, Zhang “et al.” [25] studied solar powered adsorption cooling system. Where, they investigated the operating characteristics of silica gel-water pair as adsorbent/ adsorbate utilizing lumped parameter model. Recently Alam “et al.” [26] investigated the performances of solar collector driven adsorption cooling system under the climatic condition of Tokyo, Japan.

Since Bangladesh is a tropical country, solar panels can be installed and used for day time cooling for the office buildings, cold storages in the rural areas where we can preserve food and medicine for emergency use. From this context, a two bed adsorption cooling system which is run by solar collector, with silica gel-water pair as adsorbent/ adsorbate, is analyzed mathematically under the climatic condition of Dhaka in the present study. The place is located in the northern hemisphere at 23°46’N (latitude), and 90°23’E (longitude). Investigation is done on the collector size to get optimum performance. Also the performance of the chiller had been studied for different cycle time.

II. PRINCIPLE AND OPERATIONAL PROCESS OF THE SYSTEM

A two- bed conventional adsorption cooling cycle driven by solar heat has been considered. Silica gel-water pair as adsorbent/ adsorbate has been chosen for the present study. There are four thermodynamic steps in the cycle, namely, (i) Pre-cooling (ii) Adsorption/Evaporation (iii) Pre-heating and (iv) Desorption/ Condensation process. No heat recovery or mass recovery process is considered in the present study. The adsorber(A1/A2) are alternately connected to the solar collector to heat up the bed during preheating and desorption/ condensation process and to the cooling tower to cool down the bed during pre-cooling and adsorption/ evaporation process. The heat transfer fluid transport heat from the solar collector to the desorber and returns the collector to regain heat from the collector. The valve between adsorber and evaporator and the valve between desorber and condenser are closed during pre-cooling/ pre-heating period while, these are open during adsorption/ evaporation and desorption/ condensation

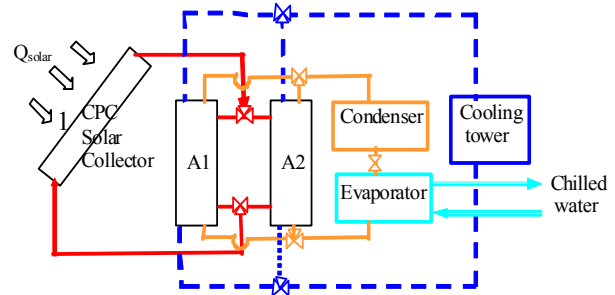


Figure1. Schematic of the solar driven adsorption space cooling system.

process. The schematic of the adsorption cooling with solar collector panel is presented in Fig. 1. The characteristics of adsorbent/adsorbate (silica gel-water) are utilized to produce useful cooling effect run by solar powered adsorption chiller. The chilled water delivered from the evaporator cools the floor of the house.

The operational process of two bed basic adsorption cooling unit can be found in the literature Saha “et al.” [7]. A detailed description of solar collector driven adsorption cooling system is available in Alam “et al.” [26].

A. Mathematical Model

During the adsorption step, the adsorber cooling temperature is taken equal to the ambient temperature. 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 is represented by

$$\frac{d}{dt} \{ (W_M C_{pM} + W_{si} C_{si} + W_{si} q_{bed} C_{si,w}) T_{bed} \} = \Delta H \cdot W_{si} \frac{dq_{bed}}{dt} + \delta \cdot W_{si} C_{si,v} \frac{dq_{bed}}{dt} (T_{eva} - T_{bed}) + \dot{m}_f C_f (T_{bed,in} - T_{bed,out}), \tag{1}$$

$$T_{bed,out} = T_{bed} + (T_{bed,in} - T_{bed}) \cdot \exp(-UA_{bed} / \dot{m}_f C_f). \tag{2}$$

Where, δ equals to zero or one depending whether adsorbent bed is working as desorber or adsorber.

The energy balance equation for the condenser is represented by

$$\frac{d}{dt} \{ (W_{cd,M} C_{cd,M} + W_{cd,r} C_r) T_{cd} \} = - L \cdot W_{si} \frac{dq_d}{dt} + W_{si} C_{r,v} \frac{dq_d}{dt} (T_{cd} - T_{bed}) + \dot{m}_{f,cd} C_f (T_{cd,in} - T_{cd,out}), \tag{3}$$

$$T_{cd,out} = T_{cd} + (T_{cd,in} - T_{cd}) \cdot \exp(-UA_{cd} / \dot{m}_{f,cd} C_f). \tag{4}$$

The energy balance equation for the evaporator is

$$\frac{d}{dt} \{ (W_{e,M} C_{e,M} + W_{e,r} C_{ml}) T_e \} = - L \cdot W_{si} \frac{dq_a}{dt} + W_{si} C_{r,l} \frac{dq_d}{dt} (T_e - T_{cd})$$

$$+ \dot{m}_{f, chill} C_f (T_{chill, in} - T_{chill, out}) \quad (5)$$

$$T_{chill, out} = T_e + (T_{chill, in} - T_e) \cdot \exp(-UA_e / \dot{m}_{f, chill} C_f) \quad (6)$$

The mass balance of the refrigerant inside the evaporator is expressed as

$$\frac{dW_{e,r}}{dt} = -W_{si} \left(\frac{dq_a}{dt} + \frac{dq_d}{dt} \right) \quad (7)$$

The concentration in bed is

$$\frac{dq}{dt} = kasp(q^* - q), \quad (8)$$

where,

$$kasp = Ds \cdot \exp(-Ea / (Rgas \cdot T)),$$

$$Ds = 15 \cdot D_{s0} / (Rp^2),$$

$$q^* = AA \cdot (Ps(Tv) / Ps(Tb))^{BB},$$

$$AA = A_0 + A_1T + A_2T^2 + A_3T^3,$$

$$BB = B_0 + B_1T + B_2T^2 + B_3T^3.$$

The saturation pressure is calculated according to the Antoine's equation, as Saha "et al." [7], where the values of A_i 's and B_i 's will also be found. The energy balance of each of the collector is calculated by the manufacturers' data.

The collector efficiency equation is considered to be same as Clauss "et al." [24]. The cyclic average cooling capacity (CACC) is calculated by the equation

$$CACC = \dot{m}_{chill} C_{chill, f} \left(\int_{beginofcycletime}^{endofcycletime} (T_{chill, in} - T_{chill, out}) dt \right) / t_{cycle} \quad (9)$$

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

$$COP_{cycle} = \frac{\int_{beginofcycletime}^{endofcycletime} \dot{m}_{chill} C_{chill, f} (T_{chill, in} - T_{chill, out}) dt}{\int_{beginofcycletime}^{endofcycletime} \dot{m}_f C_f (T_{d, in} - T_{d, out}) dt} \quad (10)$$

$$COP_{sc} = \frac{\int_{beginofcycletime}^{endofcycletime} \dot{m}_{chill} C_{chill} (T_{chill, in} - T_{chill, out}) dt}{\int_{beginofcycletime}^{endofcycletime} n \cdot A_{cr} I dt} \quad (11)$$

B. Simulation Procedure

Standard solar radiation data measured at horizontal tilt have been supported by the Renewable Energy Research Center (RERC), University of Dhaka (Latitude 23.46 N, Longitude 90.23 E). For the present study the solar radiation data for the station of Dhaka has been used. The monthly maximum and minimum average temperature (°C) at Dhaka station is supported by Bangladesh Meteorology Department (BMD).

Results are generated based on solar data of Dhaka in April, chiller configuration are same as Saha "et al." [7] and collector data are same as Alam "et al." [26]. In April the sunrise time is at 5.5h and sun set at 18.5h, where as maximum and minimum temperatures on that month are 34 °C and 24 °C respectively. The maximum solar radiation in April is measured as 988 W/m². For simulation the maximum measured radiation of the month has been considered and sine function is used (as Alam "et al." [26]).

Implicit finite difference approximation method is applied to solve the set of differential equations. The water vapor concentration q in a bed is a nonlinear function of pressure and temperature. It is almost unfeasible to divide the concentration in terms of temperature for the present time and previous time. Hence, to begin with, the temperature for present step is taken based on assumption. The pressure and concentration are then calculated for the present step based on this assumption of temperature. Later, gradually the consequent steps are calculated based on the primary concentration with the help of the finite difference approximation. During this process, the newly calculated temperature is checked with the assumed temperature. If the difference is not less than convergence criteria, then a new assumption is made. Once the convergence criteria are fulfilled, the process goes for the next time step. The tolerance for all the convergence criteria is 10⁻⁴. The initial temperature, pressure and concentration are set based on the temperature of the beginning of the day. Then the program runs for consecutive several days unless the steady conditions arrive. That is, all the conditions at the end of the day are similar to that of the beginning of the previous day. In this paper all results are presented for the 3rd day, since the system reached to its steady state condition from day 3 i.e. all output appeared to be identical for the consecutive days. The design and the operating conditions used in the simulation are given in Table I (attached in appendix).

Logical programming language FORTRAN with Compaq visual Fortran compiler has been exploited to obtain the numerical solution of the proposed model.

III. RESULT AND DISCUSSION

The comparison between measured and simulated radiation data is presented in Fig. 2. The model for the radiation shows good agreement with measured data for April.

The temperature histories of collector outlet and bed, for cycle time 800s and 1000s, are presented in Fig. 3 (a), (b) and (c). The driving temperature level for the silica gel-water pair is around 80°C (Saha "et al." [7]).

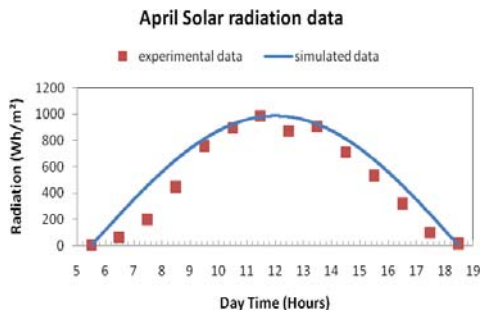


Figure 2. Measured and simulated radiation data

To achieve the required temperature level with cycle time 800s, 16 collectors and with 1000s, 14 collectors are used. The collector outlet temperature reaches 90°C while the bed temperature is 85°C. However, it is also observed that the same temperature level is achievable when collector number is reduced to 13 with cycle time 1000s (Fig. 3 (c)).

However, increase in the cycle time for all the cases increases the bed temperature which is not preferable for the silica gel-water adsorption bed. On the other hand decreasing in the cycle time causes decrease in driving

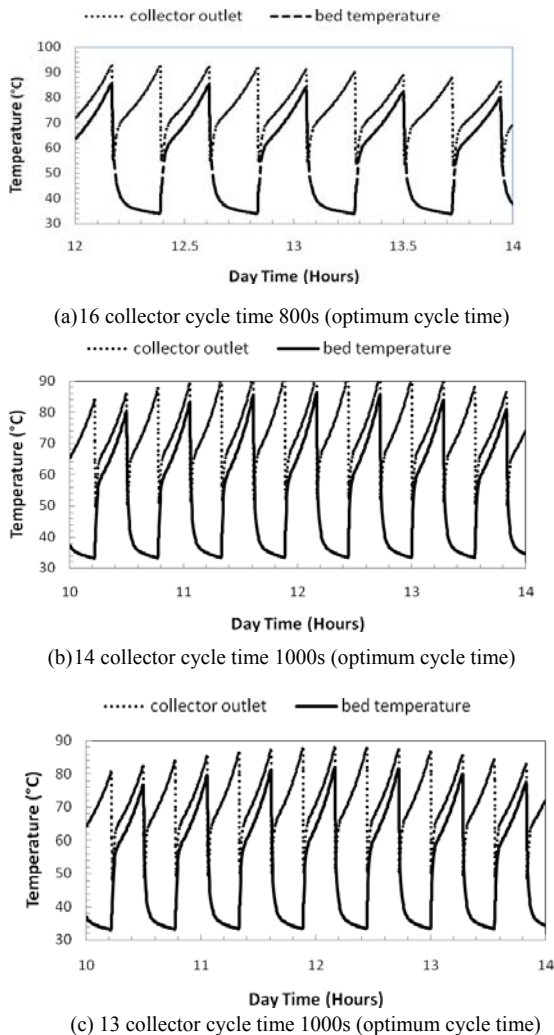


Figure 3. Temperature profiles for collector and beds

source temperature.

Clearly there remain an optimum cycle time for maximum cooling capacity (Saha “et al.” [7] and Chua “et al.” [9]), the optimum cycle time for 16 collectors is considered as 800s as driving heat source temperature is achievable with this cycle time. In the same way, the same optimum cycle times for both collector numbers 14 and 13 are considered as 1000s. It can be mentioned here that the collector size can be reduced by taking longer cycle time for solar heat driven adsorption cooling system (Alam “et al” [26]). However, it may reduce cooling capacity for taking excessive long cycle time. Therefore, it is essential to take both driving source temperature as well as cooling capacity into consideration to select optimum cycle time. Clearly Fig. 4 (c) shows that for optimum cycle time with 13 collectors the cooling capacity is 10 kW at the middle of the day with the base run condition.

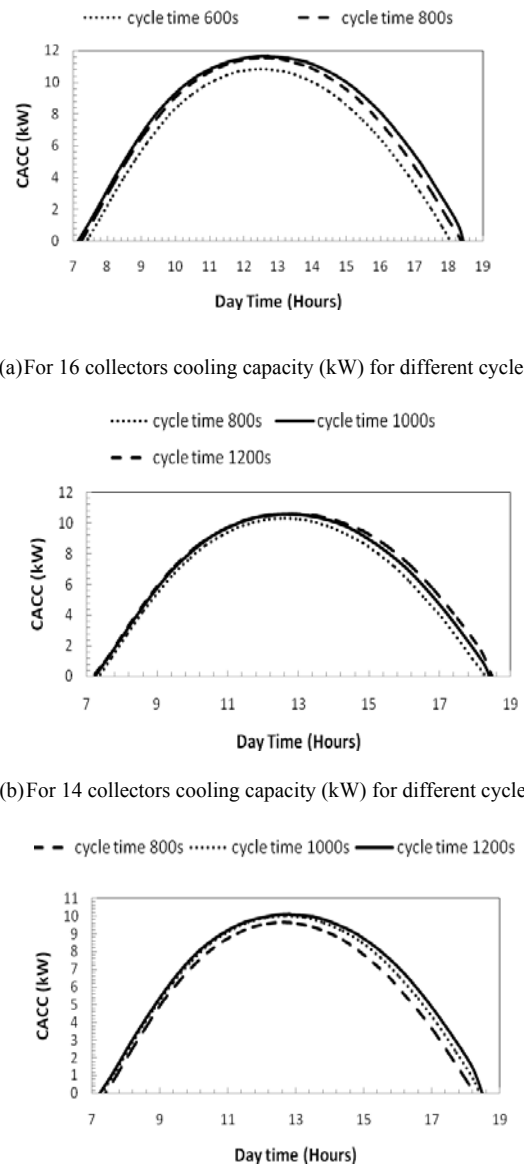


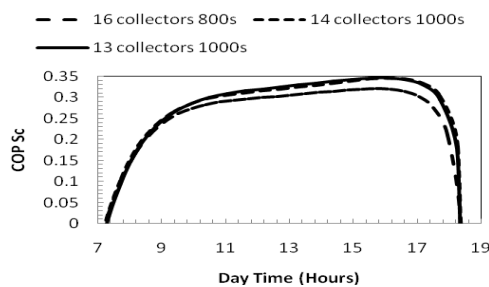
Figure 4. Cooling capacity (kW) for different number of collectors

Conversely increase in the cycle time increases the COP values of the system (Saha et al [7] and Chua “et al” [9]). The maximum solar COP 0.35 and cycle COP 0.6 is achievable by the proposed system when 1000s cycle time is considered with 13 collectors (Fig. 5 (a) and (b)).

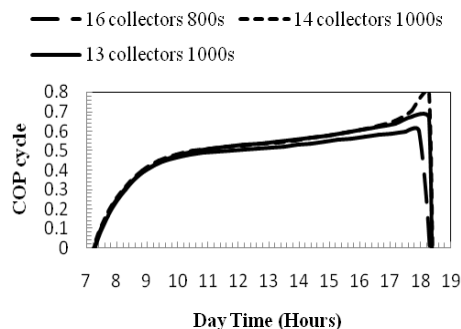
The increase of COP at afternoon happens due to the inertia of collector materials (Fig. 5 (a)). At afternoon, there is less heat input but there is relative higher cooling production due to the inertia of materials of collector. Therefore, there is slow increase of COP. However, it starts declining suddenly when the radiation is too low to heat up the heat transfer fluid. A sudden rise of cycle COP is observed at late afternoon. This happens due to the excessive long cycle time comparing with low radiation at afternoon. Due to the long cycle time at afternoon, there were some cooling production at the beginning of that cycle but there is a very less heat input in whole cycle time. If one takes variation in cycle time for the different cycle in the whole day then this behavior will not be observed for solar COP. Almost same observation was found as for the cycle COP.

The chilled water outlet temperature histories have been depicted in Fig. 6 for collector numbers 16, 14 and 13 with their respective optimum cycle time. The temperature is shown during the pick hours of the day time from 12.0h to 14.0h. The fluctuation of the chilled water outlet temperature is 4°C. Chilled water outlet temperature varies from 8°C to 12°C, if 13 collectors are used. However chilled water outlet temperature decreases below 8°C when collector number increases.

In air conditioning system, CACC and COP are not the only measurement of performances. If those values



(a) Solar COP in a cycle for different number of collectors



(b) Cycle COP for different number of collectors

Figure 5. Solar COP in a cycle and cycle COP for different number of collectors

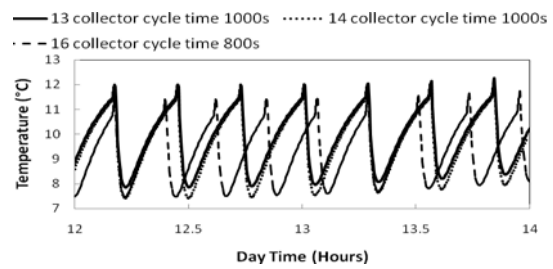


Figure 6. Chilled water outlet temperature histories

are higher but there is relatively higher temperature chilled water outlet, then the system may not provide comfortable temperature to the end user. For all the above studies the volumetric flow rate of chilled water is considered to be the same, which is 0.7kg/s. However, the chilled water outlet temperature can be controlled by adjusting the flow rate of chilled water.

Chilled water outlet temperature histories of 13 collectors with different volumetric flow rate have been depicted in Fig. 7. Decreasing the volumetric flow of chilled water to the evaporator results in decreasing the temperature of chilled water outlet. The chilled water outlet temperature goes down to 4.5°C when volumetric flow is considered as 0.3kg/s and temperature varies from 4.5°C to 9.5°C. Therefore, it may be concluded that chilled water outlet can be controlled by adjusting chilled water flow rates.

On the other hand when volumetric flow rate increases cooling capacity increases compared to the heat input. These effects are depicted in Fig. 8 (a). The overall cycle COP decreases when the volumetric flow decreases. But at afternoon there is a sudden rise in the cycle COP which is visible in Fig 8 (b). This effect is due to the lower cooling capacity and comparative higher heat input due to the inertia of the collector material.

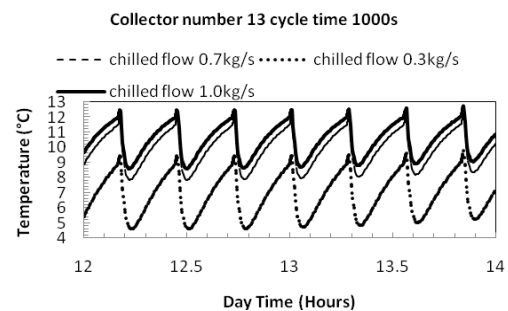
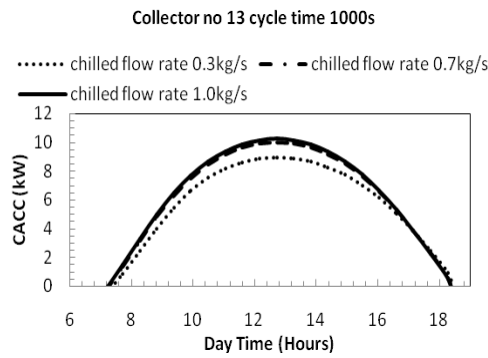
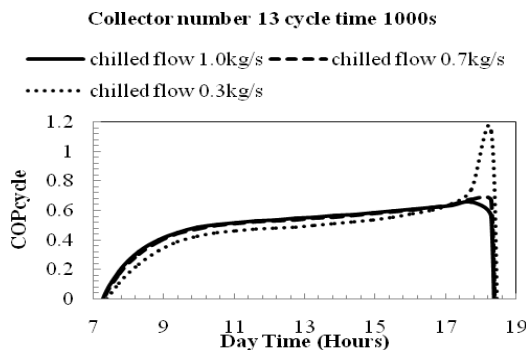


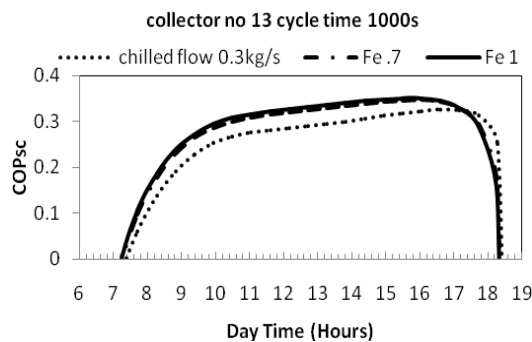
Figure 7. Chilled water outlet temperature for 13 collectors different chilled water flow rates



(a) CACC with different chilled water flow rates



(b) COP cycle with different chilled water flow rates



(c) COP sc with different chilled water flow rates

Figure 8. Performance of the chiller with 13 collectors cycle time 1000s with different chilled water flow rate

IV. CONCLUSION

A solar heat driven adsorption cooling system has been analyzed mathematically based on the climatic conditions of Dhaka, Bangladesh. A panel of CPC (concentrated parabolic collector) has been used for the present analysis. Each collector area approximates 2.415m². Based on the analysis the following conclusions can be drawn.

- For the climatic condition of Dhaka 1000s cycle time with 13 collectors is needed to raise the driving heat source temperature to 85°C for the base run condition.

- Cyclic average cooling capacity is around 10 kW at 12:00 noon for base run condition.
- Increase in the cycle time increases the COP of the present system. The maximum cycle COP is 0.6 at 12:00 noon with base run condition.
- Maximum solar COP is around 0.35.
- The performance of the system increases with the increase of the volumetric flow rate of chilled water.
- But the cooling effect to the end user can be improved by adjusting the chilled water flow rate.
- The collector area can be reduced by proper selection of cycle time.

APPENDIX A TABLE I

Design and the operating conditions used in the simulation

Symbol	Description	Value
A_{bed}	Adsorbent bed heat transfer area	2.415m ²
U_{bed}	Heat transfer coefficient of bed	1724.14W / m ² K
W_{tm}	Heat exchanger tube weight (Cu)	51.2kg
W_{fm}	Heat exchanger fin weight (Al)	64.04kg
A_{eva}	Evaporator heat transfer area	1.91m ²
U_{eva}	Evaporator heat transfer coefficient	2557.54W / m ² K
$W_{eva,M}$	Evaporator heat exchanger tube weight (Cu)	12.45kg
A_{con}	Condenser heat transfer area	3.73m ²
U_{con}	Condenser heat transfer coefficient	4115.23W / m ² K
$W_{con,M}$	Condenser heat exchanger tube weight (Cu)	24.28kg
A_{cr}	Each collector area	2.415m ²
W_{cp}	Weight of each pipe including absorber sheet	0.4913kg
N_p	Number of pipe in each collector	9
$\dot{m}_{f,hot}$	Total mass flow rate to CPC panel or to desorber	1.3kg / s
$\dot{m}_{f,cool}$	Cooling water flow rate to adsorber	1.3kg / s
W_{si}	Weight of silica gel in each bed	47kg
$W_{eva,r}$	Liquid refrigerant inside evaporator initially	50kg
$\dot{m}_{f,cond}$	Cold water flow rate to condenser	1.3kg / s
$\dot{m}_{f,chill}$	Chilled water flow rate	0.7kg / s
$W_{con,r}$	Condenser refrigerant inside condenser	0.0kg
ΔH	Heat of adsorption (silica gel bed)	2.81E + 06J / kg
R_{gas}	Water gas constant	4.62E + 02J / kg.K
Ea	Activation energy	2.33E + 06J / kg
D_{s0}	Diffusion coefficient	2.54E - 04m ² / s
R_p	Particle diameter (Silica gel)	0.35E - 03m
T_{cool}	Cooling source temperature	30° C
$T_{chill,in}$	Chilled water inlet temperature	14° C
$C_{r,l}$	Specific heat of water (liquid phase)	4.18E + 03J / kg.K
$C_{r,v}$	Specific heat of water (vapor phase)	1.89E + 03J / kg.K
C_{cu}	Specific heat of copper (Cu)	386J / kg.K
C_{al}	Specific heat of aluminum (Al)	905J / kg.K
C_{si}	Specific heat of silica gel (Si)	924J / kg.K
L	Latent heat of vaporization (water)	2.6E + 06J / kg

ACKNOWLEDGEMENT

The authors wish to thank Renewable energy research centre, University of Dhaka and Bangladesh Meteorological Department for their co-operation.

REFERENCES

- [1] T. Kashiwagi, A. Akisawa, S. Yoshida, K. C. A. Alam, Y. Hamamoto, "Heat driven sorption refrigerating and air conditioning cycle in Japan," *Proceedings of International Sorption Heat Pump Conference*, Shanghai, China, pp. 50-62, September 2002.
- [2] F. Meunier, "Solid sorption heat powered cycles for cooling and heat pump applications," *Applied Thermal Engineering*, vol. 18, pp. 714-729, 1998.
- [3] B. B. Saha, T. Kashiwagi, "Experimental investigation of an advanced adsorption refrigeration cycle," *ASHRAE Transactions*, vol. 103 (2), pp. 50-58, 1997.
- [4] D. I. Tchernev, D. T. Emerson, "High efficiency regenerative zeolite heat pump," *ASHRAE Transactions*, vol. 94 (2), pp. 2024-2032, 1988.
- [5] R.E. Critoph, R. Vogel, "Possible adsorption pairs for use in solar cooling," *International Journal of Ambient Energy*, vol. 7 (4), pp. 183-90, 1986.
- [6] R. E. Critoph, "Activated carbon adsorption cycles for refrigeration and heat pumping," *Carbon*, vol. 27 (1), pp. 63-70, 1989.
- [7] B. B. Saha, E. C. Boelman, T. Kashiwagi, "Computer simulation of a silica gel-water adsorption refrigeration cycle – the influence of operating conditions on cooling output and COP," *ASHRAE Transactions*, vol. 101 (2), pp. 348-357, 1995.
- [8] E. C. Boelman, B. B. Saha, T. Kashiwagi, "Experimental investigation of a silica gel-water adsorption refrigeration cycle –the influence of operating conditions on cooling output and COP," *ASHRAE Transaction*, vol. 101 (2), pp. 358-366, 1995.
- [9] H. T. Chua, K. C. Ng, A. Malek, T. Kashiwagi, A. Akisawa, B. B. Saha, "Modeling the performance of two bed, silica gel-water adsorption chillers," *International Journal of Refrigeration*, vol. 22 (3), pp. 194-204, 1999.
- [10] H. T. Chua, K. C. Ng, A. Malek, T. Kashiwagi, A. Akisawa, B. B. Saha, "Entropy generation analysis of two-bed, silica gel-water, non-regenerative adsorption chillers," *Journal of Physics D –Applied Physics*, vol. 31 pp. 1471-1477, 1998.
- [11] B. B. Saha, E. C. Boelman, T. Kashiwagi, "Computational analysis of an advanced adsorption refrigeration cycle," *Energy*, vol. 20 (10), pp. 983-994, 1995.
- [12] B. B. Saha, A. Akisawa, T. Kashiwagi, "Solar/waste heat driven two-stage adsorption chiller: the prototype," *Renewable Energy*, vol. 23 (4), pp. 93-101, 2001.
- [13] K. C. A. Alam, B. B. Saha, A. Akisawa and T. Kashiwagi, "Influence of design and operating conditions on the system performances of a two-stage adsorption chiller," *Chemical Engineering Communications*, vol. 191 (7), pp. 981-997, 2004.
- [14] Y. Hamamoto, K. C. A. Alam, A. Akisawa, T. Kashiwagi, "Performance evaluation of a two-stage adsorption refrigeration cycle with different mass ratio," *International Journal of Refrigeration*, vol. 28 (3), pp. 344-352, 2005.
- [15] K. C. A. Alam, M. Z. I. Khan, A. S. Uyun, Y. Hamamoto, A. Akisawa, T. Kashiwagi, "Experimental study of a low temperature heat driven re-heat two-stage adsorption chiller," *Applied Thermal Engineering*, vol. 27 (10), pp. 1686-1692, 2007.
- [16] M. Z. I. Khan, K. C. A. Alam, B. B. Saha, A. Akisawa, T. Kashiwagi, "Performance Evaluation Of Multi-Stage, "Multi-bed adsorption chiller employing re-heat scheme," *Renewable Energy*, vol. 33 (1), pp. 88-98, 2008.
- [17] M. Pons, F. Poyelle, "Adsorptive machines with advanced cycles for heat pumping or cooling applications," *International Journal of Refrigeration*, vol. 22 (1), pp. 27-37, 1999.
- [18] K. C. A. Alam, A. Akahira, Y. Hamamoto, A. Akisawa and T. Kashiwagi, "A four-bed mass recovery adsorption refrigeration cycle driven by low temperature waste/renewable heat source," *Renewable Energy*, vol. 29 (9), pp. 1461-1475, 2004.
- [19] R. Z. Wang, "Performance improvement of adsorption cooling by heat and mass recovery operation," *International Journal of Refrigeration*, vol. 24 (7), pp. 602-11, 2001.
- [20] A. Akahira, K. C. A. Alam, Y. Hamamoto, A. Akisawa and T. Kashiwagi, "Experimental investigation of mass recovery adsorption refrigeration cycle," *International journal of refrigeration*, vol. 28 (4), pp. 565-572, 2005.
- [21] A. Sakoda, M. Suzuki, "Simultaneous transport of heat and adsorbate in closed type adsorption cooling system utilizing solar heat," *Journal of Solar Energy Engineering*, vol. 108, pp. 239-245, 1986.
- [22] M. Li, R.Z. Wang, "A study of the effects of collector and environment parameters on the performance of a solar powered solid adsorption refrigerator," *Renewable Energy*, vol. 27 (3), pp. 369-382, 2002.
- [23] L. Yong, K. Sumathy, "Modeling and simulation of a solar powered two bed adsorption air conditioning system," *Energy Conversion and Management*, vol. 45 (17), pp. 2761-2775, 2004.
- [24] M. Clausse, K. C. A. Alam, F. Meunier, "Residential air conditioning and heating by means of enhanced solar collectors coupled to an adsorption system," *Solar Energy*, vol. 82 (10), pp. 885-892, 2008.
- [25] G. Zhang, D. C. Wang, Y. P. Han, W. Sun, "Simulation of operating characteristics of the silica gel-water adsorption chiller powered by solar energy," *Solar Energy*, vol. 85 (7), pp. 1469-1478, 2011.
- [26] K. C. A. Alam, B. B. Saha and A. Akisawa, "Adsorption cooling driven by solar collector: a case study for Tokyo solar data," *Applied Thermal Engineering*, 2011, "in press".



Rifat Ara Rouf born in 1970 is currently doing her Ph.D. in the field of solar heat driven cooling and heating system in Bangladesh University of Engineering and Technology (BUET), Dhaka, Bangladesh. She had completed her Master of Philosophy in Applied Mathematics from the same institute in 2007. Earlier she had completed her masters in Applied Mathematics from University of Dhaka, Bangladesh in 1996. Currently, she is working as a SENIOR LECTURER in Independent University, Bangladesh (IUB since January 2008.



Dr. K. C. Amanul Alam had completed his Dr of Engineering in Mechanical Engineering from Tokyo University of Agriculture and Technology, Japan. He had his Master of Philosophy and Master of Science in Applied Mathematics from University of Dhaka, Bangladesh. He was also post doctoral fellow at CNAM, Paris, France and JSPS post doctoral

fellow at Tokyo University of Agriculture and Technology, Japan. He started his academic career as a lecturer of Mathematics at Bangladesh University of Engineering and Technology. Currently, he is working as an Associate Professor at East West University, Bangladesh. He has published many articles in peer review international journals and attended many international conferences. His research interest includes mathematical investigation in thermal engineering specially heat driven cooling system, Numerical approach in engineering problems, Fluid mechanics, Energy systems.



Dr. Md. Abdul Hakim Khan had completed his Ph. D. from University of Bristol (U.K.) in 2001. He had his Master of philosophy completed from Bangladesh University of Engineering and Technology, Bangladesh in 1998. Earlier he had completed his Masters in Applied Mathematics from the University of Dhaka, Bangladesh.

Currently, He is Professor of Mathematics at Bangladesh university of Engineering and Technology. Dr. Khan's interest is focused in the area of Singularity analysis on fluid motion by summing power series applying Semi numerical methods: Algebraic and Differential Approximants. His interest is also in

Extension, Analysis and improvement of some fluid dynamical problem, such as, Kelvin-Helmholtz instability, Flow in a Curved pipe etc.



Tasnuva Ashrafee had completed her Master of Science (M.S.) in 2005, Department of Applied Physics, Electronics and Communication Engineering, University of Dhaka, Bangladesh. She has completed her Bachelor of Science (B.Sc.) in 2004 from the same institution. She published an article in the journal of Advanced

Materials Research. She is also a life member of Bangladesh Association of Women Scientists.



Dr. Mohammed Anwer completed his Ph. D. in Mechanical Engineering from Arizona State University (USA) in 1989. He also had his Master of Science from the same university. His Bachelor degree was completed from Bangladesh University of Engineering and Technology, Bangladesh in 1982. Currently he is serving simultaneously as

the Dean and Professor of the school of engineering and computer science at Independent University, Bangladesh (IUB). He has published many articles in peer review international journals and attended many international conferences. His research interest includes Turbulence, Flow modeling, Aerodynamics, Hydrodynamic stability, Experimental fluid mechanics and Computational fluid mechanics.