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# PROSPECT OF SOLAR COOLING BASED ON THE CLIMATIC CONDITION OF DHAKA

**R. A. Rouf[1], K. C. A. Alam[2], M. A. H. Khan[3] and T. Ashrafee[1]**

[1]School of Engineering and Computer Science, Independent University, Bangladesh, Dhaka, Bangladesh,
[2]Department of Electronics and Communication Engineering, East-West University, Dhaka, Bangladesh,
[3]Department of Mathematics, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh.

## ABSTRACT

An analytic investigation on the prospect of solar cooling for the climatic condition of Dhaka is presented. The simulation is done based on the area of the solar radiation collector and cycle time of the adsorption cooling unit. It is found that 14 CPC collectors each of area 2.42m² is required to run the system with base run conditions. It is seen that the optimum cycle time need to be adjusted according to the seasonal change for best possible performance. It is seen that maximum 11 kW (around 3RT) cooling capacity is achievable during hot and humid seasons with base run condition. Based on this motive mathematical analysis is revealed for a number of months during the hot season of Dhaka station (Latitude $23^{\circ}46'$ N, Longitude $90^{\circ}23'$ E). It may be concluded that better performance is possible by decreasing the number of collectors and adjusting optimum cycle time.

**Keywords:** Solar Cooling, Adsorption Cooling, Renewable Energy.

## 1. INTRODUCTION

The increased use of the vapor compressor driven refrigeration devices made us more dependent on the primary energy resources. As the primary energy once used up cannot be used in the same form again, therefore, it is necessary to reduce the consumption of these resources and introduce renewable energy for the sustainable development in the global energy sector. Furthermore, in the late 1980s, chlorofluorocarbons were found to be contributing to the destruction of earth's protective ozone layer. Therefore, the production of these chemicals was phased out and the search for a replacement began. Thermally driven, sorption technology is one of the possible alternatives. At present, absorption (liquid vapor) cycle is most promising technology. Nevertheless, adsorption (solid vapor) cycle have a distinct advantage over other systems in their ability to be driven by heat of relatively low, near-environmental temperatures, so that the heat source, such that waste heat or solar heat, below 100°C can be recovered. Kashiwagi et al. [1], in determination of conservation of heat energy, carried out investigation on heat driven sorption and refrigeration system.

For the last three decades investigations have been carried out both mathematically and experimentally about different features of this system. It is well known that the performance of adsorption cooling / heating system is lower than that of other heat driven cooling/ heating systems. Different choices of adsorbate/adsorbent pairs have been investigated to study about the optimum driving heat source. Zeolite - water pair studied by Rothmeyer et al. [2], Tchernev and Emerson [3] and Guilleminot and Meunier [4]. In these studies the driving heat source was reported as 200°C. In the study of Critoph [5] a lower heat source temperature was observed, i.e. over 150°C with activated carbon – ammonia pair. The use of driving heat source with temperatures of less than 100°C was reported in the study of basic adsorption cycle with silica gel – water pair investigated by Saha et al. [6] and Chua et al. [7]. Later, Saha et al. [8,9] and Alam et al.[10] showed that even less than 70°C heat source can be utilized by employing the advanced multi-stage cycle.

For the effective utilization of low temperature solar thermal energy, Sakoda and Suzuki [11] studied the simultaneous transport of heat and adsorbate in closed type adsorption cooling system. Li and Wang [12] investigated the effect of collector parameters on the performance of solar driven adsorption refrigeration cycle. Later, Yong and Sumathy [13] applied lumped parameter model for two bed adsorption refrigeration cycle with direct coupling of solar collector.

Clausse et al. [14] considered the models of whole units of a residential air conditioning sytem to investigate the performances of the system for the climatic condition of Orly, France. Recently, Alam et al. [15] studied the silica-gel water adsorption cooling cycle with direct coupling of solar collector under the climatic condition of Tokyo.

The article investigates a similar approach for the climatic condition of Dhaka, located in the northern hemisphere at $23^{\circ}46'$ N (latitude), and $90^{\circ}23'$ E (longitude). In the present study the performance of a two bed adsorption cooling system which is run by solar collector, with silica gel-water pair as adsorbent/ adsorbate, is analyzed mathematically for several months of hot seasons, namely summer and autumn.

## 2. 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 is well examined for air-conditioning process driven by low temperature(less than 100 ° C) heat source. There are four thermodynamics 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) alternatively connected to the soar 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 from the solar collector goes to the desorber and returns the collector to gain 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 process. The schematic of the adsorption cooling with solar collector panel is presented in Figure 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.

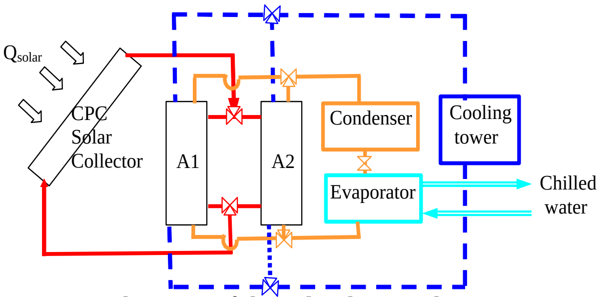



Fig 1. Schematic of the solar driven adsorption space cooling system.

### 2.1 Mathematical Model

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

$$\frac{d}{dt}\quad W_M C_{pM} + W_s C_s + W_s q C_{sw} \quad T_{bed} \quad = Qst \quad W_s \frac{dq}{dt}$$
$$+\,\delta \quad W_s C_{sv} \frac{dq}{dt}\left(T_{eva} \quad T_{bed}\right) + \dot{m}_f C_f \quad T_{bed,in} \quad T_{bed,out} \qquad (1)$$

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

where, $\delta$ equals to zero or one depending whether adsorbent bed is working as desorber or adsorber.

The energy balance for the condenser is represented by

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

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

Energy balance for the evaporator is

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

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

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

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

The concentration in bed is

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

where,

$$kasp = D_s.\exp \quad -E_a / \quad R_{gas}.T$$

$$D_s = 15.D_{s0} / \quad R_p^{\,2} \quad ,$$

$$q^* = AA. \quad P_s \quad T_v \quad / P_s \quad T_b \quad ^{BB}$$

$$AA = A_0 + A_1 T + A_2 T^2 + A_3 T^3$$

$$BB = B_0 + B_1 T + B_2 T^2 + B_3 T^3$$

The saturation pressure is calculated according to the Antonie's equation, as Saha et al [8], where the values of $A_i$ 's and $B_i$ 's will also be found.

The energy balance for the each collector can be expressed as:

$$W_{CrM,i}\frac{dT_{cr,i}}{dt}=\eta_i A_{cr,i} I+\dot{m}_{f,cr} C_f \left(T_{cr,i,in} \quad T_{cr,i,out}\right) \tag{9}$$

$$T_{cr,i,out}=T_{cr,i}+(T_{cr,i,in}-T_{cr,i})EXP\left(-\frac{U_{cp,i}A_{cp,i}}{\dot{m}_{f,cr}C_f}\right) \tag{10}$$

Where, i=1,number of pipe in a collector

The collector efficiency equation is considered to be same as Alam et al [15].

The cyclic average cooling capacity is calculated by the equation

$$CACC=\dot{m}_{chill}C_{chill,f}\left(\int_{beginofcycletime}^{endofcycletime} T_{chill,in}-T_{chill,out}\;dt\right)\Big/ t_{cycle} \tag{11}$$

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

$$COP_{cycle}=\frac{\int_{beginofcycletime}^{endofcycletime}\dot{m}_{chill}C_{chill,f}\;\; T_{chill,in}-T_{chill,out}\;\; dt}{\int_{beginofcycletime}^{endofcycletime}\dot{m}_f C_f\;\; T_{d,in}-T_{d,out}\;\; dt} \tag{12}$$

$$COP_{sc}=\frac{\int_{beginofcycletime}^{endofcycletime}\dot{m}_{chill}C_{chill}\;\; T_{chill,in}-T_{chill,out}\;\; dt}{\int_{beginofcycletime}^{endofcycletime} n.A_{cr}Idt} \tag{13}$$

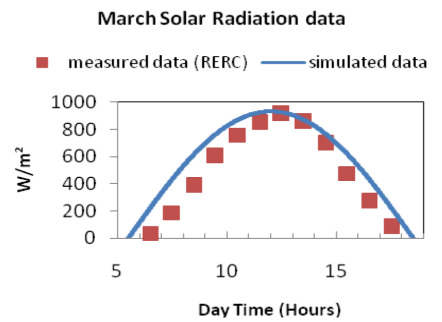
## 2.2 Simulation Procedure

Measured monthly maximum radiation data for Dhaka (Latitude $23^\circ 46'$ N, Longitude $90^\circ 23'$ E) has been used. This data is supported by the Renewable Energy Research Center (RERC), University of Dhaka. Results are generated based on solar data of Dhaka on a number of months during the summer and autumn seasons. Chiller configurations are same as Saha et al [8] and collector data are same as Alam et al [15]. During hot seasons in Dhaka, the sunrise time is at 5.5h and sun set at 18.5h, whereas maximum temperature varies between 30°C to 34ºC and minimum temperature varies between 20°C to 26ºC during this period. The maximum solar radiation, in the considered period of the year, for Dhaka station, varies between 980 W/m $^2$ to 1100 W/m $^2$. The input data are given in Table 1.

Implicit finite difference approximation method is applied to solve the set of differential Equations. The water vapor concentration in a bed is represented in Eq. 8. Where, the concentration $q$ 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 (beginning of the first day) is based on assumption. The pressure and concentration is 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 fulfilled, the process goes for the next time step. The tolerance for all the convergence criteria is $10^{-4}$. The program runs for consecutive several days (as it is set). After a few days the system appears to its steady state. In this paper all results are presented for the 3$^{rd}$ 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 illustrated in Alam et al. [15]. The nomenclature is attached in appendix.

## 3. RESULT AND DISCUSSION

14 collectors each are of 2.42m$^2$ has been taken into consideration for the present analysis. The number of collectors has been decided through the simulation data. The program is allowed to run with various numbers of collectors and different cycle time for several months during hot and summer season. First the driving temperature level which is reported as below 90 ºC for silica-gel water pair(Saha et al.[6] and Chua et al.[7]) is checked, then the performances has been checked. For the present case 14 collectors is the best option for which the performances do not affect too much and driving heat source temperature level can be controlled by adjusting the cycle time. Figure 2 presents a comparison between simulated and measured data of radiation for the months of March and August. It is seen that the model for the radiation shows a good agreement with measured data for March. The deviation seen in the model and



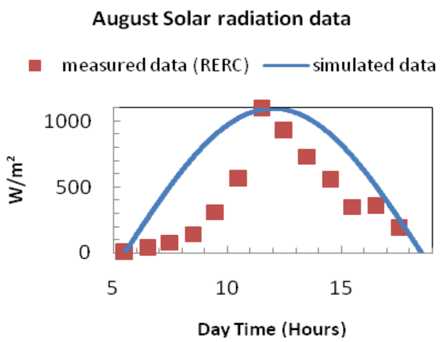




Fig 2. Solar radiation data for the months of March and August.

measured data for August is due to cloud coverage. Figure 3 illustrates the temperature histories of the collector outlet and bed temperature. It is seen that the bed temperatures are within the range of the temperature of driving heat source temperature, that is, below 90 ºC. It could be also observed that the half cycle time (heating or cooling) 1000s is required for March while 800s is required for August to reach the same temperature level. This is due to the solar radiation as solar radiation of August is higher than that of March. It can be also claimed that the number of collector can be reduced by adjusting the cycle time; however, excessive long or shorter cycle may affect the performance of the system. Therefore, it is essential to choose appropriate cycle time.

Figure 4 shows the performance of the chiller for different months and their optimum cycle time. It can be seen that almost same amount of cooling capacities for different months are achievable with different cycle times. It is also seen that cyclic average cooling capacity ($CACC$) of August is slightly higher than that of other months. This is due to the higher solar radiation in August. It is also observed that the values of cycle $COP$ for different months are also almost same though there are some little variation in late afternoon. For all cases cycle COP increases steadily up to late afternoon.

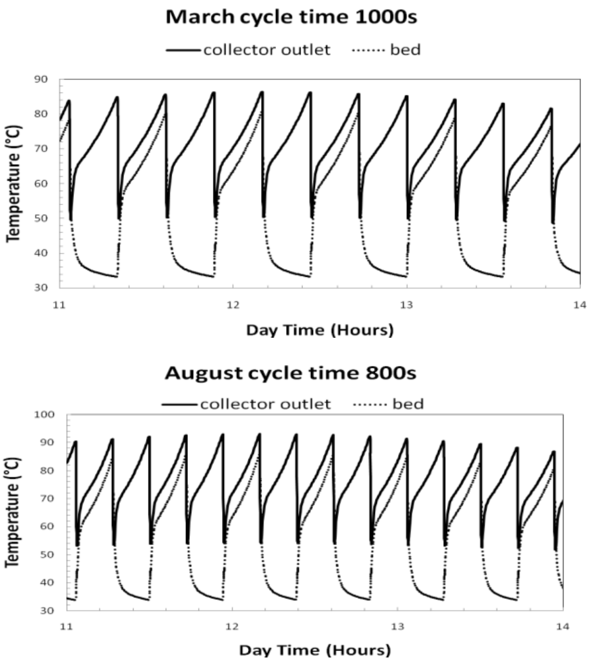



Fig 3. Temperature profile of the heat exchangers for the months of March and August

The increase of COP at after noon happens due to the inertia of collector materials. In afternoon, there is less heat input but due to the inertia of materials, of collector, 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 for the month of March 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.

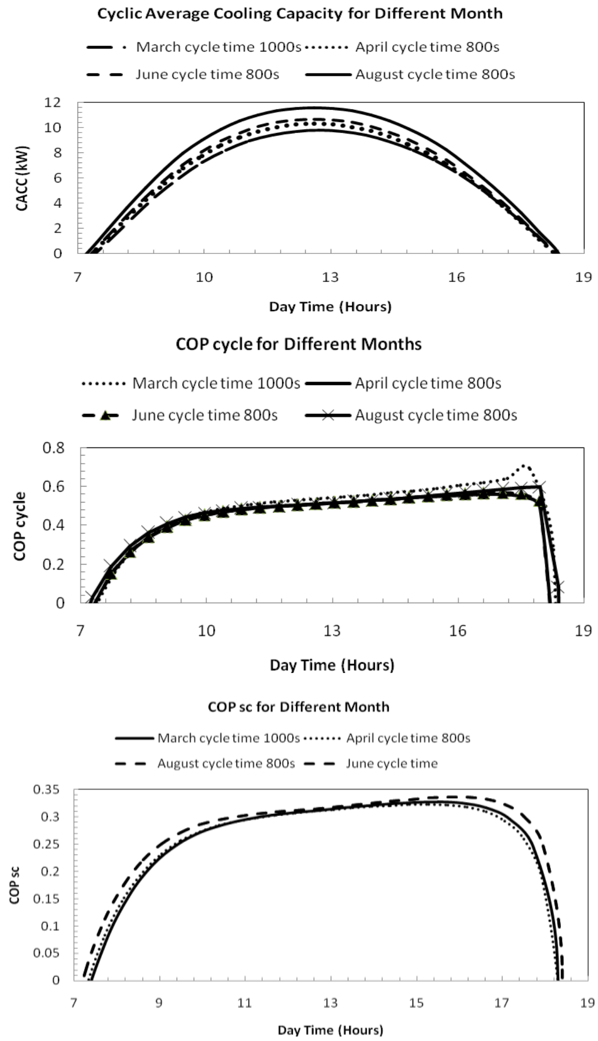



Fig 4. Comparative Performances of the chiller for different months

The maximum 0.34 solar COP is achievable with the proposed system in the region of Dhaka.

In air conditioning system, CACC and COP are not the only measurement of performances. If those values are higher but there is relatively higher temperature chilled water outlet, then the system may not provide comfortable temperature to the end user. From this context, the chilled water outlet temperatures for different months are presented in Fig.5. It can be seen that the chilled water outlet temperature varies from 8 ºC to 12.5 ºC for the month of March and it is from 7.5 ºC to 11.5 ºC for the month of August. It is well known that the less the fluctuation of chilled water temperature the better the performance of the system. However, the chilled water outlet temperature can be controlled by adjusting the flow rate of chilled water which can be the future work to analyze.

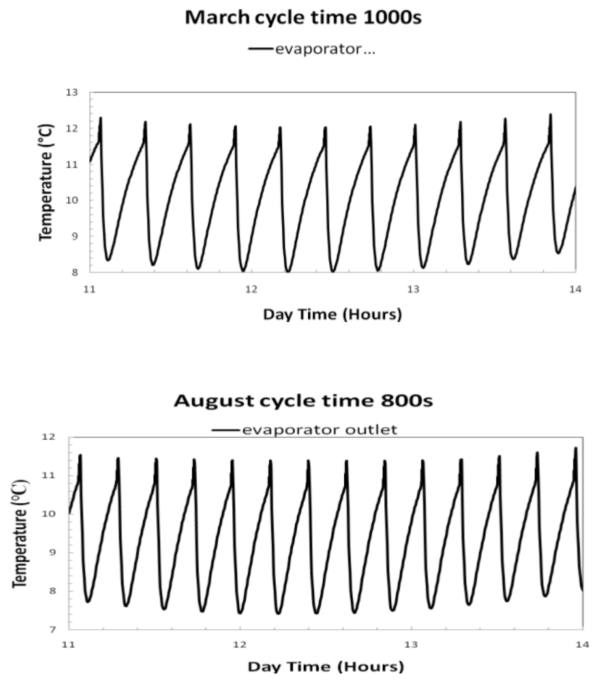



Fig 5. Chilled water outlet temperature for the month of March and August

## 4. CONCLUSION

An analytical investigation has been conducted to examine the prospect of solar driven adsorption air-condition system in Dhaka. A mathematical model is employed to investigate the performances of adsorption cooling system driven by solar collector for the climatic condition of Dhaka. 14 collectors each of area 2.42 $m^2$ can be installed to get desirable performance of the adsorption chiller with base run condition. For the two consecutive hot and humid seasons i.e. summer (which starts from April) and the autumn (which ends at September) seasons have been taken into consideration for the present analysis. It is found that 800s to 1000s cycle time is required during hot and humid seasons to get the optimum performance. The chiller is capable of producing 11 *kW* (3 RT) cooling during this hot period of the year. Also it is noticeable that, the increase in the cycle time increases the temperature of the silica gel bed. Therefore, it may be concluded that better performance is possible by decreasing the number of collectors and adjusting optimum cycle time.

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## 7. NOMENCLATURE

| Symbol | Meaning | Unit |
|---|---|---|
| $c_p$ | specific heat | $J/kgK$ |
| $I$ | solar radiation | $W/m^2$ |
| $\dot{m}$ | mass flowrate | $kg/s$ |
| Qst | heat of adsorption | $J/kg$ |
| $L$ | latent heat of vaporization | $J/kg$ |
| $M$ | mass | $kg$ |
| $q$ | adsorption capacity | $kg/kg_{ac}$ |
| $t$ | time | S |
| $T$ | temperature | $K$ |
| $U$ | heat transfer coefficient | $W/m^2K$ |
| $\dot{V}$ | volume flowrate | $m^3/s$ |
| $W$ | lumped capacitance | $J/K$ |

## 8. SUBSCRIPTS

| Symbol | Meaning |
|---|---|
| a | adsorber |
| s | Silica gel |
| amb | Ambient |
| cd | Condenser |
| CW | Chilled water |
| d | Desorber |
| ev | evaporator |
| floor | floor |
| fresh | fresh air |
| HW | hot water |
| Indoor | indoor |
| w | water |
| MW | cooled water |
| sc | solar collector |

## 9. MAILING ADDRESS

**R. A. Rouf**
School of Engineering and Computer Science,
Independent University, Bangladesh, Dhaka, Bangladesh.
**E-mail:** rifat_rouf@yahoo.com,