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

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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 m²) are essential to achieve the required heat source temperature (around 85°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°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°C heat source with 30°C cooling source. To utilize even lower temperature waste heat (around 50°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.


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 adsorbent(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 precooling 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 precooling/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 Fig. 1. The characteristics of absorbent/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

\[ \frac{d}{dt}\left\{ (W_{st} C_{st} + W_{ed} C_{ed} + W_{cd} C_{cd}) \right\} = \Delta H \cdot W_{st} \cdot \frac{dq_{wl}}{dt} + \delta \cdot W_{ed} C_{ed} \cdot \frac{dq_{cd}}{dt} (T_{cd} - T_{cd}) + \dot{m}_{fj} C_f (T_{bed} - T_{bed}) \]

(1)

\[ T_{bed,out} = T_{bed} + (T_{bed,in} - T_{bed}) \cdot \exp(-U A_{bed} / \dot{m}_f C_f) \]

(2)

Where, \( \delta \) equals to zero or one depending whether adsorber bed is working as desorber or adsorber.

The energy balance equation for the condenser is represented by

\[ \frac{d}{dt}\left\{ (W_{ct} C_{ct} + W_{cd} C_{cd}) \right\} = -L \cdot W_{ct} \cdot \frac{dq_{ct}}{dt} + W_{ct} C_{ct} \cdot \frac{dq_{cd}}{dt} (T_{cd} - T_{cd}) \]

\[ + \dot{m}_{f,cd} C_f (T_{cd,in} - T_{cd, out}) \]

(3)

\[ T_{cd,out} = T_{cd} + (T_{cd,in} - T_{cd}) \cdot \exp(-U A_{cd} / \dot{m}_f C_f) \]

(4)

The energy balance equation for the evaporator is

\[ \frac{d}{dt}\left\{ (W_{et} C_{et} + W_{ed} C_{ed}) \right\} = -L \cdot W_{et} \cdot \frac{dq_{et}}{dt} + W_{et} C_{et} \cdot \frac{dq_{ed}}{dt} (T_{ed} - T_{cd}) \]

\[ + \dot{m}_{f,ed} C_f (T_{cd,in} - T_{cd, out}) \]

(5)

\[ T_{et} = T_{et, in} + (T_{et} - T_{et}) \cdot \exp(-U A_{et} / \dot{m}_f C_f) \]

(6)
The mass balance of the refrigerant inside the evaporator is expressed as

\[ m_{f,\text{chill}} \cdot C_f \left( T_{\text{chill, in}} - T_{\text{chill, out}} \right) + \dot{m}_{f,\text{chill}} \cdot C_f \left( T_{\text{chill, in}} - T_{\text{chill, out}} \right) = W \left( dq_{\text{in}} - dq_{\text{out}} \right) \tag{5} \]

The collector efficiency equation is considered to be identical for the consecutive days. The design and steady state condition from day 3 i.e. all output appeared 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 operating conditions used in the simulation are given in Table I (attached in appendix).

The concentration in bed is

\[ \frac{dq}{dt} = k\text{asp}(q^* - q) \tag{8} \]

where,

\[ k\text{asp} = D_s \cdot \exp(-E_a / (R \cdot g \cdot T)) \]

\[ D_s = 15.6 \cdot D_{chill} / (R \cdot p^3) \]

\[ q^* = AA \cdot (P_s(T_b) / P_s(T_b))^\alpha \]

\[ AA = A_0 + A_1 \cdot T + A_2 \cdot T^2 + A_3 \cdot T^3 \]

\[ BB = B_0 + B_1 \cdot T + B_2 \cdot T^2 + B_3 \cdot T^3 \]

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 Claus “et al.” [24]. The cyclic average cooling capacity (CAC) is calculated by the equation

\[ \text{CAC} = \frac{m_{\text{chill}} \cdot C_{\text{chill}} \cdot \int_{t_{\text{begin of cycle}}}^{t_{\text{end of cycle}}} (T_{\text{chill, in}} - T_{\text{chill, out}}) dt}{t_{\text{cycle}}} \tag{9} \]

The cycle COP (coefficient of performance) and solar COP in a cycle (COP_s) are calculated respectively by the equations

\[ \text{COP}_{\text{cycle}} = \frac{\int_{t_{\text{begin of cycle}}}^{t_{\text{end of cycle}}} m_{\text{chill}} \cdot C_{\text{chill}} \cdot \left( T_{\text{chill, in}} - T_{\text{chill, out}} \right) dt}{\int_{t_{\text{begin of cycle}}}^{t_{\text{end of cycle}}} m_{\text{chill}} \cdot C_{\text{chill}} \cdot \left( T_{\text{chill, in}} - T_{\text{chill, out}} \right) dt} \tag{10} \]

\[ \text{COP}_{\text{s}} = \frac{\int_{t_{\text{begin of cycle}}}^{t_{\text{end of cycle}}} m_{\text{chill}} \cdot C_{\text{chill}} \cdot \left( T_{\text{chill, in}} - T_{\text{chill, out}} \right) dt}{\int_{t_{\text{begin of cycle}}}^{t_{\text{end of cycle}}} m \cdot A_i \cdot dt} \tag{11} \]

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 (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 non-linear 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 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]).
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 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.
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 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.
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

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A_{\text{bed}})</td>
<td>Adsorbent bed heat transfer area</td>
<td>(2.415 m^2)</td>
</tr>
<tr>
<td>(U_{\text{bed}})</td>
<td>Heat transfer coefficient of bed</td>
<td>(1724.14 W/m^2 K)</td>
</tr>
<tr>
<td>(W_{\text{ex}})</td>
<td>Heat exchanger tube weight (Cu)</td>
<td>51.2 kg</td>
</tr>
<tr>
<td>(W_{\text{ex}})</td>
<td>Heat exchanger fin weight (Al)</td>
<td>64.04 kg</td>
</tr>
<tr>
<td>(A_{\text{ev}})</td>
<td>Evaporator heat transfer area</td>
<td>(1.91 m^2)</td>
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<tr>
<td>(U_{\text{ev}})</td>
<td>Evaporator heat transfer coefficient</td>
<td>(2557.54 W/m^2 K)</td>
</tr>
<tr>
<td>(W_{\text{ev}})</td>
<td>Evaporator heat exchanger tube weight (Cu)</td>
<td>12.45 kg</td>
</tr>
<tr>
<td>(A_{\text{con}})</td>
<td>Condenser heat transfer area</td>
<td>(3.73 m^2)</td>
</tr>
<tr>
<td>(U_{\text{con}})</td>
<td>Condenser heat transfer coefficient</td>
<td>(4115.23 W/m^2 K)</td>
</tr>
<tr>
<td>(W_{\text{con}})</td>
<td>Condenser heat exchanger tube weight (Cu)</td>
<td>24.28 kg</td>
</tr>
<tr>
<td>(N_p)</td>
<td>Each collector area</td>
<td>2.415 m²</td>
</tr>
<tr>
<td>(W_{\text{ex}})</td>
<td>Weight of each pipe including absorber sheet</td>
<td>0.4913 kg</td>
</tr>
<tr>
<td>(m_{\text{f,con}})</td>
<td>Total mass flow rate to CPC panel or to</td>
<td>1.3 kg/s</td>
</tr>
<tr>
<td>(m_{\text{f,bed}})</td>
<td>Condenser water flow rate to adsorber</td>
<td>1.3 kg/s</td>
</tr>
<tr>
<td>(W_{\text{sil}})</td>
<td>Weight of silica gel in each bed</td>
<td>47 kg</td>
</tr>
<tr>
<td>(m_{\text{f,exp}})</td>
<td>Liquid refrigerant inside evaporator initially</td>
<td>50 kg</td>
</tr>
<tr>
<td>(m_{\text{f,cond}})</td>
<td>Cold water flow rate to condenser</td>
<td>1.3 kg/s</td>
</tr>
<tr>
<td>(m_{\text{f,bed}})</td>
<td>Chilled water flow rate</td>
<td>0.7 kg/s</td>
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<tr>
<td>(W_{\text{con}})</td>
<td>Condenser refrigerant inside condenser</td>
<td>0.0 kg</td>
</tr>
<tr>
<td>(\Delta H)</td>
<td>Heat of adsorption (silica gel bed)</td>
<td>(2.81 E + 06 J/kg)</td>
</tr>
<tr>
<td>(R_{\text{gas}})</td>
<td>Water gas constant</td>
<td>(4.62 E + 02 J/kg K)</td>
</tr>
<tr>
<td>(Ea)</td>
<td>Activation energy</td>
<td>(2.33 E + 06 J/kg)</td>
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<tr>
<td>(D_{\text{v0}})</td>
<td>Diffusion coefficient</td>
<td>(2.54 E - 04 m^2 /s)</td>
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<tr>
<td>(R_p)</td>
<td>Particle diameter (silica gel)</td>
<td>(0.35E - 03 m)</td>
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<tr>
<td>(T_{\text{surf}})</td>
<td>Cooling source temperature</td>
<td>30 °C</td>
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<tr>
<td>(T_{\text{chill,con}})</td>
<td>Chilled water inlet temperature</td>
<td>14 °C</td>
</tr>
<tr>
<td>(C_{\text{c,H}\text{,j}})</td>
<td>Specific heat of water (liquid phase)</td>
<td>(4.18 E + 03 J/kg K)</td>
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<tr>
<td>(C_{\text{c,H}\text{,v}})</td>
<td>Specific heat of water (vapor phase)</td>
<td>(1.89 E + 03 J/kg K)</td>
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<tr>
<td>(C_{\text{c,Si}})</td>
<td>Specific heat of copper (Cu)</td>
<td>(386 J/kg K)</td>
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<tr>
<td>(C_{\text{Al}})</td>
<td>Specific heat of aluminum (Al)</td>
<td>(905 J/kg K)</td>
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<tr>
<td>(C_{\text{Si}\text{,j}})</td>
<td>Specific heat of silica gel (Si)</td>
<td>(924 J/kg K)</td>
</tr>
<tr>
<td>(L)</td>
<td>Latent heat of vaporization (water)</td>
<td>(2.6 E + 06 J/kg)</td>
</tr>
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</table>
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