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Computer Simulation of the Performance of An Open Solar Assisted Desiccant Cooling System

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Abstract

The performance characteristics of an open- cycle solar assisted desiccant cooling system were evaluated using computer simulation . A computer program was built and developed to simulate the effect of various design and operating conditions on the performance of the individual components making up the simulated system. They include a V-corrugated plate type solar air heater, a rotary silica gel desiccant dehumidifier, a sensible cooler and an evaporative cooler. The overall system performance was evaluated in terms of several parameters including the supply air temperature, the sensible cooling capacity and the coefficient of performance of the system. The effect of operating parameters such as the outdoor air temperature, outdoor moisture content, regeneration temperature, dehumidifier efficiencies, sensible cooler effectiveness and evaporative cooler effectiveness was incorporated in the analysis. Actual hourly weather data for Benghazi on selected clear summer days were used to assess the system performance during the day. The simulation was accomplished with a direct supply of hot air, from the solar air heater, for regenerating the dehumidifier bed. The performance of the solar air heater and the desiccant dehumidifier was first simulated independently to assess their compatibility when used in the system.

Simulation results of the overall system performance showed that the heat exchanger effectiveness, evaporative cooler effectiveness and ambient conditions have the major influence on the system performance. Whereas, the influence of dehumidifier performance and regeneration temperature on the system performance was less pronounced. It was found that the simulated system can generate a cool supply of air at acceptable comfort conditions for most hours of typical clear summer days under the local weather conditions of Benghazi. The supply air temperature obtained from the system was found to be nearly constant during the selected days and at an average value for Benghazi of 18°C.

Keywords: Cooling Systems , Desiccant Dehumidifier, Solar Energy

1. Introduction

Solar operated open-cycle cooling systems using solid desiccant dehumidifiers have been proposed as an attractive alternative to the conventional vapor compression units. In these systems air is dried by passing it over the desiccant bed and the heat of adsorption is removed by sensible cooling. The air is further cooled by an evaporative cooler and then it is supplied to the conditioned space. The major components of these systems include desiccant dehumidifiers, heat exchangers, evaporative coolers and solar collectors.

Dunkle [1] discussed an open-cycle desiccant cooling system designed for air conditioning in humid tropical or sub tropical areas. Rush et al.[2] classified the operation of open-cycle adsorption cooling systems into the ventilation mode and the recirculation mode. Nelson et al.[3] made a feasibility study of open-cycle air conditioning systems that use solid desiccants and solar energy. Collier et al.[4] presented a review of the thermodynamics of three desiccant cooling systems; the ventilation cycle, the recirculation cycle and Dunkle's cycle. Worek and Charoensupaya[5] modeled numerically the performance of an adiabatic solid desiccant cooling system operating in the ventilation mode. Maclaine -cross[6] reported a new class of high performance, solid desiccant open cooling cycles using low temperature solar, gas or waste heat.

Joudi and Madhi [7] investigated experimentally the performance of a solar assisted open-cycle adsorption evaporative cooling system. Their tests were carried out hourly with a direct supply of hot air from a V-corrugated plate type solar air heater for regenerating the desiccant bed of a rotary dehumidifier. They found that it is possible to generate a cool supply of air at satisfactory comfort conditions for buildings cooling, using solar energy only for regeneration, for most clear summer days in Benghazi.

The dehumidifier of a desiccant cooling system may be arranged as a cooled matrix. Also, it may be arranged as an adiabatic rotating wheel with axial parallel or counter flow of both process and regenerating air streams. Van den Bulck et al.[8] indicated that the best prospect is for the rotary counter flow conception since it combines high performance with compactness.

The investigations of heat and mass transfer in porous media, as in a dehumidifier, range from mathematical modeling to the development of empirical design tools. Banks and others [9, 10, 11, 12, 13] introduced various techniques to compute the outlet state of an adiabatic dehumidifier. He concluded that for an energy recovery regenerator a previous linear analogy method was satisfactory, whereas a non-linear analogy method is required for regenerative dehumidifiers.

In addition, heat exchangers and evaporative coolers are usually used together to cool the dehumidified warm process air. Heat exchangers are usually counter-flow, air-to-air, rotary thermal regenerators or cross-flow, air-to-water, finned tube heat exchangers. Van den Bulck et al.[14] reported a value of 0.9 for the effectiveness of counter-flow heat exchangers.

Many attempts have been made locally to improve the performance of the conventional evaporative cooler and its suitability to provide comfort requirements. For evaporative coolers used in desiccant cooling systems, Joudi and Madhi [7] reported an average value of 0.85 for their effectiveness.

Solar air heaters are a basic component of solar desiccant cooling systems. Dhiman and Tiwari [15] presented an analytical model to study the performance of a two-channel flat plate air heater. Biondi et al. [16] analyzed the technical performance of seven solar air heaters having simple flat plate absorbers. The idea of a V-groove in the absorber plate for increasing the solar absorptance and turbulence in the air was given by Hollands [17]. An angle of 55-60° had been recommended by Sayigh [18] for such corrugations. Hollands and Shewen [19] who studied the optimization of flow passage geometry of solar air heaters. Joudi and Mohammed [20] investigated experimentally the performance of a V-corrugated type solar air heater for use in a desiccant cooling systems.

The present work is a computer simulation study of the effect of various design and operating conditions on the performance of an open-cycle solar assisted desiccant cooling system. The simulation is for an actual system that had been tested experimentally in previous works. Comparison was successfully carried out to validate the simulation technique.

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2. **Mathematical Modeling**

Fig.(1) shows a schematic diagram of the simulated system employed in this work. In this system , which operates in the ventilation mode , outdoor process air is passed through the dehumidifier where it's moisture is adsorbed by the desiccant material and the process air is dried. The drying process releases an amount of energy , which for a common adsorbent like silica gel is about 5% to 15% greater than the heat of condensation of water vapor. Most of the released heat is transferred to the process air stream and a small amount is retained in the desiccant bed due to the low bed heat capacity. The dry warm air is then sensibly cooled by a water cooled cross-flow heat exchanger and is further cooled by passing it through an evaporative cooler before it enters the conditioned space. Outdoor air is heated to the regeneration temperature as it passes through a V- corrugated absorber plate type solar air heater. The heated air is then passed through the dehumidifier to remove the moisture from the desiccant bed . After that, the air is exhausted to the atmosphere. Fig. (2) shows the processes of this cycle on a psychometric chart. The analysis of system components is presented in what follows.

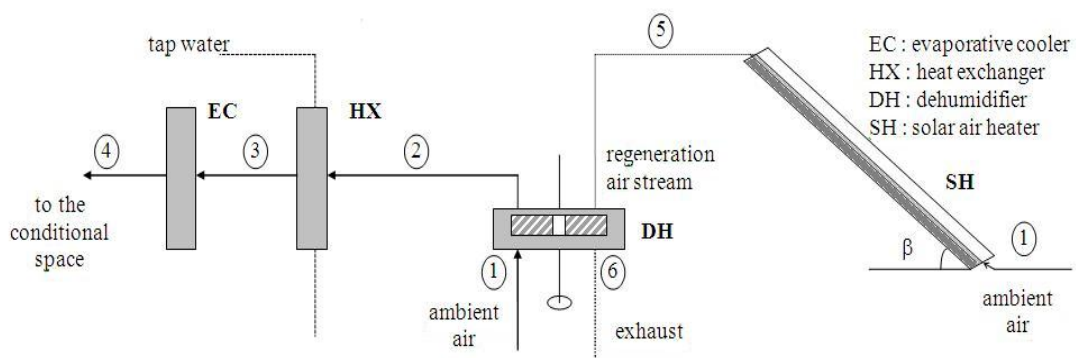

Fig (1) : schematic diagram of the simulated OCDCS

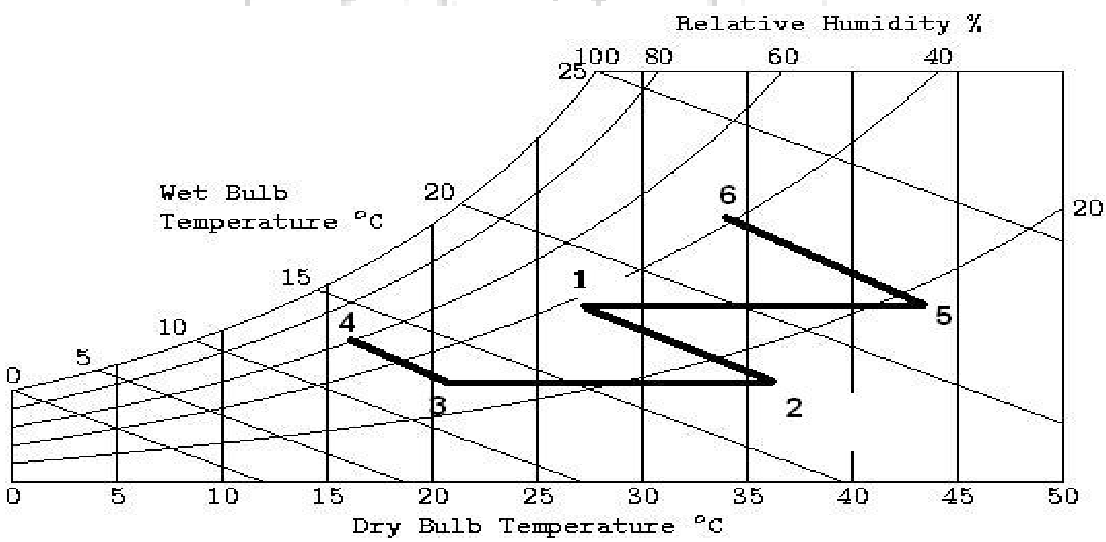

Fig (2) : The Process of the simulated OCDCS on Psychometric chart

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a : Solar Air Heater

The solar air heater used in the present simulation is a V-corrugated absorber plate type with an angle of corrugation of 60°. It uses a single glass cover with air flowing through the triangular passages between the corrugated absorber and the rear flat plate, measuring 1m * 2m. It was tested experimentally by Ref.[20].

The useful energy gain from a solar air heater can be calculated from the Hottel - Whillier - Bliss equation [21] .

$$q_u = F_R \left[S - U_L \left(T_{in} - T_a\right)\right] \quad (1)$$

Where,

$$F_R = \frac{G\,Cp}{U_L}\left[1 - \exp\left(\frac{-F'\,U_L}{G\ Cp}\right)\right] \quad (2)$$

Solar air heater performance is usually expressed by the collection efficiency defined by the equation:

$$\eta_c = \frac{q_u}{I_{gb}} \quad (3)$$

The overall heat transfer coefficient is the sum of the top, bottom and edge heat loss factors. In the present work the top heat loss factor was calculated by using the analytical technique. This technique is based on the heat transfer analysis of the absorber – cover–ambient subsystem. Using the concept of thermal resistance, the top heat loss factor for a single glass cover solar air heater is given as:

$$U_t = \frac{1}{R_1 + R_2} \quad (4)$$

Where R_1 and R_2 are the resistance to heat transfer from absorber to glass cover and from glass cover to ambient air, respectively. These resistance can be written as:

$$\left.\begin{aligned} R_1 &= \frac{1}{h_{c_{p\text{-}g}} + h_{r_{p\text{-}g}}} \\ R_2 &= \frac{1}{h_w + h_{r_{g\text{-}a}}} \end{aligned}\right] \quad (5)$$

Where the radiative heat transfer coefficients were calculated using conventional heat transfer equations, as given in Ref. [21], with the sky temperature being related to the ambient temperature by an equation suggested by Swinbank [22]:

$$T_s = 0.0552\, T_a^{1.5} \tag{6}$$

The natural connective heat transfer coefficient between the V– corrugated absorber plate and the flat glass cover was obtained from an equation suggested by Meyer et al. [23] in the form:

$$\overline{Nu}_L = c\, Gr_L^n \tag{7}$$

Where c and n are constants depending on the collector tilt angle and the Vee – aspect ratio which is the ratio of mean plate spacing to Vee – height and have the values of 0.07 and 0.39 respectively for this case.

The wind heat transfer coefficient was calculated by the dimensional equation [21].

$$h_w = 5.7 + 3.8\, V \tag{8}$$

The estimation of the top heat loss factor by using heat transfer analysis required an iterative procedure since the glass cover temperature is unknown. In the present work, the glass cover temperature was first approximated using an equation suggested by Samdarshi and Mullick [24] as:

$$T_g = T_p - \frac{T_p - T_a}{1 + f} \tag{9}$$

where the factor f is given by the empirical equation:

$$f = \sqrt{5 / h_w}$$

After the evaluation of the top heat loss factor using the above equations, the mean glass cover temperature may then be determined analytically from.

$$T_g = T_p - \frac{U_t\left(T_p - T_a\right)}{\left(1 / R_1\right)} \tag{10}$$

On an hourly basis the absorbed solar radiation is calculated from the equation [21]:

$$S = D_f \,.\, S_f \left[I_b\, (\tau\alpha)_{e,b} + I_d (\tau\alpha)_{e,d} + I_r (\tau\alpha)_{e,r}\right] \tag{11}$$

Where $(\tau\alpha)_{e,b}$, $(\tau\alpha)_{e,d}$ and $(\tau\alpha)_{e,r}$ are the effective transmittance– absorptance product for beam, diffuse and reflected solar radiation, respectively. The dust factor which accounts for the reduction in absorbed energy due to dust on the glass cover was assumed equal to 0.99, while the shade factor which accounts for the reduction in absorbed energy when some of the air heater structure intercepts solar radiation was assumed 0.99 during midday and 0.98 before and afternoon [20].

The three components of solar radiation I_b , I_d and I_r received by a south–-facing surface with any angle of inclination were calculated using standard equation from Ref. [25]:

$$I_b = I_{DN} \cos\theta_i ,$$

$$I_d = I_{DN}\, C \left(\frac{1+\cos\beta}{2}\right) \qquad (12)$$

$$I_r = \rho_g\, H_{gb} \left(\frac{1-\cos\beta}{2}\right)$$

Solar irradiance calculations are usually based on values published by ASHRAE [26] for the apparent extraterrestrial solar intensity A, the atmospheric extinction coefficient B, and the diffuse radiation factor C. The values of these three variables are listed for the 21st. day of each month. In the present work the values of A, B, and C are calculated as a function of day number using equations derived by Joudi [27]. These equations are:

$$A = 1158\left[1+0.066\cos\left(\frac{360N}{370}\right)\right]$$

$$B = 0.175\left[1-0.2\cos(0.93\,N)\right] - 0.0045\left[1-\cos(1.86\,N)\right] \qquad (13)$$

$$C = 0.0965\left[1-0.42\cos\left(\frac{360\,N}{370}\right)\right] - 0.0075\left[1-\cos(1.95\,N)\right]$$

Where N is the day number in the year.

The effective transmittance–absorptance products for the beam, diffuse and reflected components of solar radiation were calculated by using the procedure outlined in Ref.[21].

For a collector with a triangularly corrugated absorber and a flat rear plate attached to the absorber, which is the case at hand, Duffie and Beckman [21] reported the following equation for the collector efficiency factor:

$$F' = \left[1+\frac{U_L}{\dfrac{h_c}{\sin\frac{\gamma}{2}}+\dfrac{1}{\frac{1}{h_c}+\frac{1}{h_r}}}\right]^{-1} \qquad (14)$$

In equation 14, the heat transfer coefficient between the air stream and the absorber plate and between the air stream and the rear plate are assumed equal. Also, the heat transfer coefficient between the air and the absorber plate is divided by sine half the angle of corrugation to account for the corrugated absorber. Furthermore, the temperature difference between the absorber plate and the rear plate has a negligible effect on F' within the range of operation temperature employed and may be neglected.

The convective heat transfer coefficient h_c in the equation for the collector efficiency factor was determined from the correlation that was given by Anderson [28] for fully developed turbulent flow between parallel plates, with the upper being heated and the lower plate insulated, in the following form:

$$Nu_{D_h} = 0.0158 \quad Re_{D_h}^{0.8} \; Pr^{1/3} \tag{15}$$

The air properties are evaluated at the mean fluid temperature through the collector which was shown by Klein et al. [29] to be:

$$T_{f,m} = T_{in} + \frac{q_u}{U_L F_R}\left(1 - \frac{F_R}{F'}\right) \tag{16}$$

Finally , the mean absorber plate temperature and the heated fluid outlet temperature can be evaluated from:

$$T_{p,m} = T_{in} + \frac{q_u}{U_L F_R}\left(1 - F_R\right) \tag{17}$$

$$T_{F,out} = \left(T_a + \frac{S}{U_L}\right)\left(1 - \exp\left(-F'.U_L \,/\, G\,C_P\right)\right) + T_{F,in}.\ \exp\left(-F'\,U_L \,/\, G\,C_P\right) \tag{18}$$

The evaluation of the useful energy gain requires the knowledge of the overall heat loss factor and the internal fluid heat transfer coefficient. These factors are themselves functions of the mean plate and mean fluid temperatures. However, the mean plate temperature as well as the mean fluid temperature can be determined only by knowing the values of the useful energy gain and the overall heat loss factor. Therefore, an iterative procedure must be used to evaluate the solar air heater performance.

b : Desiccant Dehumidifier

In the present work the concept of F_i potentials which was introduced by references [9, 10, 11], and the potential efficiencies technique suggested by references [12& 13] were used together to find the outlet state of the process air from the silica gel rotary dehumidifier. The simplified non - linear potentials used here are [30]:

$$F_1 = \frac{-2865}{T^{1.49}} + 4.344\ W^{0.8624} \tag{19}$$

$$F_2 = \frac{T^{1.49}}{6360} - 1.127\, W^{0.07969} \qquad (20)$$

Where lines of constant F_1 resemble lines of constant enthalpy on a psychrometric chart, and lines of constant F_2 resemble lines of constant relative humidity.

After the evaluation of the potentials F_1 and F_2 for the inlet conditions of both process and regeneration air streams, the combined potentials for the outlet process air stream are then obtained from the following regenerative dehumidifier efficiencies [13] :

$$F1_{P,o} = \eta_{F_1}\left(F1_{reg,i} - F1_{P,i}\right) + F1_{P,i} \qquad (21)$$

$$F2_{P,o} = \eta_{F_2}\left(F2_{reg,i} - F2_{P,i}\right) + F2_{P,i} \qquad (22)$$

The outlet process air temperature and moisture content from the dehumidifier are then obtained by solving equations 19 and 20 simultaneously with the combined potentials calculated from equation 21 and 22, respectively. Also, from the above equations it can be seen that low values of η_{F_1} , close to 0, coupled with high values of η_{F_2} , close to 1, result in outlet process air states close to that obtained with an ideal dehumidifier. In the present work three pairs of (η_{F_1}, η_{F_2}) values were used in the subsequent calculations. They were (0.05, 0.95) , (0.08, 0.8) and (0.1,0.7). These three pairs refer to dehumidifiers having "good", "medium" and "poor" performances, respectively, as indicated by Leersum [31] .

c : Sensible and Evaporative Coolers

The dehumidified process air is sensibly cooled by a water cooled cross - flow heat exchanger. The outlet state from the heat exchanger was calculated using the usual heat exchanger effectiveness correlation given as :

$$T_{P,o} = T_{P,i} + \eta_{hx}\left(T_{w,i} - T_{P,i}\right) \qquad (23)$$

The inlet process air state to a component is assumed uniform at the bulk mean outlet state of the upstream component. Also, the moisture content of the process air through the heat exchanger is assumed constant because no mass transfer occurs in this heat exchanger. The temperature of the cooled water entering the heat exchanger was assumed equal to 3°C greater than the wet bulb temperature of the ambient air. This was done in order to obtain more factual conditions of water and process air in case a cooling tower is used in large scale applications to provide cooling water for the heat exchanger.

The process air leaving the sensible heat exchanger is then evaporatively cooled through a wet pad evaporative cooler before it enters the conditioned space. The evaporative cooler was modeled by a simple effectiveness correlation. The dry bulb temperature of the process air leaving the evaporative cooler is then calculated from :

$$T_{db,o} = T_{db,i} + \eta_{ev}\left(T_{db,i} - T_{wb,i}\right) \qquad (24)$$

d : Overall System Performance

The " efficiency" of the desiccant cooling system is usually expressed in terms of two important parameters. These parameters are the cooling capacity and the coefficient of performance COP. The cooling capacity of the process air supplied by the system is usually defined as the difference in enthalpy between the supply air and any given interior condition. However, it should be pointed out that the actual cooling capacity in such systems is only sensible. Moisture generated within the space is inadvertently picked up by the cool air rather than removed as in the normal workings of mechanical air conditioners, It can not be truly said that evaporative coolers remove latent heat. This is particularly true in the ventilation mode. Joudi and Madhi [7] indicated that the sensible cooling capacity based on temperature difference is more factual than calculating it based on enthalpy difference. The cooling capacity of the current system was calculated as [7] .

$$CC = m_P \; Cp \, (T_R - T_S) \qquad (25)$$

Several definitions exist for the coefficient of performance of desiccant - evaporative solar cooling systems. They are reviewed and appraised by Ref.[7] who defined the COP of these systems as the heat removed from the process air stream by the cycle divided by energy input to the cycle . Energy input includes electric energy to circulate air, water and rotating the desiccant wheel. Solar energy is considered free. On this basis, the coefficient of performance with reference to Fig. (2) is,

$$COP = \frac{RE}{EP} \qquad (26)$$

where $RE = m_P \, Cp \, (h_1 - h_4)$ (27)

3. Computational Model

The computational model consists of a main program and a number of subprograms which are coupled with the main program , all written in Visual BASIC language . The main variables in the computational model which require an iterative procedure for their evaluation were defined as double precision variables. A brief description of the subprograms which were used to evaluate the current system performance is presented here .

1. Input of the initial data for the computational model.
2. Determination of the sun position in the sky for each hour during the simulation period.

3. Calculation of the hourly direct, diffuse, reflected, and global solar radiation, on a south–facing collector with any angle of inclination .
4. Input of the design parameters of the simulated collector and the meteorological data.
5. Performance evaluation of the V-corrugated absorber plate type solar air heater.
6. Calculation of the hourly values of solar radiation absorbed by the absorber plate.
7. Calculation of the physical properties of air as a function of air temperature.
8. Calculation of the overall heat loss coefficient by an iterative technique with an estimate of glass plate temperature.
9. Performance simulation of the regenerative dehumidifier
10. Determination of the required psychrometric properties of the moist air at various stages of the system.
11. Computation of the processed air outlet temperature from the heat exchanger.
12. Determination of the psychrometric properties of the processed air leaving the evaporative cooler.
13. Computation of the overall desiccant cooling system performance.

The main program calls the subprograms to execute the necessary calculations as they are needed, beginning with input data and ending up with the required results.

4. Results & Discussion

The effect of various design and operating conditions on the performance of the open - cycle desiccant cooling system was examined using the developed simulation technique. These conditions include heat exchanger effectiveness, evaporative cooler effectiveness, dehumidifier efficiencies, regeneration temperature, ambient temperature and ambient relative humidity. Also, the performance of the cooling system was evaluated during the simulated period using actual and severe weather data and solar heated air for regenerating the desiccant bed. The performance of this system was evaluated in terms of three parameters which are supply air temperature, sensible cooling capacity and system coefficient of performance. Fig. (3) shows the processes of the simulated cycle for two different air inlet conditions on a psychrometric chart. The effectiveness of the heat exchanger and the evaporative cooler were set at their reference values of 0.85 and 0.8, respectively. The term "medium" which refers to the dehumidifier potential efficiencies of 0.08 & 0.8 was taken as the reference for the dehumidifier performance. In this figure , fresh ambient air is drawn through the desiccant bed where it’s water vapor is adsorbed accompanied with a rise in air temperature as shown in process 1-2. The dry heated air is then cooled sensibly by the heat exchanger to point 3. Thereafter, it is cooled evaporatively to state 4 after which it is ready for supply to a conditioned room at state R . A second open-loop air stream starts

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at point 1 and passes through the solar air heater where it is heated to state 5. This air stream regenerates the desiccant bed and is discharged to the atmosphere at point 6. The temperature of the air at various states in the cycle of Fig. (3) was calculated using fixed values for the heat exchanger and evaporative cooler effectiveness and medium performance dehumidifier. In this manner the temperature at the end of each process is determined independently of mass flow rate of the air or actual component sizes [30].

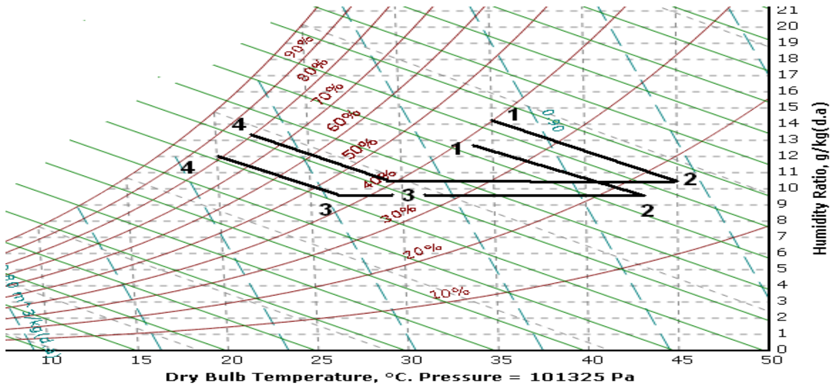

Fig (3) : The processes of the simulated cycle for two actual air inlet conditions on a psychometric chart

Fig. (4) shows the effect of ambient relative humidity, regenerative air temperature and ambient air temperature on the supply air temperature obtained from the complete system. It is observed that the supply air temperature increases for all cases with ambient relative humidity. The simulated desiccant cooling system can obviously provide supply air at a much lower temperature than that obtained using an evaporative cooler only. It was found that at an ambient condition of 35°C dry bulb (d.b.) and 5% relative humidity and a regeneration temperature of 65 °C, the system can provide cool supply air at a temperature of 12°C d.b. when reference values for the effectiveness of the heat exchanger and the evaporative cooler and a medium dehumidifier are employed. This supply air temperature increases to about 19.5 °C d.b. at an ambient condition of 35 °C d.b. and 35% relative humidity with all other conditions remaining the same . This definitely highlights the effect of relative humidity on the system performance.

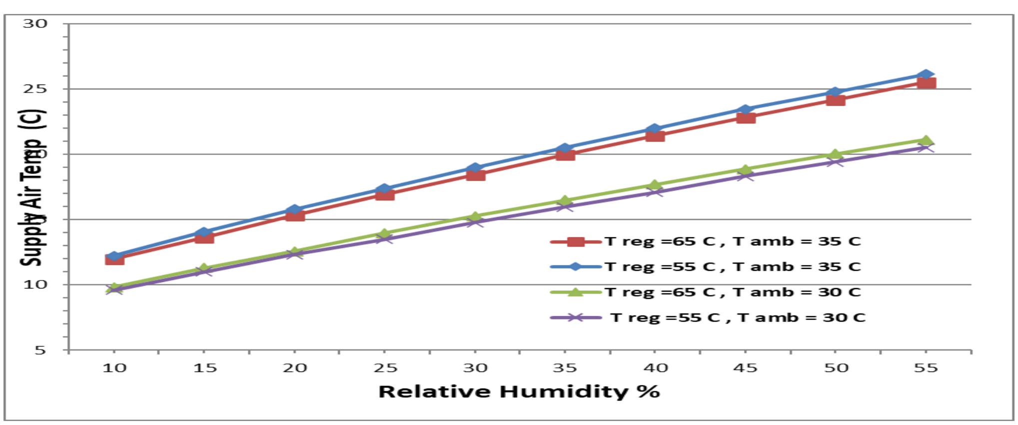

Fig (4) : Variation of supply air temperature with ambient air relative humidity for different ambient temperatures

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The effect of regeneration air temperature on the supply air temperature is shown in Fig. (5). In this figure the effectiveness of the heat exchanger and the evaporative cooler were fixed at their reference values . As the regeneration temperature increases from 45 °C to 95 °C , the supply air temperature decreases by about 2.5 , 2.7 and 2.8 °C with ambient air relative humidity of 30 , 40 and 50% . This reduction in the supply air temperature is mainly due to the fact that as the regeneration temperature increases, the moisture removal from the process air by the silica gel increases. Thus, the moisture content of the process air is lowered further so that after effective sensible cooling it enters the evaporative cooler at a lower wet bulb temperature. However, it should be pointed out that the sensible cooling required, to achieve a certain inlet wet bulb temperature to the evaporative cooler, increases with increasing regeneration temperature for all process air inlet conditions to the system.

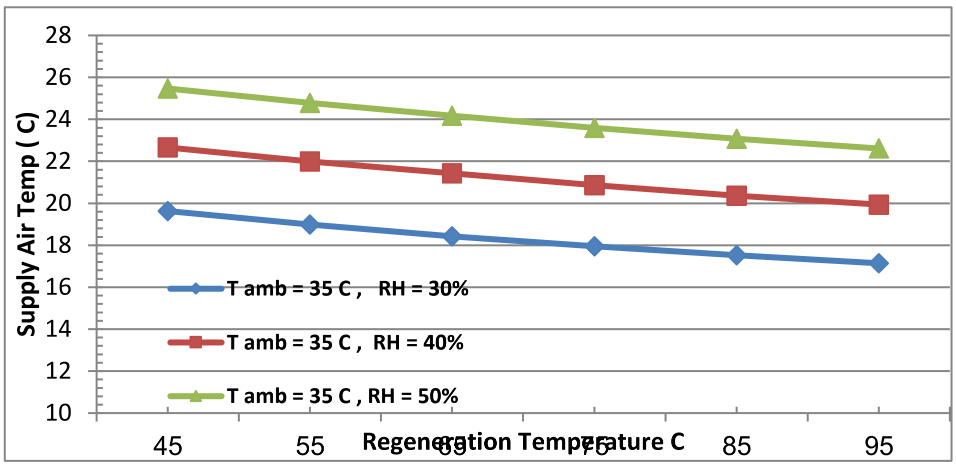

Fig (5) : Variation of supply air temperature with regeneration air temperature for different ambient relative humidity

The variation of supply air temperature with heat exchanger effectiveness is shown in Fig. (6). In this figure the evaporative cooler effectiveness was set equal to its reference value while the regeneration temperature was set equal to 65°C. It was found that as the heat exchanger effectiveness increases from 0.6 to 0.9, the supply air temperature decreases by about 11%, 17% for the system with ambient relative humidity of 50% and 30%, respectively. This indicates an improvement in the system performance. This improvement is attributed to the fact that as the heat exchanger effectiveness increases, both dry and wet bulb temperatures of the process air stream entering the evaporative cooler become lower, thereby the outlet dry bulb temperature of the supply air stream would be lowered further. Also The effect of evaporative cooler effectiveness on the supply air temperature is approximately similar to the effect of heat exchanger.

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It should be pointed here that the supply air temperature of the system is also affected by the variation of the inlet cooling water temperature to the heat exchanger. This temperature was taken 3°C higher than the wet bulb temperature of the ambient air. Of course, when the dry bulb temperature and the relative humidity of the ambient air increase, the wet bulb temperature of the process air at the inlet to the evaporative cooler will increase causing a higher supply air temperature by the system.

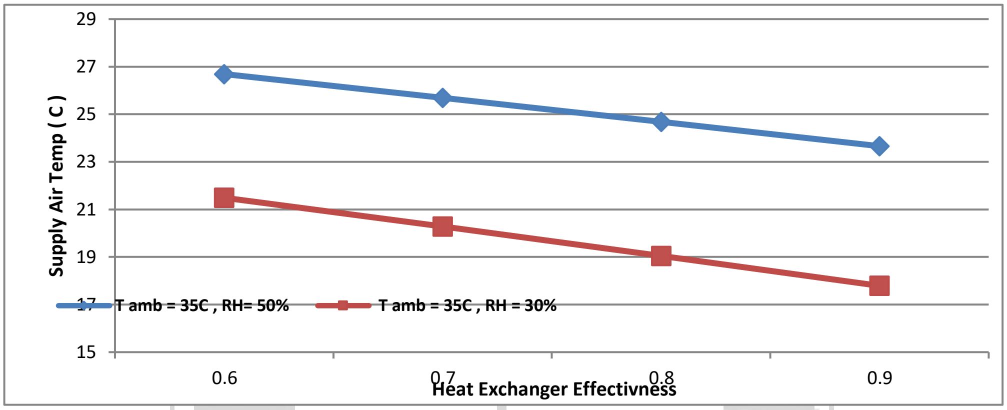

Fig (6) : Variation of supply air temperature with Heat Exchanger effectiveness

The other important parameter in the evaluation of a desiccant cooling system is the sensible cooling capacity. Fig. (7) and Fig (8) shows effect of ambient relative humidity on the sensible cooling capacity of the simulated system for different ambient air temperatures and process air mass flow rates, respectively . The relative humidity of the ambient air increases when the wet bulb temperature takes a higher value. This higher value is carried over through the system components to the evaporative cooler where its cooling effect is reduced due to a lower wet bulb depression. This cause a reduction in the sensible cooling capacity of the system due to an increase in the supply air temperature. However, it should be mentioned that although the cooling capacity decreases dramatically at higher ambient wet bulb temperatures, the system can provide supply air temperatures much lower than those obtained from an evaporative cooler alone. Also it was found that increasing process air mass flow rate increases the sensible cooling capacity.

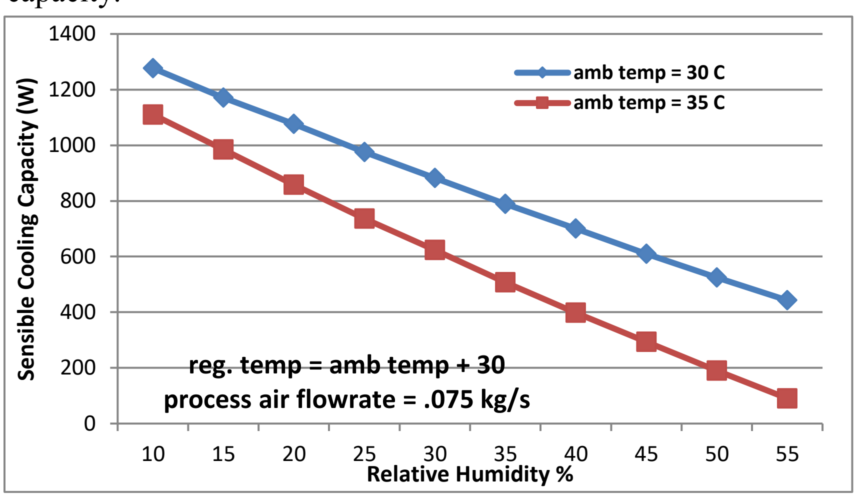

Fig (7) : Variation of sensible cooling capacity with ambient relative humidity for different ambient temperatures

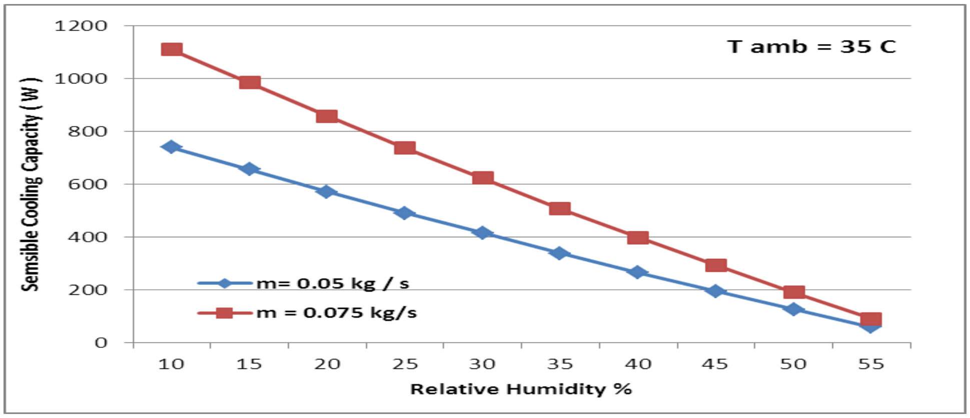

Fig (8) : Variation of sensible cooling capacity with ambient relative humidity for different process air mass flow rates

The computer simulation shows that the cooling capacity of the simulated system is greatly affected by the performance of the heat exchanger and evaporative cooler and also by the regeneration temperature, while the effect of dehumidifier efficiencies is rather small except at high regeneration temperatures. However, it should be pointed out that a regeneration air temperature above 70 °C is very difficult to obtain from an actual solar air heater with acceptable collection efficiency under the local weather conditions. To obtain regeneration temperatures above 70 °C an auxiliary heat source must be used which makes the operation of the system uneconomical. At a limit of 70 °C regeneration temperatures, the effect of the dehumidifier becomes less pronounced than that of the heat exchanger and the evaporative cooler on the sensible cooling capacity of the system.

The desiccant cooling system investigated in the present work is classified as a refrigeration system since it provides a lower temperature than that of the surrounding and it's effectiveness for cooling may be expressed by the coefficient of performance "COP". The coefficient of performance was estimated by dividing the refrigeration effect of the system by the total electric power consumed in a similar actual system under the same operating conditions. The refrigeration effect of the system was obtained by multiplying the mass flow rate of the process air by the enthalpy difference between the inlet and outlet states of the cycle as given by equation (27). The refrigeration effect is the ability of the system to cool ambient air from its initial state to the final supply state. Electric power consumed in the system is the sum of the power used to rotate the dehumidifier bed and the power used to circulate water, process and regenerating air streams. The power consumed in air circulation depends on the mass flow rates of the process and regenerating air streams was found to be 500 and 375 W for mass flow rates of the process air stream of 0.075 and 0.05 kg/s, respectively. These values are adopted in the present work to evaluate the COP of the simulated system. Also, the ambient air was assumed to be heated to the regeneration temperature using solar energy only with no other auxiliary heat whatsoever.

The results discussed aforehand show that there are several factors that affect the coefficient of performance. These factors include ambient conditions, supply air conditions, process air flow rate, components performance and the total electric power input to the system. The effect of ambient relative humidity on the COP is shown in Fig. (9). In this figure the performance of the individual system components were set at their reference values. The results indicate that the COP increases with increasing the relative humidity of the inlet process air stream to a certain extent and then starts to decrease with further increase of relative humidity. This decrease in the COP is mainly due to the combined effect of the higher inlet dry bulb temperature and moisture content of the process air stream as well as the moderate regeneration temperature. However, this decrease is rather small in the ranges of temperature and relative humidity considered. It should be noted that the factor that determines the COP for different mass flow rates is essentially the power consumed to circulate the air in the cycle.

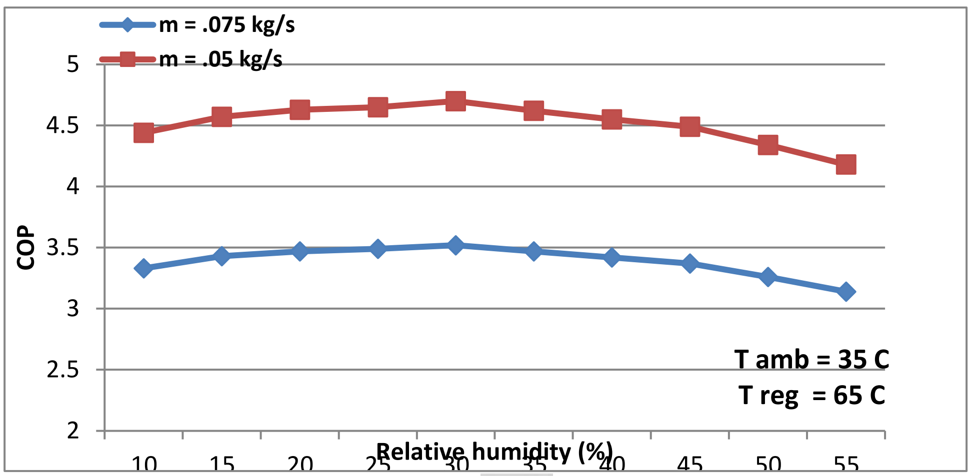

Fig (9) : Variation of the coefficient of performance with ambient relative humidity for different mass flow rates

In addition to the earlier discussed results, the simulated open - cycle desiccant-evaporative cooling system was evaluated for clear summer days in Benghazi using actual weather data. Table (1) shows the results of the computer simulation for a typical three different typical summer days. The mass flow rate of the process and regenerating air streams were set equal to 0.075 and 0.06 kg/s, respectively, while the component performances were set equal to their reference values. The variation in the supply air temperature during the day is rather small due to the combined effect of the variation of ambient temperature, relative humidity and regeneration air temperature during the day.

The average system supply air temperature during the simulated period for the three selected days were 20.8 , 22.4 and 15.8°C, respectively. The variation in the supply air temperature for these selected three days is mainly due to the variation of ambient conditions. The above days are typical for summer in Benghazi.

Table(1) also shows the hourly values of the sensible cooling capacity of the system for the selected three days . The room design conditions are 26.7°C d.b. and 50% relative humidity. It should be mentioned that for a mass flow rate of process air of 0.075 kg/s, a change in the supply air temperature

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by only 1°C results in a change in the sensible cooling capacity of approximately 75 W. For a higher flow rate this effect becomes very clear. Thus, constant or near constant values of the sensible cooling capacity of the system are difficult to obtain due to the variation of ambient conditions and regeneration temperature during the day. However, the combined effects of these factors may result in a near constant value of the sensible cooling capacity for some hours.

Table (1) : Simulation results of the system for a typical summer using actual weather data with reference input values

a)

HOUR	Ambient Temp. (C)	Relative Humidity (%)	Supply Air Temp (C)	Regeneration Temp (C)	Cooling Capacity (W)	COP
8.0	25.0	53.0	17.79	30.9	671.4	1.14
9.0	28.0	50.0	19.31	38.8	556.9	1.72
10.0	31.0	47.0	20.88	45.6	438.9	2.25
11.0	34.0	43.0	22.19	51.3	340.1	2.75
12.0	35.0	40.0	22.06	53.7	349.5	3.03
13.0	36.0	37.0	22.00	55.2	354.4	3.18
14.0	36.0	35.0	21.40	54.2	399.6	3.20
15.0	35.0	35.0	20.75	45.6	448.3	2.97
16.0	33.0	40.0	20.87	39.2	439.1	2.40
17.0	31.0	45.0	20.83	35.4	442.2	1.81

b)

HOUR	Ambient Temp. (C)	Relative Humidity (%)	Supply Air Temp (C)	Regeneration Temp (C)	Cooling Capacity (W)	COP
8.0	27.0	73.0	23.32	32.9	254.5	0.66
9.0	28.0	68.0	22.96	38.8	281.8	1.28
10.0	29.0	63.0	22.64	43.7	306.1	1.77
11.0	30.0	58.0	22.30	47.3	331.7	2.18
12.0	31.0	54.0	22.24	49.8	336.3	2.42
13.0	31.0	54.0	22.19	50.2	339.7	2.46
14.0	31.0	50.0	21.31	49.3	406.0	2.49
15.0	31.0	51.0	21.72	47.0	375.2	2.27
16.0	31.0	54.0	22.64	43.6	306.1	1.93
17.0	30.0	58.0	22.97	32.9	280.9	1.36

c)

HOUR	Ambient Temp. (C)	Relative Humidity (%)	Supply Air Temp (C)	Regeneration Temp (C)	Cooling Capacity (W)	COP
8.0	28.0	33.0	15.75	30.8	824.8	1.79
9.0	30.0	27.0	15.28	40.8	860.4	2.34
10.0	32.0	24.0	15.49	46.6	844.3	2.79
11.0	33.0	20.0	14.67	50.3	906.3	3.12

12.0	34.0	19.0	14.89	52.7	889.9	3.29
13.0	35.0	19.0	15.50	54.2	844.0	3.39
14.0	35.0	19.0	15.54	53.2	841.1	3.35
15.0	35.0	21.0	16.29	51.0	784.3	3.25
16.0	34.0	24.0	16.86	46.6	741.0	2.90
17.0	34.0	28.0	18.42	30.8	623.8	2.51

This computer simulation reveals that the simulated system can perform effectively and is able to provide a cool supply of air at satisfactory conditions, using solar energy only for regenerating the desiccant bed, for most hours of typical clear summer days under the local weather conditions of Benghazi or any other similar weather cites in Libya.

5. Conclusions

An open-cycle solid-desiccant evaporative cooling system using solar energy as a regeneration heat source was simulated using the developed computer simulation technique. System performance characteristics were evaluated on hourly basis for the period from 8 a.m. to 5 p.m. during typical clear summer days in Benghazi. From the results of the performance simulation of the complete desiccant cooling system, the following points can be concluded:

1- The performance of the system is greatly affected by the performance of the heat exchanger and the evaporative cooler and also by the regeneration temperature, while the effect of dehumidifier performance is rather small except at the higher regeneration temperatures.

2- The system can provide cool air at temperatures much lower than those obtained from a conventional evaporative cooler for the range of ambient conditions considered.

3- Increasing the ambient relative humidity and dry bulb temperature has a pronounced negative effect on the supply air temperature and the sensible cooling capacity. Whereas, it increases the coefficient of performance of the system. Moreover, The COP slightly decreases at higher relative humidity.

4- The simulated desiccant cooling system can supply cool air at satisfactory comfort conditions, using solar energy only for regenerating the desiccant bed, for most hours of typical clear summer days under the local weather conditions of Benghazi.

5- The variation of supply air temperature during the day is rather small due to the combined effect of ambient conditions and regeneration temperature. An average value for the supply air temperature of 18 °C was obtained during the simulated period for selected summer days in Benghazi.

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Nomenclature

	Symbols	Units
A	Apparent extraterrestrial solar intensity	$[W / m^2]$
B	Atmospheric extinction coefficient	[(air mass) $^{-1}$]
C	Diffuse radiation factor	-
Cp	Specific heat of air	[J / kg . K]
CC	Sensible cooling capacity	[W]
COP	Coefficient of performance	-
D_f	Collector dust factor	-
D_h	Hydraulic diameter of air passage	[m]
EP	Electric power input to system	[kW]
F′	Collector efficiency factor	-
F_i	Combined heat & mass potential, i = 1, 2	[dimensions arbitrary]
F_R	Collector heat removal factor	-
g	Acceleration due to gravity (9.81)	$[m / s^2]$
G	Air flow rate per unit of collector area (mass flux)	$[kg / s\, m^2]$
Gr_L	$\frac{g\beta_T\left(T_p - T_g\right) L^3}{\nu^2}$ = Grashof number based on L	-
h	Enthalpy of air	[kJ / kg]
h_c	Convective heat transfer coefficient	$[W / m^2 . K]$
h_r	Radiative heat transfer coefficient	$[W / m^2 . K]$
h_w	Wind heat transfer coefficient	$[W / m^2 . K]$
H	Solar irradiance on a horizontal surface	$[W / m^2]$
I	Solar irradiance on a tilted surface	$[W / m^2]$
I_{DN}	Direct normal irradiance	$[W / m^2]$
K	Air thermal conductivity	$[W/m^2.C]$
L	Length of collector	[m]
N	Number of day in the year	-
Nu_L	$\frac{h_c L}{K}$ = Nusselt number based on L.	-

Nu_{D_h}	$\frac{h_c\ D_h}{K}$ = Nusselt number based on D_h	-
Pr	$\frac{\mu\ Cp}{K}$ = Prandtle number	
q_u	Useful energy gained per unit collector area	[W / m^2]
r_1 , r_2	Parallel and perpendicular reflection coefficients .	-
Re_{D_h}	$\frac{\rho_a\ u\ D_h}{\mu}$ = Reynolds number based on D_h	-
R	Thermal resistance	[m^2 . K / W]
RE	Refrigerating effect	[kW]
S	Absorbed solar radiation	[W / m^2]
S_f	Collector shade factor	-
T	Temperature	[K]
u	Velocity of air in the collector	[m / s]
U	Heat loss factor	[W/ m^2 .K]
V	Wind speed	[m / s]
W	Moisture content of air	[kg/kg dry air]

Greek Symbols

	Symbols	Units
α	Absorptance	-
β	Collector tilt angle with horizon	[degree]
β_T	Coefficient of volumetric expansion	[K^{-1}]
γ	Angle of corrugation	[degree]
η	Efficiency	-
η_{ev}	Evaporative cooler effectiveness	-
η_{Fi}	F_i efficiency of dehumidifier , i = 1 , 2	-
η_{hx}	Heat exchanger effectiveness	-
θ_i	Solar radiation incident angle	[degree]
μ	Dynamic viscosity of air	[Pa . s]
ν	Kinematic viscosity of air	[m^2 / s]
ρ_a	Density of air	[kg / m^3]
τ	Transmittance	-
(τ_1,τ_2)	Parallel and perpendicular transmittance components	-
($\tau\alpha$)	Transmittance - absorbtance product	-

Subscripts

a	Ambient .
b	Beam .
c	Collector .
d	Diffuse.
db	Dry bulb.
e	Effective.
f	Fluid .
g	Glass.
gb	Global .
i	Inlet to a component.
in	Inlet to collector .
L	Overall .
m	Mean .
N	State number , 1,2,...., 5 .
o	Outlet from a component .
out	Outlet from collector .
p	Absorber plate .
P	Process air stream .
r	Reflected .
reg	Regeneration air stream.
R	Room state .
S	Supply state.
t	Top .
w	Wood frame of collector, water .
wb	Wet bulb.