PERFORMANCE SIMULATION OF SOLAR HYBRID LIQUID DESICCANT COOLING FOR VENTILATION AIR PRECONDITIONING

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ABSTRACT

The performance of solar hybrid liquid desiccant cooling for ventilation air preconditioning has been simulated for the month of August in Miami, Florida. In the system analyzed, 95 % by weight triethylene glycol was used as the desiccant. The air dehumidifier and the desiccant regenerator consisted of packed bed absorption towers operating at high liquid flow rates. The heat required for desiccant regeneration was provided by a solar collector/storage subsystem. Performance of the desiccant system was analyzed as a function of system design parameters such as the desiccant storage volume, the regenerator size, the hot water storage volume, and the solar collector area. The chiller electrical energy requirement, the regeneration auxiliary energy demand, and the solar fraction for regeneration were evaluated as functions of the variables listed above. The simulation revealed that by using solar hybrid liquid desiccant cooling for ventilation air preconditioning in a hot and humid climate, as much as 80 % electrical energy can be saved compared to a conventional vapor compression system. However, if electrical energy is to be used as auxiliary energy for the desiccant regeneration, a large solar fraction for regeneration (> 0.86) is needed in order to save electrical energy compared to a conventional system. A configuration where the system operated at a low desiccant temperature and concentration (20 °C and 90 % by weight) was also considered. In this configuration, it would be required to cool the desiccant in a chiller, increasing the chiller load as compared to a conventional system.

NOMENCLATURE

COP	coefficient of performance
EER	energy efficiency ratio (ratio of cooling in Btu/h to the electrical power input in W)
H	enthalpy (kJ/kg)
I_T	solar radiation incident on a tilted surface (MJ/m^2-DAY)
Q	energy requirement (GJ/DAY, MJ/DAY, or GJ/month)
Y	air humidity ratio (kg water/kg dry air or g water/kg dry air)
ϵ_H	enthalpy effectiveness
ϵ_Y	dehumidification effectiveness

Subscripts

a	air
equ	equilibrium
IN	inlet
OUT	outlet

INTRODUCTION

Conditioning of ventilation air in a hot and humid climate is an energy intensive process. Since ventilation air must be adequately dried for building humidity control, the ratio of latent to total cooling load is large. A number of strategies are possible for ventilation air conditioning, as exemplified by Rengarajan et al. (1996). In this study, the increased ventilation requirement due to ASHRAE Standard 62-1989 was found to increase the annual energy requirement and operating cost by 10-15 % for a large office building in Miami, Florida. Through mathematical modeling, the authors found that the conventional vapor compression system was unable to meet the increased latent cooling load, with the result that the indoor relative humidity frequently exceeded 60 %. Pretreating the outside air with a 100 % outside air DX (direct expansion) unit or a gas fired desiccant unit maintained the indoor relative humidity below 60 % for a larger part of the time (95 %), as compared to pretreatment using a heat pipe assisted water coil (90 % of the time) and an enthalpy recovery wheel (93 to 95 % of the time). Also, the desiccant system was found to reduce the annual electric energy use significantly. Other researchers have demonstrated the viability of

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desiccant systems for ventilation air conditioning. For example, Meckler (1995) showed that the installed chiller capacity could be reduced by 30 % by using the desiccant preconditioning unit. Thornbloom and Nimmo (1995) compared a solar liquid desiccant dehumidification system to a conventional vapor compression system for treating the ventilation air required for a supermarket in Miami, Florida. In their system, calcium chloride was used as the desiccant and it was regenerated in a trickle solar collector regenerator. A packed bed dehumidifier handled the latent cooling and a vapor compression unit handled the sensible cooling. A cost analysis showed that the annual operating cost of the desiccant system was significantly lower than for a conventional system. In another recent study, Spears and Judge (1997) presented results from a one year evaluation of a gas-fired desiccant ventilation air conditioner for a Wal-Mart super center. The control of the indoor humidity was significantly better in the store that used the desiccant system as compared to a store using standard air conditioning. Besides the benefit of improved comfort, the store using the desiccant system saved 13 % energy compared to the control store.

With such promising results, desiccant cooling is of great interest for the application of ventilation air preconditioning. Of the two basic types of desiccants, liquid desiccants offer some advantages over solid desiccant systems: according to Howell (1987) the pressure drop through a liquid desiccant system is smaller than the pressure drop through a solid desiccant wheel; the ability to pump the liquid makes it possible to connect several small dehumidifiers to one large regeneration unit (Harriman, 1990), which may be advantageous in large buildings; and concentrated desiccant may be stored for use during the times when no suitable source of regeneration heat is available. This paper gives the results from a system performance simulation of solar hybrid liquid desiccant ventilation air conditioning for the month of August in Miami, Florida. A desiccant dehumidification system handles the latent cooling load, while a conventional chiller is used to sensibly cool the air. The system examined in this study uses triethylene glycol as a desiccant and packed bed absorbers as the dehumidifier and the desiccant regenerator. Solar heat is provided indirectly through a solar collector/storage system. Findings from a previously conducted experimental and theoretical study of the performance of the packed bed dehumidifier/regenerator (Goswami and Öberg, 1997) were used as the basis for the design of these components. For example, high liquid flow rates are used in the dehumidifier and regenerator to ensure wetting of the packing. The present investigation focuses on the influence of system design parameters such as the desiccant storage volume, the regenerator size, the hot water storage volume, and the solar collector area. Insight into the design of solar hybrid desiccant systems is provided through an evaluation of the chiller electrical energy requirement, the regeneration auxiliary energy demand, and the solar fraction for regeneration, as functions of the parameters listed above.

SIMULATION MODEL DESCRIPTION

This study simulates the preconditioning of 0.5 m ^{3}/s (1000 cfm) ventilation air using a solar hybrid liquid desiccant system (Figure 1), and compares the electrical energy requirement for this process to that of a conventional system. The air is assumed to be cooled and dehumidified from the ambient conditions to 24 ° C and 50 % relative humidity, corresponding to a humidity ratio Y 9.5 g/kg. In the desiccant system, the air is dried in a packed bed dehumidifier before it is sensibly cooled by the chiller. Before the desiccant (95 % by weight triethylene glycol) enters the dehumidifier, it is cooled by exchanging heat with water from a cooling tower. Desiccant storage provides a buffer so that the desiccant can be regenerated during the hours of the day when solar energy is available. In this study, it was assumed that the regenerator operates at a constant desiccant flow rate between 10 AM and 7 PM solar time, and that no regeneration takes place during the rest of the day. For regeneration, the desiccant is heated and brought into contact with a moisture scavenging air stream in a packed bed tower. The temperature of the desiccant entering the regenerator is set at 65 °C, with regeneration heat provided by a flat plate solar collector and hot water storage subsystem, and by an auxiliary source if needed.

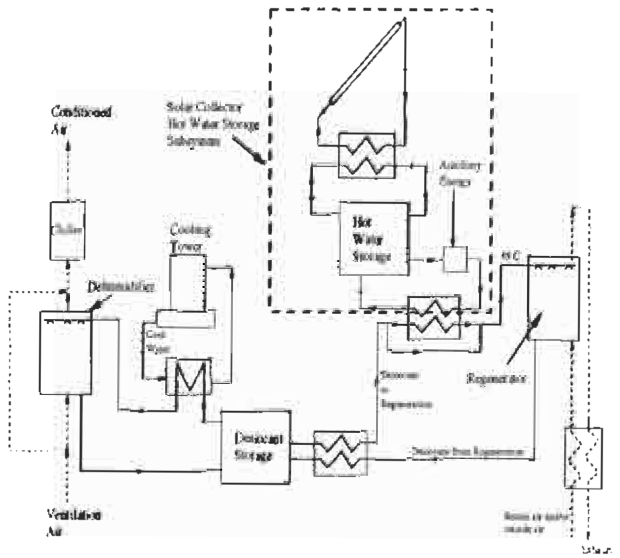


Figure 1. Solar Hybrid Liquid Desiccant Cooling System for Ventilation air Preconditioning

The performance simulation was carried out in three steps. Initially, hourly weather data for the month of August in Miami, Florida, was obtained using a weather generating subroutine available in the simulation program TRNSYS (Klein et al., 1990). Next, the performance of the system, excluding the solar subsystem, was modeled by carrying out mass and energy balances on each component. This analysis was conducted using a FORTRAN computer program. Hourly chiller loads and regeneration energy requirement for the desiccant system were calculated as a function of the desiccant storage volume and the desiccant flow rate to the regenerator during daytime hours. This flow rate influences the size of the regenerator part of the system, including the size of the solar subsystem. The electrical energy requirements to meet the chiller loads were found by dividing the cooling loads by the coefficient of performance (COP). In the desiccant system, the chiller mostly handles the sensible cooling load. Thus, it should be noted that since the air does not have to be cooled below its dew point to condense moisture, the chiller may be able to operate at a higher evaporator temperature compared to that in a conventional system. Therefore, the COP for the chiller in the desiccant system may be higher than the COP for a conventional chiller. Nevertheless, a constant COP of 2.9 (corresponding to an EER=10) was assumed for both chillers in this study. Finally, using the hourly regeneration heat requirement obtained from the simulation described above, the solar hot water storage subsystem was modeled separately using TRNSYS (Klein et al., 1990). Results from this part of the simulation included the monthly solar fraction for regeneration (i.e., the part of the regeneration heat provided by solar energy), and auxiliary energy requirement as a function of the hot water storage volume and the solar collector area. Note that the source of auxiliary energy is not specified. Also, the auxiliary heat only comes on if the temperature in the tank is too low to heat the desiccant to 65 °C. For this simulation, the solar collectors were at a tilt angle of 20 °, facing south.

Components in the desiccant system were described using algebraic equations representing energy and mass balances, with certain simplifying assumptions described below. The cooling tower was modeled by using a linear relationship between the temperature of the water leaving the cooling tower and the ambient wet bulb temperature. This relationship was obtained from a curve fit of cooling tower performance data given by ASHRAE (1992). The performance of the packed bed dehumidifier and regenerator was modeled using the following relationships: a dehumidification effectiveness, ϵ_Y (equation 1), defined as the actual change in humidity ratio across the packed bed divided by the maximum possible change; and an enthalpy effectiveness, ϵ_H (equation 2), defined as the actual change in air enthalpy across the packed bed divided by the maximum possible change.

$$\epsilon_Y = \frac{Y_{IN} - Y_{OUT}}{Y_{IN} - Y_{equ}} \tag{1}$$

$$\epsilon_H = \frac{H_{a,IN} - H_{a,OUT}}{H_{a,IN} - H_{equ}} \tag{2}$$

Here, subscript equ refers to the value of the humidity ratio and enthalpy of air in equilibrium with the desiccant at the local desiccant temperature and concentration. Both the dehumidification effectiveness and the enthalpy effectiveness were assumed to be constant at 0.8. Findings from experimental and theoretical modeling of the packed bed absorber/regenerator have shown that this is a conservative assumption, as the values may be as high as 0.9 (Goswami and Öberg, 1997). In keeping with the operating ranges specified by Goswami and Öberg (1997), the liquid to air mass flow ratios in the dehumidifier and the regenerator were set at 4.5 and 3.75, respectively. It should also be noted that only the amount of air necessary to meet the load is passed through the dehumidifier. That is, if the conditions of the air and the desiccant entering are such that the humidity ratio of the air leaving the dehumidifier will be lower than 9.5 g/kg, some of the air is bypassed so that the humidity ratio at the mixing point following the dehumidifier is 9.5 g/kg. The desiccant flow rate through the dehumidifier is then adjusted to maintain the same liquid to air mass flow ratio. Furthermore, if the dehumidifier cannot meet 100 % of the latent load, the remaining latent cooling requirement is imposed on the chiller. The effectiveness of the liquid-to-liquid heat exchangers was assumed to be 0.8, and the effectiveness of the air-to-air heat exchanger in the regenerator section was assumed to be 0.6.

In the system simulation, desiccant storage volumes between 2.5 m^3 and 10 m^3 were considered, and the desiccant flow rate to the regenerator was varied between 7650 kg/hr and 12000 kg/hr. The performance of the solar system was modeled using the output from the system simulation for the case with 5 m^3 desiccant storage and the desiccant flow rate to the regenerator equal to 7650 kg/hr. With this flow rate, and assuming that the amount of air to be exhausted from the building equals the amount of ventilation air, dry return air can be used in the regenerator, which gives the desired desiccant to air flow ratio 3.75. In the solar subsystem simulation, the solar collector area was varied between 200 m^2 and 600 m^2, and the hot water storage volume was varied between 5 m^3 and 15 m^3. Due to additional system components in the desiccant system compared to the conventional system some parasitic electrical energy will be required, e.g., pumping power for desiccant and water, and additional fan power due to increased system air pressure drop. For the present study, these parasitic energy requirements have been assumed negligible compared to the fan and chiller power requirement of a conventional system.

RESULTS AND DISCUSSION

Figure 2 shows the daily solar radiation incident on the solar collectors for the month of August in Miami, Florida. The daily cooling loads necessary to bring 0.5 m^3/s from the ambient conditions to 24 °C and 50 % relative humidity are plotted in Figure 3. It can be seen that the latent cooling load makes up a large part of the total cooling load. The system performance is presented below as a function of design parameters such as storage size and collector area.

Effect of Desiccant Storage Volume and Desiccant Regenerator Size

The driving force for the mass transfer process in the dehumidifier is the difference in the vapor pressures in the air and the desiccant. When the desiccant vapor pressure is lower than the vapor pressure in the air, water is absorbed from the air into the desiccant. A higher desiccant concentration and/or a lower desiccant temperature decreases its vapor pressure. In order to meet the dehumidification requirement of the ventilation air conditioning, the desiccant concentration and temperature entering the dehumidifier must be such that the equilibrium air humidity ratio at the top of the dehumidifier (Y_{equ}) is low enough so that the outlet air humidity ratio (Y_{OUT}) can meet the load. During the hours when the regenerator is not operating, the desiccant concentration in the storage will steadily decrease. At some point, it may be too low to satisfy the latent cooling requirement. The average concentration in the desiccant storage tank during a 24 hour period can be maintained at higher levels by increasing the storage volume, by regenerating for longer hours, and/or by increasing the desiccant flow rate to the regenerator during the hours when the regenerator is operating. For a given packed bed height, if the inlet conditions of the air and desiccant to the regenerator, and the liquid to air mass flow ratio are constant, the change in the desiccant concentration through the regenerator is constant regardless of the desiccant flow rate. Thus, by increasing the desiccant flow rate to the regenerator, more water is removed from the desiccant storage per unit time. However, increasing the desiccant flow rate requires a larger regenerator, making a larger solar subsystem necessary.

Figure 4 shows the monthly percent latent load met by the dehumidifier, as a function of the desiccant storage volume and the desiccant flow rate to the regenerator. Over 90 % of the latent load is met by the desiccant system for the entire range of operating conditions. The percent of the latent load met by the dehumidifier slightly increases with increasing desiccant flow rate and/or increasing desiccant storage volume. Figure 4 also shows the percent electrical energy saved at the

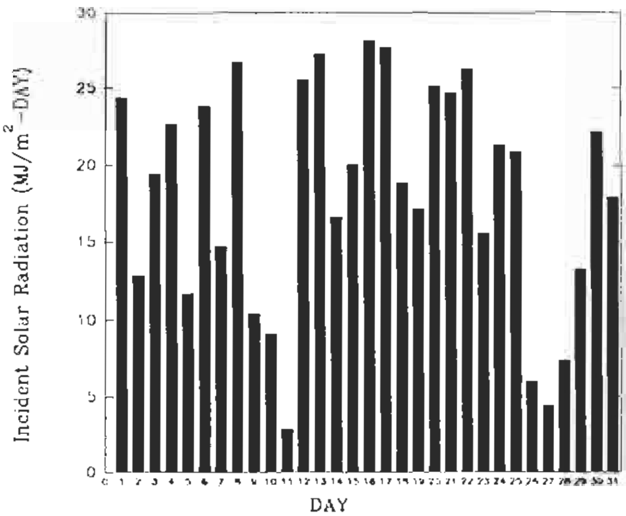


Figure 2. Daily Solar Radiation Incident on the Tilted Collector Surface for the Month of August in Miami, Florida

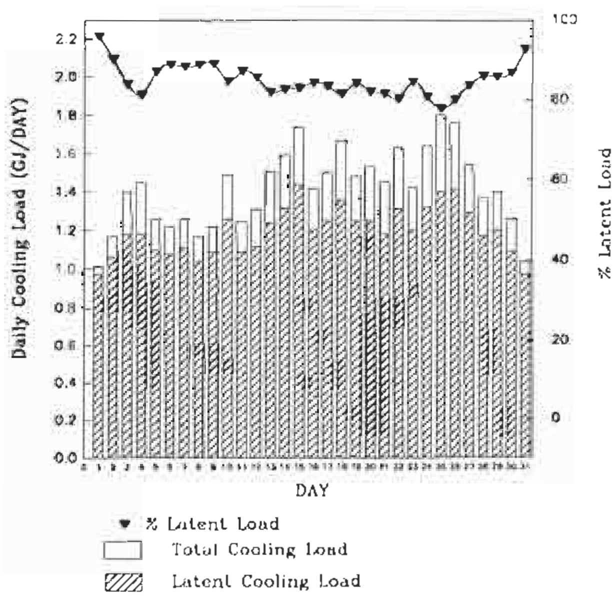


Figure 3. Daily Total and Latent Cooling Loads for Preconditioning of 0.5 m^3/s Ventilation Air for the Month of August in Miami, Florida

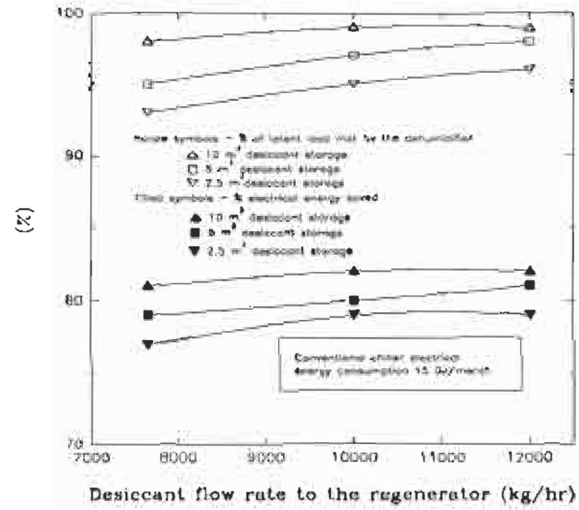


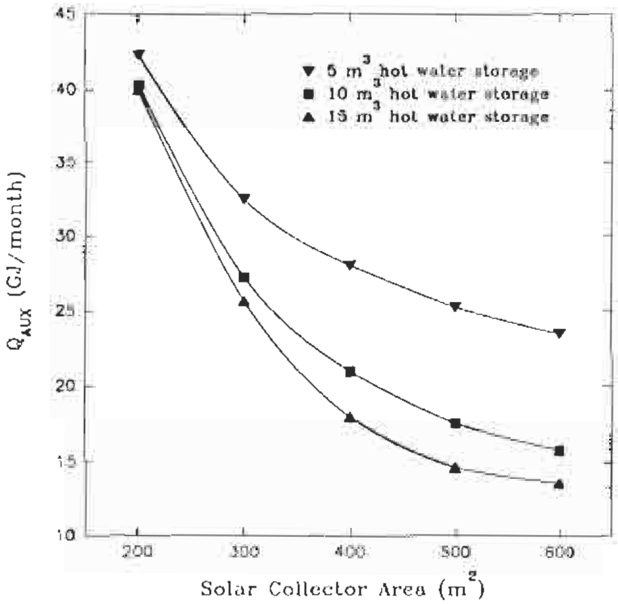
Figure 4. The Influence of Desiccant Storage Size and Regenerator Flow Rate on the Fraction of the Latent Load Handled by the Dehumidifier, and the Percent Electrical Energy Saved at the Chiller

chiller, which is only marginally influenced by the two parameters. In summary, Figure 4 illustrates that in order to handle 100 % of the latent load in the dehumidifier, the desiccant storage volume and the desiccant flow rate to the regenerator must be very large. However, the resulting additional electrical energy savings at the chiller may be insignificant, so it is more advisable to size the system at lower percentages.

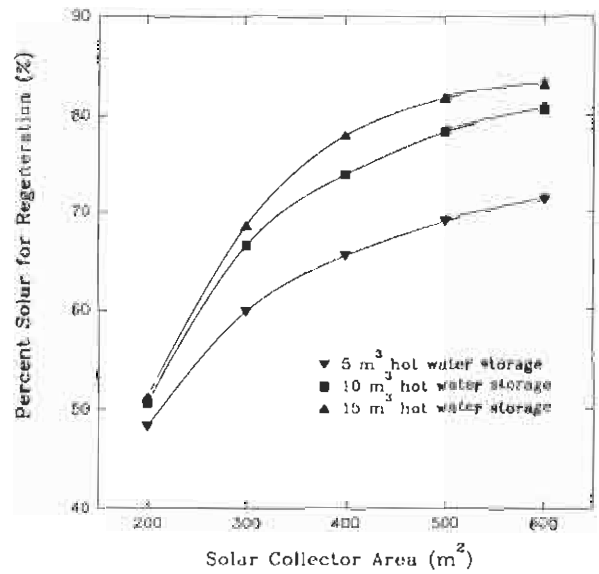
Effect of Hot Water Storage Volume and Solar Collector Area

The amount of auxiliary energy is a function of the amount of solar heat provided by the solar subsystem. Therefore, the monthly auxiliary energy requirement was determined as a function of the hot water storage size as well as the solar collector area, as shown in Figure 5a. For a given hot water storage volume, the auxiliary energy requirement decreases rapidly with increasing solar collector area until a collector area is reached where the slope of the curve flattens out. Figure 5b shows the daily auxiliary energy requirement for three days with varying cloudiness. For a very clear day (August 16) the auxiliary energy requirement is eliminated by using a collector area of 300 m^2, while for a very cloudy day (August 11) increasing the collector area has no effect on the auxiliary energy requirement. Figure 5a displays that a combination of large collector area and large hot water storage volume gives the largest reduction in auxiliary energy requirement. However, not even 600 m^2 collector area in combination with a 15 m^3 hot water storage eliminates the need for auxiliary energy to precondition 0.5 m^3/s (1000 cfm) ventilation air.

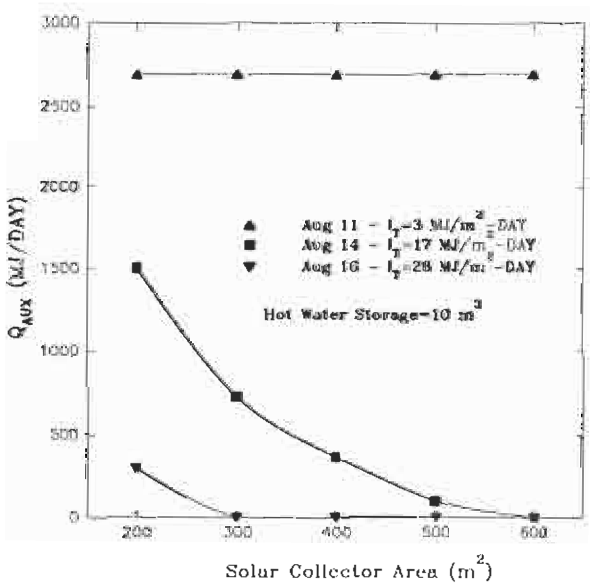
It is also of interest to know the fraction of the regeneration energy that can be provided by solar energy. Figure 6a shows the monthly percent solar energy for regeneration as a function of the hot water storage volume and the solar collector area. For a fixed solar collector area, increasing the hot water storage volume from 5 m^3 to 10 m^3 significantly increases the percent of the regeneration energy that is



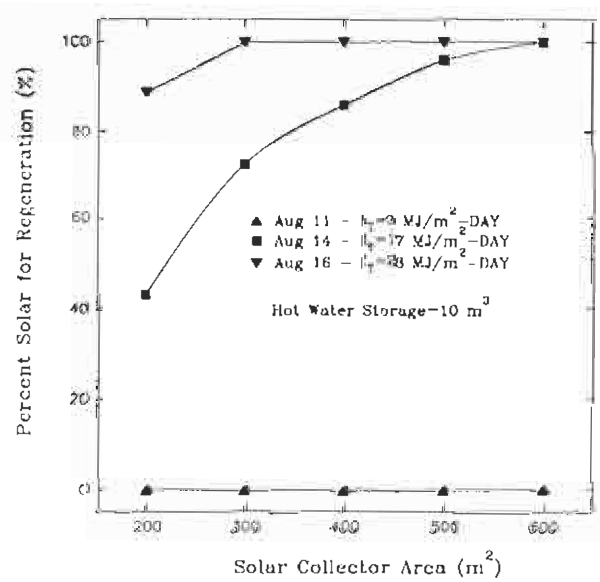
(a)



(a)



(b)



(b)

Figure 5. Auxiliary Energy Requirement versus Solar Collector Area: (a) Monthly Performance for Various Hot Water Storage Sizes; (b) Daily Performance for a Range of Climatic Conditions

Figure 6. Percent Solar for Regeneration versus Solar Collector Area: (a) Monthly Performance for various Hot Water Storage Sizes; (b) Daily Performance for a Range of Climatic Conditions

provided by solar energy. An additional increase of the storage volume from 10 m ^{3} to 15 m ^{3} does not have as large an effect. Also, for a given hot water storage volume, as the solar collector area increases the percent solar energy for regeneration increases. However, as the solar collector area becomes large the slope of this curve levels off. Figure 6b shows that for a very clear day, a 300 m ^{2} collector area makes it possible for the solar subsystem to provide all of the regeneration heat. However, on a very cloudy day the solar system does not provide any heat for regeneration regardless of collector area.

Figures 5 and 6 indicate that to design a system using 100 % solar energy for regeneration would require a very large solar collector area in combination with a very large hot water storage volume. Thus, it seems more likely that a system would use some auxiliary energy. Figure 7 shows that a linear relationship exists between the auxiliary energy requirement and the percent solar energy used for regeneration as obtained from all the simulations performed on the solar subsystem. The monthly electrical energy saving at the chiller as compared to a conventional system is also indicated in Figure 7. Note that the auxiliary energy requirement should not be larger than the electrical energy savings at the chiller if electric heaters are to be used to supply auxiliary heat. Thus, the minimum percent solar for using electrical auxiliary energy is found from the curve fit by setting the auxiliary energy requirement equal to the savings at the chiller. In this study, this minimum value was found to be about 86 %. Similarly high values (about 88 %) were obtained by Hernández et al. (1996) in their performance analysis of a solar-assisted hybrid liquid desiccant system combining a solar absorption chiller with a desiccant dehumidifier.

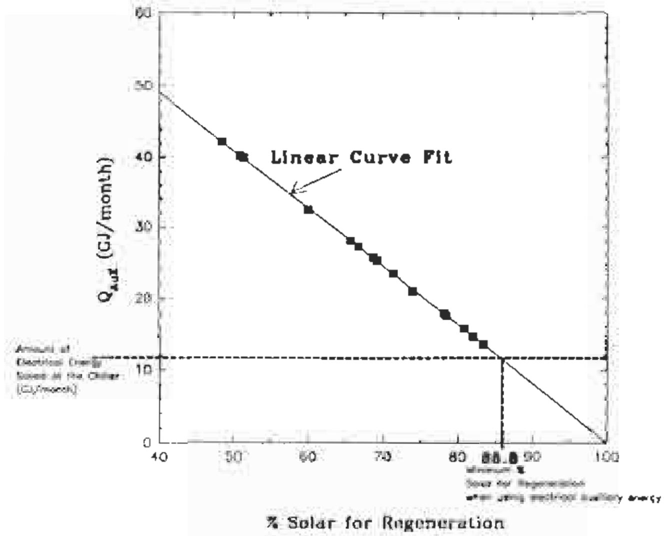


Figure 7. Auxiliary Energy Requirement versus Percent Solar for Regeneration

Low Temperature and Low Concentration Desiccant System

Operating the desiccant system at a temperature below 20 °C has been suggested as an alternative mode of operation. Sick et al. (1988) studied a liquid desiccant system in which the desiccant was cooled in a chiller before entering the dehumidifier. Thus, both latent and sensible cooling of ventilation air was obtained in the packed bed desiccant conditioner. Because of its simplicity, such a configuration appears attractive. Furthermore, when using triethylene glycol as the desiccant, lower temperatures will minimize the evaporation of glycol in the dehumidifier. Chung et al. (1995 and 1993) carried out experimental studies of the dehumidification of air in packed bed absorption towers using desiccant temperatures between 15 °C and 21 °C. In a humid climate, such low desiccant temperatures are not always possible to obtain by using cooling water from a cooling tower, so cooling using a chiller may be required.

To examine the use of a cool and more dilute desiccant in the solar hybrid desiccant system, a simulation was carried out for an average day in August in Miami Florida, using triethylene glycol at 20 °C. Because the desiccant is now cooler, a more dilute desiccant can be used while still maintaining the vapor pressure low enough to achieve dehumidification. The system layout was modified as shown in Figure 8. By using the cool desiccant, both the dehumidification and sensible cooling of the air took place in the dehumidifier. The chiller was placed between the desiccant storage tank and the dehumidifier. Overall, this layout decreases the number of system components, which could help in reducing the first cost of the system. Another benefit of operating at low desiccant temperature and concentration is that the desiccant temperature at the regenerator entrance is now 45 °C compared to 65 °C when using a higher desiccant concentration. A lower regeneration temperature results in a higher efficiency of the solar collectors.

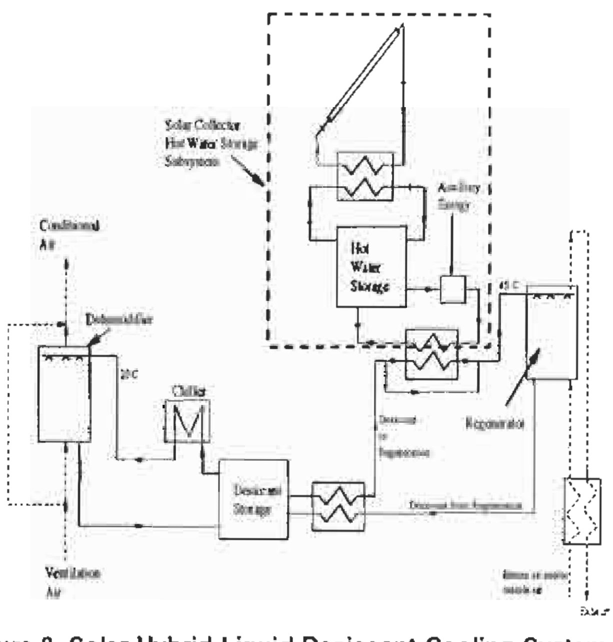


Figure 8. Solar Hybrid Liquid Desiccant Cooling System for Ventilation Air Preconditioning using a Cool, Dilute Desiccant

Results from the simulation of the low desiccant temperature system are shown in Figure 9. The hourly chiller load of this system is compared to that of a conventional vapor compression system. As shown, the desiccant system has a much higher chiller load compared to a conventional system. This is due to the parasitic heat added to the desiccant storage, especially during the hours of regeneration (10 AM to 7 PM). Configurations where a cooling tower was added between the desiccant storage and the chiller and between the regenerator side and the desiccant storage were also examined. No significant improvement of the desiccant system was obtained by these additions, which can be attributed to the high wet bulb temperatures in Miami. Therefore, despite the apparent benefits of using a cool dilute desiccant, it is not desirable from an energy point of view. These findings differ from the results presented by Sick et al. (1988), who showed the chiller load and the operating cost to be reduced for the solar desiccant system as compared to a conventional system. This may be explained by the lower ratio of desiccant flow rate to air flow rate in the dehumidifier as compared to the ratio in the present study. For a lower desiccant to air flow ratio, the amount of desiccant to be cooled per unit mass of air to be conditioned is lower. As previously mentioned, the higher desiccant flow rates employed in the present study were selected based on experiments conducted on a packed bed absorption tower (Goswami and Öberg, 1997). With a low flow rate, adequate wetting of the packing was not possible, and resulted in low effectiveness of the dehumidification and regeneration processes. Thus, in actual desiccant cooling systems, the need for relatively high desiccant to air flow ratios may severely penalize this particular configuration.

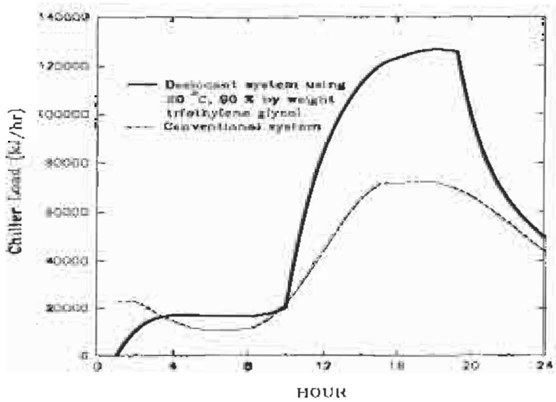


Figure 9. Hourly Chiller Load for a Conventional and a Low Temperature/Low Concentration Desiccant System

CONCLUDING REMARKS

By using solar hybrid liquid desiccant cooling for ventilation air preconditioning in a hot and humid climate, as much as 80 % electrical energy can be saved compared to a conventional vapor compression system. To save electrical energy using electric heaters as the source of auxiliary heat, the amount of auxiliary energy cannot exceed the amount of electrical energy saved at the chiller. Hence, a large solar fraction for regeneration (> 0.86) would be needed in order to save electrical energy compared to a conventional system. Because of cloudy days where no useful energy is provided by the solar system, very large collector area and hot water storage volume are required in order to obtain such high monthly solar fractions. Therefore, a seasonal performance analysis, exploring a number of sources for auxiliary heat is warranted at this stage. This should include a cost analysis, as well as an energy efficiency analysis starting at the primary energy source.

Since the simulation was conducted on a per air flow rate basis, the results presented in this paper can be scaled up or down depending on the flow rate needed. Some cases where desiccant ventilation air preconditioning may result in large annual electrical energy savings and improved indoor humidity control are laboratories, supermarkets, and health care facilities. Therefore, future studies of desiccant cooling for these applications are of great interest.

ACKNOWLEDGMENT

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