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# DESICCANT COOLING USING UNGLAZED TRANSPIRED SOLAR COLLECTORS

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## ABSTRACT

The use of unglazed transpired solar collectors for desiccant regeneration in a solid desiccant cooling cycle was investigated because these collectors are lower in cost than conventional glazed flat-plate collectors. Using computer models, the performance of a desiccant cooling ventilation cycle integrated with either unglazed transpired collectors or conventional glazed flat-plate collectors was obtained. We found that the thermal performance of the unglazed system was lower than the thermal performance of the glazed system because the unglazed system could not take advantage of the heat of adsorption released during the dehumidification process. For a 3-ton cooling system, although the area required for the unglazed collector was 69% more than that required for the glazed collector, the cost of the unglazed collector array was 44% less than the cost of the glazed collector array. The simple payback period of the unglazed system was half of the payback period of the glazed collector when compared to an equivalent gas-fired system. Although the use of unglazed transpired collectors makes economic sense, some practical considerations may limit their use in desiccant regeneration.

## 1. INTRODUCTION

Solar energy can be used to help offset the net cooling load of a building. Many system arrangements or cycles are possible: solar collectors (flat plate, parabolic trough, and evacuated tube) can be used to provide energy for absorption cooling, desiccant cooling, and even refrigeration cycles. Active combinations or solar hybrids coupled with heat-activated heat pumps or electricity-driven air conditioners are possible. The concept of solar cooling is appealing because the cooling load is roughly in phase with the solar energy availability. Furthermore, combined heating, hot water, and cooling season usage helps to decrease the amortization of the system. Although a large potential market exists for solar cooling, the existing solar cooling systems are not competitive with electricity-driven or gas-fired air-conditioning systems. The cost of solar cooling systems need to be reduced by lowering the cost of components and improving their performance. The reduction in required collector area and lower collector cost per area will reduce the cost of the solar components. The purpose of this study is to investigate the technical and economic feasibility of integrating low-cost unglazed transpired solar collectors with a desiccant cooling system.

We used the desiccant cooling ventilation cycle for this study. The desiccant cooling ventilation cycle uses a rotary desiccant dehumidifier, a heat exchanger, two evaporative coolers, a desiccant regeneration heater, and ancillary equipment such as fans and pumps (see Figure 1). In this cycle, outside air is dried in the dehumidifier and then cooled by regenerative evaporative coolers. The regeneration heater (powered by natural gas, waste heat, or solar energy) heats the air, which reactivates the desiccant by driving the moisture from it. This cycle is an alternative to vapor compression units that use chlorofluorocarbons and electricity. Gas-fired desiccant cooling systems have entered the market and, for some applications, may compete well with electricity-driven air conditioners. Currently, the capital cost of desiccant systems regenerated with conventional glazed solar collectors is too high to compete with gas-fired systems. If the capital cost of solar collectors is lowered, the environmental advantages and potential of peak-load reduction would make solar desiccant cooling systems an attractive option.

In recent years, unglazed transpired solar collectors have been used to preheat ventilation air (1,2,3). Because the collector is unglazed, its cost is estimated to be about one-third of the cost of a glazed collector (4). Using an unglazed transpired collector (UTC) for regeneration may be more economical; however, the lower efficiency of the UTC at needed regeneration temperatures must be considered when making this comparison. Studying this tradeoff was the motivation for this study. Using computer models, we investigated the performance of a desiccant cooling ventilation cycle with two types of air collectors: UTCs and flat-plate glazed collectors. The main difference between these two solar collectors is that the

UTC is glazed and transpired, i.e., it has no glass cover and the absorber plate has many small holes through which air is pulled with a fan. The glazed collector is a conventional flat-plate solar collector that heats air, and it has been studied by many investigators. Figures 1 and 2 show the schematic diagrams of the ventilation desiccant cooling cycle with the two different solar collectors for desiccant regeneration. This paper presents the results of our analytical study.

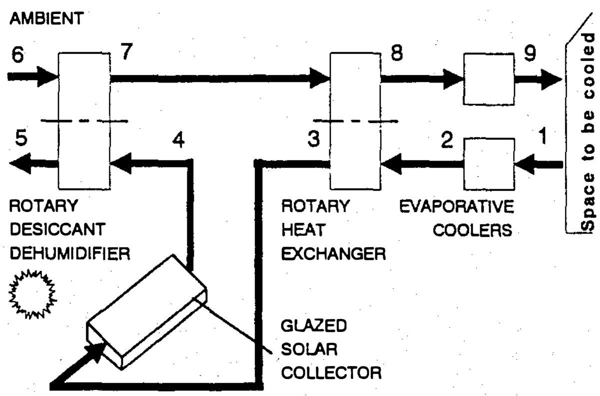


**Fig. 1 Desiccant Cooling System with Glazed Flat-Plate Collector**

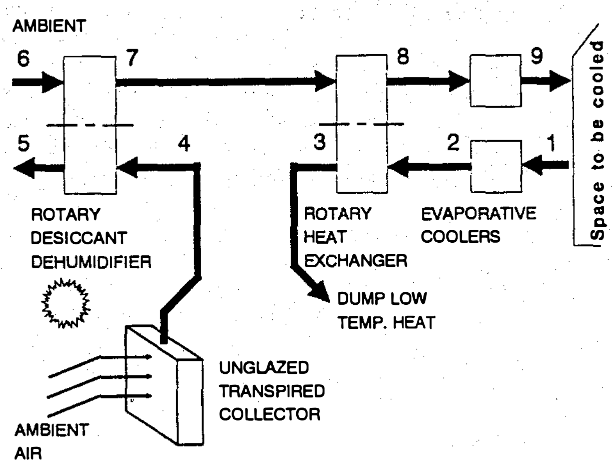


**Fig. 2 Desiccant Cooling System with Unglazed Transpired Collector**

## 2. ANALYSIS

### 2.1 Unglazed Transpired Collector

Unglazed transpired collectors can be used to heat ambient air in once-through solar energy systems. With this type of unglazed collector, air adjacent to the front surface of the absorber is drawn through the perforated absorber so that most of the heat that would otherwise be lost by convection from the absorber is captured by the air flow into the collector (see Figure 3). In windy conditions, only the energy in the thin thermal boundary layer is lost over the edge of the collector. The boundary layer is thin because of air transpiration suction. Kutscher et al. (1,2) have investigated heat losses from an unglazed transpired collector and predicted the efficiency of the collector under a variety of operating parameters. This type of design shows promise for application such as ventilation preheating and crop drying. A German patent (5) describes an unglazed perforated roof absorber for heating ventilation air. Schulz (6) describes a fabric absorber used in Germany for crop drying. A U.S./Canadian company is currently manufacturing and marketing unglazed perforated walls for ventilation preheating (3) and has patented the concept.

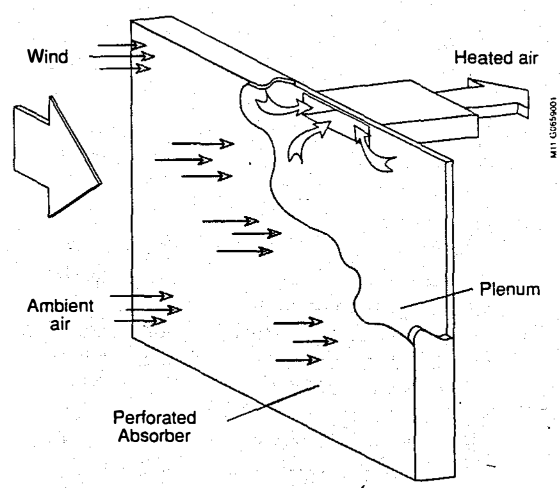


**Fig. 3 Unglazed Transpired Collector Oriented Vertically [Reprinted from Ref. (1)]**

Kutscher et al. (1,2) have developed a model to predict the performance of the unglazed transpired collector. Their model is based on an overall heat balance, incorporating estimated radiative and convective heat losses. The model predicts the collector efficiency, $\eta$, based on the solar insolation level, I; suction velocity, v; ambient air temperature, $T_{amb}$; and collector temperature, $T_{coll}$:

$$\eta = \rho\, c_p\, v\, (T_{coll} - T_{amb}) / I$$

The collector temperature is obtained from overall heat balance and depends on the wind speed, U; absorber surface emissivity, $\varepsilon$; collector absorptance, $\alpha$; and other mentioned parameters. We used their model, coded into a PC spreadsheet, to predict the outlet temperature and efficiency of the collector based on the collector area, incident sunlight, cross-wind speed, emissivity, ambient temperature, and collector suction velocity. The spreadsheet was slightly modified so that the heat output per unit area of the collector (= $\eta$ I) was calculated for use as regeneration heat input for analysis of the desiccant cooling system. This model is based on ambient air being pulled through the surface with the bottom of the plenum sealed. Kutscher (4) has suggested the possibility of opening up the bottom of the plenum and adding warm recirculated air at the bottom of the collector, but this has not been studied, and the effects are unknown.

## 2.2 Desiccant Cooling System

The performance of desiccant cooling cycles has been studied by many investigators. We used the Collier code (7). The code simulates the performance of rotary desiccant dehumidifiers, heat exchangers, and evaporative coolers. The dehumidifier model is based on a finite difference method solving simultaneous heat and moisture transfer in the dehumidifier. The effectiveness models are used to predict the performance of the heat exchanger and evaporative coolers. The code was slightly modified for the unglazed transpired collector. For the unglazed collector, the warm air leaving the heat exchanger (see point 3 in Figure 2) was exhausted to the surroundings. Rather than having this warm air continue on to point 4, the unglazed collector provided all of the regeneration heat, starting with the ambient temperature and humidity. We did not need to change the model for cases using the conventional flat-plate collector. The heat for regeneration of the desiccant for the unglazed transpired collector, (

$$Q_{regen,UTC} = \dot{m}\ C_p\ (T_4 - T_{amb})$$

where $T_4$ is regeneration temperature and $\dot{m}$ is the air flow rate through the dehumidifier. For the glazed collector, the heat for desiccant regeneration, $Q_{regen,GC}$, was calculated from

$$Q_{regen,GC} = \dot{m}\ C_p\ (T_4 - T_3)$$

where $T_3$ is the temperature of air leaving the heat exchanger.

It should be noted that moisture adsorption by the desiccant is an exothermic (heat releasing) process. Therefore, the temperature of the air leaving the dehumidifier on the supply side (point 2 on Figures 1 and 2) will increase. One purpose of the heat exchanger is to recover this heat to regenerate the desiccant and to cool the air before it goes into the evaporative cooler. The temperature $T_3$ is higher than $T_{amb}$ because of recovery of the heat of adsorption. As a result, the required external input heat for regeneration will be higher for the unglazed transpired collector than for the glazed collector. For a given cooling capacity, the thermal coefficient of performance (COP) of the unglazed transpired collector will, therefore, be lower than that of the glazed collector. The thermal COP is defined as *the cooling load removed* divided by *the regeneration heat input.* The energy required to run the fans and pumps is similar for the two systems and is small compared to the thermal energy input. Therefore, we did not consider them in our analysis.

Table 1 summarizes the characteristics and conditions of the system that is modeled. Although any desiccant material could be studied, we selected silica gel, a commonly used desiccant, for the purpose of this investigation. The physical dimensions of the studied dehumidifier are similar to those of a dehumidifier tested at NREL. The rotational speed of the dehumidifier affects the outlet air temperature and humidity from the dehumidifier and therefore affects the performance of the cooling system. For a given dehumidifier and operating conditions, the performance can be optimized by selecting an optimum rotational speed. It should be noted that the rotational speed is inversely proportional to the cycle time between adsorption and regeneration processes.

**TABLE 1**
**SYSTEM PARAMETERS AND CONDITIONS**

| | |
|---|---|
| **Dehumidifier** | Matrix Density: 157 kg desiccant/m$^3$<br>Matrix Heat Capacity: 1960 kJ/kg K<br>Total Frontal Area: 0.49 m$^2$<br>Matrix Depth: 0.2 m<br>Passage Hydraulic Diameter: 2.3 mm<br>Total Transfer Area: 95 m$^2$<br>Adsorption or Regeneration Air Flow Rate: 0.2 kg/s<br>Adsorption/Regeneration: balanced flow and balanced area<br>Number of Heat Transfer Units: 22.5<br>Ratio of moisture transfer to heat transfer resistances in desiccant: 1 |
| **Desiccant** | Silica gel (Davison, Grade 40) |
| **Regeneration** | 80°C, 70°C, or 60°C air temperature |
| **Outdoor Conditions** | (1 atm., 35°C, 0.014 kg moisture/kg ) |
| **Indoor Conditions** | (1 atm., 26.7°C, 0.011 kg moisture/kg air) |
| **Sensible Heat Exchanger** | Effectiveness of 0.93 |
| **Evaporative Coolers** | Effectiveness of 0.95 |

## 2.3 Integrated System

To analyze the performance of the integrated unglazed-collector/desiccant-system, both the unglazed transpired collector model and the desiccant system model were used simultaneously. The cooling capacity, required regeneration heat input, and thermal COP of the desiccant cooling system were obtained by using the regeneration temperature and conditions stated in Table 1. The outlet temperature and heat output from the unglazed collector were obtained by using suction velocity and other needed conditions. The results of the two models were matched by setting the outlet temperature from the unglazed transpired collector to be the same as the desiccant regeneration temperature. Then the collector area needed to meet the required regeneration heat for the design cooling capacity was calculated for a specific solar insolation level.

For the glazed collector, the required collector area was calculated in a similar manner using an efficiency-temperature model. The outlet air temperature from the glazed collector was set to be equal to the regeneration temperature. A glazed air collector was selected with the following efficiency-temperature relation:

$$\eta = 0.757 - 3.28\,(T_{out} - T_{amb})\,/\,I$$

## 3. RESULTS AND DISCUSSION

In this section we first present some results on unglazed transpired collector performance. Then we present the results of performance of the desiccant cooling ventilation cycle integrated with UTCs or glazed collectors.

### 3.1 Unglazed Transpired Collector

Before it was decided what specific conditions for the UTC would be used to calculate the required area and subsequent costs, extensive parametric runs were obtained for different levels of insolation, cross winds, emissivities, ambient temperatures, and temperatures required for regeneration. A few of these are presented here to show the performance changes of the UTC with operating and design conditions. Figure 4 shows the effect of various insolation levels at wind speeds of 0 and 5 m/s on the heat output of a UTC for an absorber with emissivity of 0.1, an absorptivity of 0.9, a size of 3m x 3m, and an ambient temperature of 30°C. Considering 3.5 m/s is a high wind speed, a wind speed of 5 m/s is probably a good limit for the practical wind speeds that need to be considered. Thus, for a given insolation level, the two curves showing 0 and 5 m/s are good boundaries for reasonable delivered temperatures and heat outputs from the UTC.

Similar to that of other solar collectors, the efficiency of the UTC ($\eta$ = heat output/insolation level) decreased with the increased collector (and delivered) temperature. There are two important observations to note on Figure 4: First, at low and moderate suction velocities the cross-wind velocity can have a significant effect on the output heat and delivered temperature of a collector of this size. For larger collector sizes the impact is less. The second is that this effect can be virtually eliminated by increasing the suction velocity to between 0.03 and 0.05 m/s assuming homogeneous suction velocity over

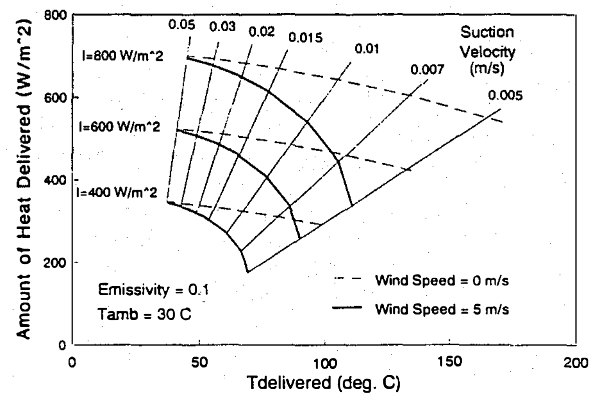


**Fig. 4 Effect of Insolation and Wind on Performance of Unglazed Transpired Collector**

the collector surface. Although increasing the suction velocity prevents the absorber's heat from blowing away, it also prevents the air from remaining in contact with the absorber long enough to reach very high temperatures. So there is a trade-off between obtaining high temperatures and reducing the effect of wind. The selection of suction velocity depends on what air temperatures should be delivered. It might seem desirable to use a suction velocity of 0.05 m/s, but the temperature of the air would only be raised 12°C or 13°C above ambient temperature, which is not adequate for regeneration of dessicant such as silica gel. Even without wind one should consider trade-off between delivery temperature and efficiency.

Figure 5 illustrates the effect of the ambient air temperature, the emissivity of the collector surface, and the suction velocity on the UTC performance. It can be observed that the emissivity has a significant impact on the thermal performance of the UTC at high delivery temperature, so a selective surface should be used if possible. For estimation of the collector area, we have used an absorber with an emissivity of 0.1, but even under the best conditions this emissivity is difficult to obtain and maintain. Even if such a low emissivity were obtainable, the emissivity would increase because the collector may be mounted horizontally or tilted and would collect dust. The emissivity that should be used for a more accurate model needs to be obtained from field testing. If a higher emissivity were used in the model, there would be higher radiation losses and a subsequent increase in collector area and cost. Figure 5 also shows that an increase in the ambient temperature raises the output temperature by a similar amount, while decreasing the heat output due to higher radiation losses as a consequence of the higher collector surface temperature.

### 3.2 Integrated Solar Desiccant System

As stated before, the models for the solar collector and the desiccant cooling system were used together to predict the performance of the integrated system and the required surface area for the collector array. Figure 6 compares the thermal COP and cooling capacity of two solar desiccant cooling systems, one integrated with an unglazed transpired collector and the other with a glazed collector. As mentioned before, and as can be seen here, the performance of the system depends on the rotational speed of the dehumidifier. The optimum dehumidifier rotational speed depends on the regeneration temperature for a specific dehumidifier design. As expected, and as can be seen in Figure 6, the cooling capacity (i.e., the amount of cooling delivered to the space) of the desiccant system depends on the regeneration temperature and is independent of the type of air heater.

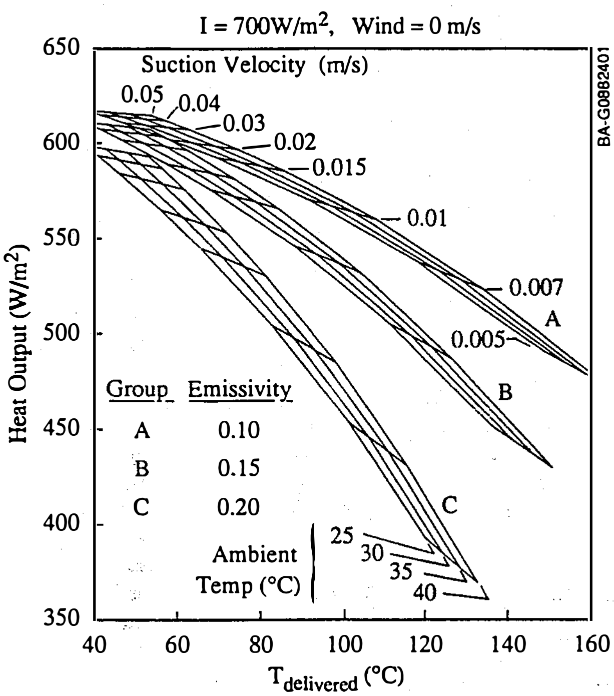


**Fig. 5 Effect of Emissivity and Ambient Temperature on Performance of Unglazed Transpired Collector**

The thermal COP of the UTC-desiccant system is lower than that of the glazed collector desiccant system. For example, at a regeneration temperature of 70°C, the thermal COP of the UTC system is about 0.41, while the thermal COP of the glazed system is about 0.92. These values correspond to a dehumidifier rotational speed of 3.33 rev/h in which the cooling capacity is maximum. The thermal COP for the unglazed system is lower because the heat of adsorption released during moisture adsorption is not used by the unglazed collector to preheat the air before it enters the collector (note points 3 and 4 on Figure 2). This is because the unglazed transpired collector works on the principle of drawing ambient air over its entire surface, and with its current design, it is not suited for drawing air from ducts. On the other hand, the glazed collector can use most of the heat of adsorption recovered by the heat exchanger (note points 3 and 4 on Figure 1).

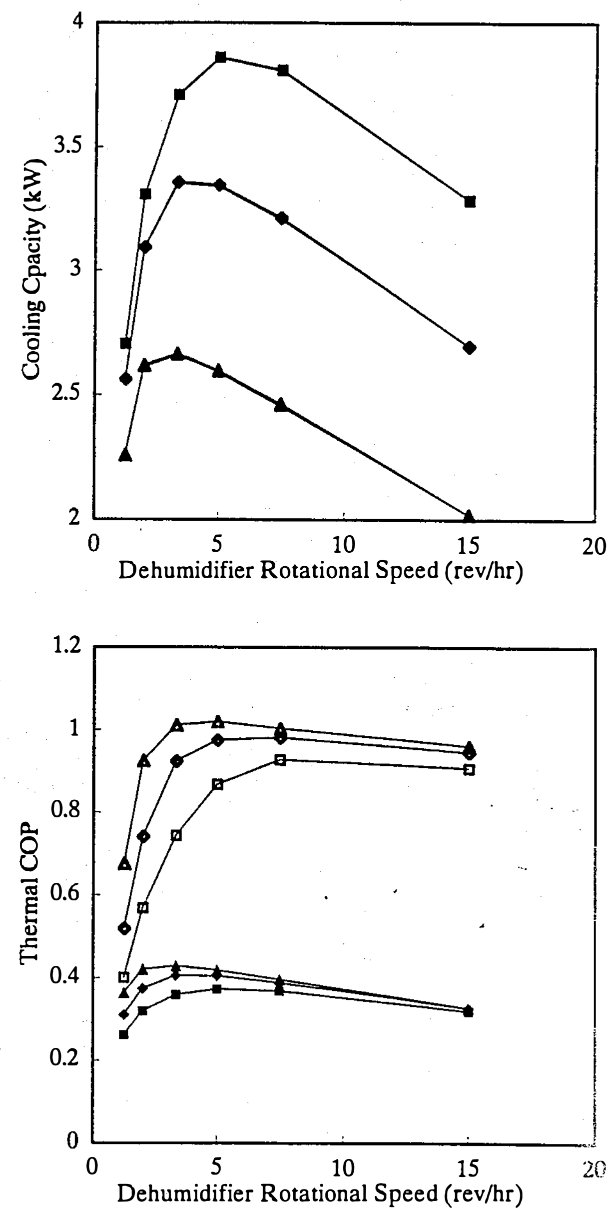


**Fig. 6 Performance of Solar Desiccant Cooling System with either Unglazed Transpired Collector (UTC) or Glazed Collector: Cooling Capacity (top figure), Thermal COP (bottom figure)**

The required collector surface areas and costs were calculated as follows. Using results on Figure 6, the optimum dehumidifier rotational speed was selected at maximum cooling capacity for a given temperature. Then the thermal COP of each solar desiccant system was obtained at this optimum speed. The regeneration heat required to be supplied by the collector, which is *cooling capacity* divided by *thermal COP*, was calculated. Based on the regeneration temperature and incident solar radiation, the efficiency of the collector was obtained, and the collector area required to match the required regeneration heat input was calculated. For the transpired collector, areas were selected to satisfy the required suction velocity. The cost of the glazed collector was estimated at about $30/ft$^2$ ($323/m$^2$). The cost of the unglazed transpired collector was estimated at about 1/3 of the cost of the glazed collector at $10/ft$^2$ ($108/m$^2$), because fewer materials and less labor were involved in the fabrication of the unglazed collector (4).

Table 2 summarizes results of the above analysis for UTC and glazed desiccant cooling systems that can deliver about 3 tons (10.56 kW) of cooling. When the unglazed transpired collector panels were placed in a vertical configuration, as is done for ventilation preheat applications, the performance was marginal. The reason is that the small amount of the sun's energy hitting the vertical surfaces during the summer requires a large panel area (94.4 m$^2$). When the panels were placed horizontally, however, the required area decreased, because the amount of incident solar energy was higher, and the efficiency of the unglazed collector was greater, but the area still remained large (41.4 m$^2$). Although the unglazed system cannot use the heat of adsorption recovered by the heat exchanger, the higher efficiency of the unglazed collector helps to offset the lower thermal COP of the unglazed system (78% unglazed collector efficiency compared to 58% glazed collector efficiency). Although the thermal and solar COPs of the unglazed transpired desiccant system are lower than those of the glazed desiccant system, the lower cost per unit area of the unglazed collector makes the unglazed collector array the less expensive of the two. The unglazed collector area is 69% larger than the glazed collector area, but the cost for the unglazed array is still 44% less. It should be noted that when UTC is in a horizontal orientation, the unglazed collector cannot provide much energy for ventilation preheat in winter time. An optimum tilt angle should be obtained for both summertime and wintertime usage.

### 3.3 Cost Analysis

Simple payback times for the unglazed transpired and glazed collectors were estimated relative to conventional means of heating, i.e., natural gas. Capital cost for a 75,000 Btu/h gas furnace for regenerating a 3-ton system is about $1,500 (7). The price of natural gas was estimated at $5/million Btu. Assuming a thermal COP of 0.92 and a gas furnace efficiency of 80%, the cost of natural gas would be $0.245/h. Assuming 8 h/day of operation with 150 days in a cooling season, the cost of fuel will be $293/year, resulting in payback periods of 10 and 22 years for the unglazed transpired and glazed collectors, respectively. This assumes that the initial cost of a desiccant cooling system and the maintenance cost are

**TABLE 2. SUMMARY RESULTS OF SOLAR DESICCANT COOLING SYSTEMS**

| Collector Type | Glazed | Unglazed Transpired | |
|---|---|---|---|
| Orientation | Horizontal | Horizontal | Vertical |
| Regeneration Temperature (°C) | 70 | 70 | 70 |
| Cooling Capacity (kW) | 10.56 | 10.56 | 10.56 |
| Dehumidifier Rotational Speed (rev/hr) | 3.33 | 3.33 | 3.33 |
| Thermal COP ($Q_{cool}/Q_{regen}$) | 0.92 | 0.41 | 0.41 |
| Regeneration Heat Required (kW) | 11.48 | 25.93 | 25.93 |
| Incident Solar Radiation (W/m$^2$) | 800 | 800 | 500 |
| Efficiency of Solar Collector | 0.58 | 0.78 | 0.55 |
| Air Mass Flow Rate (kg/s) | 0.63 | 0.63 | 0.63 |
| Suction Velocity (m/s) | none | 0.017 | 0.008 |
| Area of Solar Collector (m$^2$) | 24.7 | 41.4 | 94.4 |
| Solar Collector Cost ($) | 7,978 | 4,471 | 10,196 |
| System Solar COP ($Q_{cool}$/I $A_{coll}$) | 0.5 | 0.32 | 0.22 |

the same for the gas systems and solar systems. Although the unglazed solar collector is an improvement over the glazed collector, its projected payback period is too high to be a viable economic alternative to conventional gas heating for desiccant regeneration in desiccant cooling systems. It should be noted that for other applications, such as ventilation preheat, which requires a lower temperature rise across the collector, the unglazed transpired collector may be more economical than natural gas heating.

## 4. CONCLUSIONS

The thermal COP of a desiccant cooling system regenerated with an unglazed transpired solar collector is less than 50% lower than the thermal COP of the desiccant system regenerated with a conventional glazed flat-plate collector. The reason for lower thermal COP is that the unglazed transpired collector could not use the heat recovered by the sensible heat exchanger in the desiccant cooling system. The collector efficiency of the unglazed collector is about 20% higher than that of the glazed collector at a regeneration temperature of 70°C. We found that the unglazed transpired collector system requires 69% more absorber area than the glazed flat-plate collector (41.4 $m^2$ versus 24.4 $m^2$) to provide 3 tons of cooling. Although the unglazed system requires a larger collector area than the glazed collector system, the lower cost of the unglazed transpired collector still makes it the more attractive choice of the two, provided there is enough roof or ground area for the collector. When compared with natural gas regeneration, however, it was found that the gas system would be significantly less expensive. With the optimistic assumptions made in this study, the unglazed transpired collector makes economic sense for desiccant regeneration relative to a glazed flat-plate collector. However, practical considerations, (such as having very low emissivity for a long period of time, the use of horizontal orientation during winter, and lower absorber heat exchanger effectiveness) may make this configuration of unglazed transpired collectors less attractive for desiccant regeneration.

## 5. ACKNOWLEDGMENTS

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## 6. NOMENCLATURE

| | |
|---|---|
| $A_{coll}$ | collector area ($m^2$) |
| COP | coefficient of performance, dimensionless |
| $c_p$ | air specific heat (J/kg °C) |
| I | solar insolation level (W/$m^2$) |
| $\dot{m}$ | air mass flow rate (kg/s) |
| $Q_{cool}$ | amount of cooling removed (kW) |
| $Q_{regen, GC}$ | regeneration heat for glazed collector (kW) |
| $Q_{regen, UTC}$ | regeneration heat for unglazed transpired collector (kW) |
| $T_3$ | temperature of air leaving heat exchanger (°C) |
| $T_4$ | regeneration temperature (°C) |
| $T_{amb}$ | ambient temperature (°C) |
| $T_{coll}$ | collector temperature (°C) |
| $T_{out}$ | outlet temperature from the collector (°C) |
| U | wind speed (m/s) |
| UTC | unglazed transpired collector |
| v | suction velocity (m/s) |
| $\alpha$ | collector absorptance, dimensionless |
| $\varepsilon$ | absorber surface emissivity, dimensionless |
| $\eta$ | collector efficiency, dimensionless |
| $\rho$ | density of air (kg/$m^3$) |

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