Improving Gas-Fired Heat Pump Capacity and Performance by Adding a Desiccant Dehumidification Subsystem

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This paper examines the merits of coupling a desiccant dehumidification subsystem to a gas-engine-driven vapor compression air conditioner. A system is identified that uses a rotary, silica gel, parallel-plate dehumidifier. Dehumidifier data and analysis are based on recent tests. The dehumidification subsystem processes the fresh air portion and handles the latent portion of the load. Adding the desiccant subsystem increases the gas-based coefficient of performance 40% and increases the cooling capacity 50%. Increased initial manufacturing costs are estimated at around $500/ton ($142/kW) for volume production. This cost level is expected to reduce the total initial cost per ton compared to a system without the desiccant subsystem.

Air-conditioning commercial and residential buildings use a significant amount of energy. In an effort to decrease cooling costs and reduce energy consumption, the U.S. Department of Energy (DOE) has sponsored research to improve cooling system efficiencies. Technologies being examined include vapor-compression heat pumps driven by gas-fired internal combustion engines. Using gas as the primary energy source may lead to reduced operating costs compared with conventional electric vapor-compression machines in areas of the United States that have relatively high electric costs.

The efficiency of a thermally activated heat pump system could be increased if the gas engine waste heat was recovered. This waste heat can be used to regenerate a desiccant dehumidifier that removes moisture from the process air. By combining vapor compression with desiccant dehumidification into a hybrid system, the sensible load can be met by the vapor compression evaporator, which does not have to operate at the low temperatures needed for dehumidification. The latent portion of the load can be met by the desiccant subsystem.

The potential for a hybrid system that efficiently handles both latent and sensible cooling loads in gas-engine-driven vapor-compression air conditioners is great. The system can be used in buildings with high internal moisture generation (such as supermarkets and health clubs) and buildings in humid climates with significant ventilation requirements. The system high building growth rates in the humid southeastern portion of the United States point toward an ever-increasing potential for such systems.

In this paper, we evaluate the potential of a thermally activated heat pump coupled to a desiccant cooling subsystem. An estimate of the additional capital cost of a promising desiccant subsystem is included.

Air Conditioning Systems

The majority of building cooling equipment systems use electric-driven vapor-compression machines. Vapor-compression systems meet sensible loads very effectively. Air passed over the coils of the vapor-compression evaporator gives up heat to the refrigerant; this process chills the air. High heat transfer rates and high-efficiency components make vapor compression attractive for sensible air cooling.

It is less efficient to use vapor-compression equipment to meet latent loads. To remove moisture from the air, the air must be cooled past its dew point at that temperature the water condenses on the coils. In some systems, the air is now colder than the required delivery temperature dictated by comfort and the sensible load. Depending on the cycle and system configuration, additional energy may be needed to reheat the air to the delivery temperature. These processes are illustrated in Figure 1.

Vapor Compression Systems

The efficiency of a thermally activated heat pump system could be increased if the gas engine waste heat was recovered. This waste heat can be used to regenerate a desiccant dehumidifier that removes moisture from the process air. By combining vapor compression with desiccant dehumidification into a hybrid system, the sensible load can be met by the vapor compression evaporator, which does not have to operate at the low temperatures needed for dehumidification. The latent portion of the load can be met by the desiccant subsystem.
most vapor-compression systems, the evaporator coil must operate 40°F to 60°F (5°C to 15°C) cooler than the delivery temperature in order to chill the air to its dew point. The coefficient of performance (COP) defined as cooling output divided by energy input, of the vapor-compression system is lowered by around 10% to 20% as a result (Howe 1983; Schlepp and Shusta 1984).

**Desiccant Cooling Systems**

Desiccant systems are well suited to reduce latent loads. The process air is brought in contact with a material with a high affinity for water. Moisture is adsorbed or absorbed by the desiccant material. During this process the heat of adsorption or absorption in the desiccant, resulting in higher air temperature. To meet sensible loads, a stand-alone desiccant system over-dries, then evaporatively cool the air to the desired supply temperature (see Figure 2). Since the dehumidification process results in dryer and warmer supply air, the process may be considered a way to convert latent loads into sensible loads.

The desiccant requires regeneration (drying) to remove the moisture picked up from the process air. A hot, regeneration airstream is passed over the moisture-laden desiccant. The regeneration air can be heated by solar collectors, electric heaters, gas burners, or waste heat sources.

**Hybrid Cooling Systems**

Combining components of a vapor-compression system with a desiccant system results in a hybrid that can efficiently meet the sensible and latent cooling loads. The vapor-compression machine in a hybrid system operates with higher evaporator temperatures, resulting in a higher thermal COP than vapor-compression-alone units. In addition, the hybrid system requires no reheat. The dehumidifier must only remove the moisture to meet the latent load since the sensible load is met by the vapor-compression machine. No over-drying is required (see Figure 3). This process reduces the size of the dehumidifier and the amount of energy required to regenerate the desiccant material compared with a stand-alone desiccant system.

Another advantage of the hybrid system is it reduces the required energy input (increased overall COP). Heat rejected by some components may be used to regenerate the desiccant, which eliminates or reduces the need for external regenerative heat. Hybrid systems have the option of using the heat rejected by the vapor-compression condenser for regeneration. If a gas engine is used to drive the compression (instead of the usual electric motor), significant amounts of engine waste heat may be available.

**Desiccant Materials and Dehumidifier Geometry**

Several different hybrid systems have been installed in specialized applications with unique cooling requirements (MacLaine-Cross 1988). In each of these cases, as well as studies examining the analytical, experimental, and beginning commercial feasibility of hybrid systems, the dehumidifier component has performed with the greatest efficiency on overall system performance. Dehumidifiers have not yet been produced in volume. Some think that the desiccant dehumidifier systems may be the answer to reducing both heat rejection and performance improvements.

The performance of a desiccant dehumidifier depends mainly on the type of desiccant material used, the internal geometry of the dehumidifier (i.e., how the desiccant is deployed within the dehumidifier matrix), and the operating parameters.

The material type affects size, range of operation (temperature, humidity, efficiency), cost, and service life of a dehumidifier. The desiccant choice also affecting the thermal COP and cooling capacity of a system. The geometry of a dehumidifier affects its pressure drop, size, and cost and, thus, the thermal and electrical COPs and cost of a cooling system. Control strategies can also affect the overall performance. The optimum combination of desiccants and geometries can provide high-efficiency and low-cost dehumidifiers for air-conditioning applications.

In this study, after an initial screening we selected three desiccant systems for detailed analysis: silica gel, lithium chloride, and molecular sieve. Silica gel has been recommended (e.g., Schlepp and Shusta 1984) for cold storage, solar water heating, and other applications. Molecular sieve has been recommended (e.g., Jurinak 1982) for gas-fired applications because of high efficiency and high moisture cycling capacity at high regeneration temperatures (248°F to 428°F [120°C to 220°C]).

We have completed considerable research on advanced dehumidifier geometries (e.g., Bharathan et al. 1983a; Bharathan and Parsons 1986). For this study we selected parallel-passage, rotary dehumidifiers for detailed analysis. Parallel-passage geometries have high static heat transfer rates and low pressure drop. A typical measure of heat transfer to pressure drop ratio, the Stanton number divided by the friction factor (St/f), is typically 0.49 for parallel passages compared with 0.06 for packed beds. A rotary dehumidifier was chosen over curved geometries for ease of control and constant outlet conditions.

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The basic wheel design as previously described was used for evaluating desiccant materials. Based on experience from experiments described in Bharathan et al. (1987), for typical small on medium cooling loads and a desired pressure loss of around 9.3 to 0.4 in of water (130 to 150 Pa), we arrived at a wheel outside diameter of the ft (1.0 m) and a wheel depth of 4 in (0.1 m). A design flow rate of 1200 cfm (0.6 kg/s) was used for both the process and regeneration streams for the dehumidifier. The wheel was divided equally into regeneration and process volumes. The maximum number of transfer units is achieved with the minimum practicable passage size. A matrix with a center-to-center spacing of 31 mil (0.8 mm) for the substrate sheets is practicable with current wheel manufacturing techniques. The wheel has 16 support spokes, which is typical of existing commercial rotary heat exchangers (FYT 1984).

For the design of the solid particle wheel (silica gel and molecular sieve), microbead particles were used with a size range of 3 to 4 mil (15 to 105 mm). The substrate is 0.4 mic (10 µm) thick polyester tape coated on both sides with 0.4 mil (10 µm) of adhesive. The resulting nominal air passage gap is 23 mil (0.580 mm). Blockage of the nominal frontal area is 27.5%, and the actual flow area for each matrix impregnated with the salt is used. We set the thickness of the matrix equal to 8.7 mil (0.22 mm) so the blockage and design pressure drop would be equal for solid and hygroscopic salt desiccant wheels. To prevent the salt solution from dripping from the matrix, the matrix should be impregnated with a porous fiberglass to hold 2.4 lb (1.1 kg) of lithium chloride and a desiccant fraction of 0.116.

The hygroscopic-salt desiccant wheel design is similar to that of the solid particle wheel except that instead of a thin tape with particulate material attached by adhesive, a porous fiberglass substrate sheets is practicable with current wheel manufacturing techniques. The wheel has 16 support spokes, which is prototypical of existing commercial rotary heat exchangers (FYT 1984).

Performance of the desiccant wheel design was studied in Bharathan et al. (1987), for typical small on medium cooling loads and a desired pressure loss of around 9.3 to 0.4 in of water (130 to 150 Pa), we arrived at a wheel outside diameter of the ft (1.0 m) and a wheel depth of 4 in (0.1 m). A design flow rate of 1200 cfm (0.6 kg/s) was used for both the process and regeneration streams for the dehumidifier. The wheel was divided equally into regeneration and process volumes. The maximum number of transfer units is achieved with the minimum practicable passage size. A matrix with a center-to-center spacing of 31 mil (0.8 mm) for the substrate sheets is practicable with current wheel manufacturing techniques. The wheel has 16 support spokes, which is typical of existing commercial rotary heat exchangers (FYT 1984).

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Dehumidifier performance was determined using the detailed analogy method (combined potential and specific capacity ratio). This method allows the coupled, simultaneous, partial differential governing equations to transform into independent equations. The key features of the equations are retained, and properties of different desiccants are easily integrated yet solutions are much more consistent. This provides a flexibility advantage in finite difference methods. Detailed descriptions of the model are beyond the scope of this paper but can be found in Parsons et al. (1987), Bharathan et al. (1987a, b), and Macalma-cross (1978).

For a silica gel rotary dehumidifier test article (Bharathan et al. 1987b), predicted effectiveness of exchange using the analogy method was confirmed to lie within 10% of the experimentally measured values. The proposed dehumidifier design for the present application is similar to the tested configurations, and thus the predicted performance is expected to be well within acceptable engineering uncertainty limits for a silica gel parallel-passage dehumidifier. The method was extended to molecular sieve and lithium chloride wheels but predictions have not been verified through actual experiments.

Comparison of Three Desiccant Materials Performance

Comparisons of outlet process and regeneration air states for the three materials selected (microbead silica gel, lithium chloride, and molecular sieve) were performed for a variety of conditions. Although the exhaust portion of the thermally activated heat pump (TAHP) waste heat is available at high temperature, we found that the amount of waste heat available, at typical latent load fractions, and commonly desired cooling delivery temperatures result in regeneration temperatures in the same range as found in solar desiccant systems (16°F to 19°F [75°C to 90°C]). Figure 5 shows the outlet air states for the three wheel materials with flow rates of 1200 cfm (0.6 kg/s) for both airstreams. Inlet airstream humidities of 0.013 lb water/lb dry air were used and the process and regeneration airstream inlet temperature was set at 95°F (35°C) and 95% relative humidity. Various outlet states are obtained by varying the rotational speed. There is a minimum outlet process air humidity at a certain rotational speed that corresponds to the maximum outlet regeneration air humidity.

The microbead silica gel wheel produces the lowest outlet humidity, but the difference between the silica gel and lithium chloride wheel is not large. At the lowest humidity point, the silica gel wheel produces an outlet condition of 15°F (8°C) and 0.0057 lb water/lb dry air, and the lithium chloride wheel produces an outlet of 16°F (6°C) and 0.0061 lb/lb. The molecular sieve wheel does not remove as much water vapor from the process air as the other materials since it requires higher regeneration temperatures. It may be possible to take advantage of the portion of heat available at high regeneration temperatures, which is available at typical solar conditions. Although the exhaust portion of the thermally activated heat pump (TAHP) waste heat is available at high temperature, we found that the amount of waste heat available, at typical latent load fractions, and commonly desired cooling delivery temperatures result in regeneration temperatures in the same range as found in solar desiccant systems (16°F to 19°F [75°C to 90°C]). Figure 5 shows the outlet air states for the three wheel materials with flow rates of 1200 cfm (0.6 kg/s) for both airstreams. Inlet airstream humidities of 0.013 lb water/lb dry air were used and the process and regeneration airstream inlet temperature was set at 95°F (35°C) and 95% relative humidity. Various outlet states are obtained by varying the rotational speed. There is a minimum outlet process air humidity at a certain rotational speed that corresponds to the maximum outlet regeneration air humidity.

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System Configurations

Many component arrangements are possible for conventional desiccant systems (Cyun 1986; Kerlinborough et al. 1986; Meckler 1986) and hybrid systems (Hove 1983; Schlopp and Schulte 1984; Sonnino 1986). Most hybrid systems begin with two airstreams similar to conventional desiccant systems whereby the regeneration airstream, which is delivered to the space to meet the load, typically is first passed through a dehumidifier where it is dried and warmed. Most hybrid systems use an indirect natural gas burner, assumed to be an indirect natural gas burner, with an efficiency of 90%.

The first system configuration examined here and a representative psychrometric chart of the air processes are shown in Figure 6. In this layout, only the flow required to meet the fresh air requirement is passed through the dehumidifier. The cooled, dehumidified fresh air (point 3) and the remainder of the process air (point 6) are passed through the vapor compression system for sensible cooling only. In the dehumidification regeneration stream, a sensible heat exchanger is used to supply stream heat from the regeneration stream, used to remove the moisture from the dehumidifier, is typically heated by a series of components that might include the vapor compression condenser, a regenerative heat exchanger, waste heat recovery, and auxiliary heat sources.

System Modeling

The indirect evaporative cooler, direct evaporative cooler, and sensible heat exchangers are all modeled as constant effectiveness devices. The gas-fired internal combustion engine was modeled after equations presented by Segaser (1977). By using a load fraction (design load/full capacity), of 0.9, the engine efficiency is 33%, water jacket waste heat is 25% at 212°F (100°C), and exhaust gas waste heat is 6% at 212°F (100°C). The exhaust gas heat recovery is limited by condensation of corrosive combustion products at around 158°F (70°C). It was judged that the remaining waste heat (lube oil, radiation, etc., making up the remaining 17%) was not economical to recover.

Heat pump performance is modeled as presented by Howe (1983). Nonstandard operating conditions were accounted for by using data from a commercial heat pump (Hove 1983). The equation for COP is

\[ \text{COP} = \frac{3.58 + 0.0162}{\text{FLR tons}} - 0.073 \text{ FLR tons} - 0.0398 \theta_{\text{cond}, \text{in}} - \theta_{\text{comp}, \text{in}} = 15 \]

where FLR is the fractional load ratio (load/full capacity) set to 0.90. In this study, tons are the load in those units. The term \( \theta_{\text{cond}, \text{in}} \) is the condenser air inlet temperature, and \( \theta_{\text{comp}, \text{in}} \) is the evaporator air inlet saturation temperature.

In this paper, performance is evaluated at the ASHRAE Standard 55-74 comfort zone, which is slightly outside the ASHRAE comfort zone defined by Standard 55-74. We feel use of the ASHRAE standard is appropriate for off-design or seasonal performance evaluations. Yearly performance values are presented in this study, an example cooling load of 9.5 tons (37 kW) was used. This value includes the remaining 17% (rounding off to 17%) was not economical to recover.

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include the load imposed on the machine by the fresh air requirement. A fresh air requirement of 112 cfm/ton (15 L/s) of cooling was used in this study. 1100 cfm (15.6 m3/s). This value is representative of many buildings, but higher ventilation rates may be required for energy-efficient construction or specialized applications. Using this fresh air requirement implies a partial recirculation mode of operation. These loads resulted in a 3.2 ft (1 m) diameter wheel for the first system configuration.

Combining the building and ventilation loads using the described ARI design point results in a total load on the hardware of 12.5 tons (46 kW), of which 40% is latent capacity.

A final assessment of the load calculations is that the enthalpy difference between the room and supply air is limited to 4.05 Btu/lb (15 kW/m3) for comfort. This limitation resulted in a room air delivery state of the 62°F (16.5°C) and 0.0097 lb/lb absolute humidity. The total room air flow rate for these loads is 4450 cfm (2.47 kg/s) (1000 cfm [0.56 kg/s] fresh air).

**PERFORMANCE RESULTS**

The two TARP-desiccant hybrid systems were compared using component efficiencies and pressure drops summarized in Table 1. In addition, fan efficiency was assumed to be 50%. These values correspond to relatively conservative existing commercial equipment.

Table 2 summarizes load characteristics and Table 3 summary of the two hybrid systems using these commercial components. A total equivalent thermal COP is used and defined as:

$$\text{total cooling capacity}$$

$$\text{gas input + electric input}/0.3$$

The factor of 0.3 is based on a conversion efficiency from fuel to electricity of 50%. For a complete evaluation, relative electricity and gas cost factors should be included to arrive at an overall COP. These factors vary widely across the country, so a generic factor was used for relative comparison of systems in this paper.

**System 1**

The regeneration temperature of 69°F (20°C) was chosen to minimize auxiliary energy and to achieve high overall COP.

Note that the dehumidifier inlet state is not identical to the conditions used in the wheel comparisons of Figure 5. Because we are using the ARI design point, the outlet state is slightly different.

Adding on the desiccant dehumidification subsystem (including the dehumidifier, heat exchangers, and additional fans) to the vapor compression unit increases the performance of the total system in several ways. First, the temperature and humidity of the air entering the vapor compression evaporator in the hybrid system is different than if a vapor compressor were meeting the total cooling load. The COP of the vapor compression unit is dependent on the state of the entering air (see modeling discussion). In the case fresh and recirculation airflows were used in a vapor compression system alone, the entering wet-bulb temperature would have been 75°F (24°C) and the resulting vapor compression COP (defined as cooling/compressor work) would have been 3.18. The hybrid system evaporator wet-bulb for the conditions presented in Table 2 is 62°F (16°C) and the vapor compression COP is 3.18. This higher COP results in a decrease in the amount of fuel required to meet a given load.

Second, since the vapor compression subsystem does not handle any dehumidification load, process air does not need to be reheated. The reheat energy for the above load, had it been met by vapor compression alone, would have been approximately 24,400 Btu/h (7.2 kW), which could be provided by a number of sources including heat waste.

The most important advantage of adding the dehumidification subsystem is the added cooling capacity. The desiccant subsystem adds 4.4 tons (15.5 kW) of cooling capacity when using the engine waste heat. These conditions require only a small amount of supplementary gas heat (4400 Btu/h [1.3 kW] compared with 1.0 × 10^6 Btu/h [30.9 kW] to run the gas engine) to regenerate the desiccant. If we consider the waste heat as a free heat source, the energy input to obtain this increased capacity is only 4400 Btu/h (1.3 kW) in this example. A gas COP of the desiccant subsystem computed on this somewhat artificial basis is 11. At other conditions, this add-on COP can be infinite since no additional auxiliary is needed. All regeneration heat is supplied by the waste heat from the engine.

A gas-engine-driven cooling system operating with vapor compression only and under conditions comparable to the hybrid system presented in Table 2 has an overall gas COP of 1.01 (with reheat not included since reheat could be taken from any number of "free" heat sources). Using the desiccant dehumidification subsystem increases the gas COP by more than 402 to 1.42.

**System 2**

In the second system configuration, all the process air passes through the dehumidifier subsystem components and the vapor compression evaporator. Room air was mixed with the minimum required fresh air as the source of the supply air to minimize the required capacity, and the ambient mixed with the remaining room air as the source of the regeneration stream. Both air process paths used a flow rate of 4450 cfm (2.47 kg/s).
Results for a regeneration temperature of 122°F (50°C) are presented. Higher regeneration temperatures resulted in increasing auxiliary regeneration energy requirements, and lower regeneration temperatures resulted in excess waste heat being available.

Note that the electric COP is not broken down into vapor compression and desiccant subsystem contributions since in this system the two subsystems have the same airflow paths.

System Selection

The System 2 design is quite different from the System 1 design. The gas COP for System 2 is somewhat higher, but since all the process air passes through the dehumidifier and other desiccant subsystem components, the electric COP decreased by around 40%. Combined equivalent COP is slightly greater for System 1. Although these performance parameters are important, perhaps the largest difference between the two systems is the component size. The vapor compression/engine capacity is around 23% lower for System 2 since the desiccant subsystem handles more of the sensible load through the evaporative coolers. The desiccant subsystem components for System 2 are around 4.5 times larger than those used in System 1 because the full supply airflow passes through them. Larger airflow rates result in a dehumidifier diameter of greater than 6.4 ft (2 m) for System 2, and the dehumidifier for System 1 is 3.3 ft (1 m) in diameter. The probable increase in cost for System 2 over System 1 is perhaps not warranted by the increase in performance. Therefore, we chose System 1 as the better of the two systems.

System Layout and Cost

The chosen system was laid out and the manufacturing costs for the two subsystems, as shown in Figure 8. The cost of the desiccant subsystem regeneration is not broken down into its components for System 2 are around 4.5 times larger than those used in System 1 because the full supply airflow passes through the desiccant subsystem to achieve the final temperature boost in the dehumidifier regeneration airstream. An alternative (assumed in the costing and layouts of this system), which is more practical and convenient, is to increase the heat exchangers can be improved by increasing their effectiveness. This can be done by decreasing air velocity and increasing flow length in the components. This, however, increases the size, cost, and air pressure drop. A complete economic and performance trade-off study is required to fully resolve these issues.

The cost of the desiccant subsystem regeneration heat exchanger probably should be credited against the cost of the internal combustion engine cooling radiator. This credit was not included in our cost estimate.

Integration of the two subsystems allows an increase in the vapor compression evaporator refrigeration pressure and temperature, which results in an increase in the heat pump COP, which is reflected in the increased overall system COP. There may also be an increase in cooling capacity at higher evaporator temperature that would require resizing the coil. However, we assumed that the cooling capacity is the same.

The dehumidification subsystem can be contained within an envelope 3.0 by 3.3 by 3.3 ft (1 by 1.0 by 1.0 m) and has the operating specifications shown in Table 4.

Table 5 provides a summary of the cost breakdown for the system described as a nominal 4.6 ton (15.5 kW) capacity. The cost estimates presented in this section come from Maclaine-cross (1986) and Maclaine-cross and Parsons (1986) plus current quotes from manufacturers. These sources are recent, and all presented costs can be assumed to be 1987 dollars for volume production. Volume production of 10,000 units/year is essential to bring dehumidifier and mass transfer systems (dehumidifier, heat exchangers, and evaporative coolers) to full supply airflow passes through them. Larger airflow rates result in a dehumidifier diameter of greater than 6.4 ft (2 m) for System 2, and the dehumidifier for System 1 is 3.3 ft (1 m) in diameter. The probable increase in cost for System 2 over System 1 is perhaps not warranted by the increase in performance. Therefore, we chose System 1 as the better of the two systems.

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CONCLUSIONS AND RECOMMENDATIONS

The performance of a TAHP system can be greatly improved by adding a desiccant cooling subsystem. Specifically, for the selected system layout at ARI standard design conditions:

1. The gas COP is increased by approximately 50%.
2. The total cooling capacity is increased by approximately 50%.

The selected hybrid system that separately handles latent and cooling load can be designed as a stand-alone add-on package with simple, standard system interfaces. For a unit that adds about 4.4 tons of cooling capacity to a 10.5-ton TAHP, components can be integrated in a 3.3 by 3.3 by 5.9 ft (1 by 1 by 1.8 m) package.

We estimate the first manufacturing cost of the system i configuration add-on desiccant subsystem package to be $2150 or about $490/ton of additional cooling capacity for volume production. Current TAHP system first costs are about $1700/ton (Domingo 1986), which includes manufacturer, distributor, and retail profit but not installation. Maclaine-cross and Parsons (1986) indicate a cost factor of approximately two should be applied to the $490/ton figure to find a retail cost. Therefore, it is likely that adding a desiccant subsystem may decrease the total first cost per unit of cooling delivered and increase the gas COP by 40% (fuel consumption is around 60% of a heat-pump-only system). Operating cost advantages of the proposed hybrid system depend on local electric and gas rate structures. Relative comparisons in this study were made using a rough 0.3 gas-to-electric conversion factor.

Use of silica gel resulted in the highest moisture removal effectiveness of the three materials examined at regeneration temperatures of 203°F (95°C) or less. Predictions using lithium chloride resulted in only 5% less moisture removal. Molecular sieve dehumidifier performance predictions indicated 5% less moisture removal than silica gel; thus, this material is not as suitable for these regeneration temperatures and mass exchange in parallel-passage rotary desiccant dehumidifiers for solar cooling applications. Experimental study of heat and mass exchange in parallel-passage rotary desiccant dehumidifiers for solar cooling applications. SERI/TR-252-2983. Golden, CO: Solar Energy Research Institute.


It is possible to regenerate the desiccant dehumidifier with fairly low regeneration temperatures and meet cooling loads with typical latent heat to sensible heat ratios with very little auxiliary added heat. The quantity of waste heat available and the desire to minimize auxiliary input resulted in regeneration temperatures of 122°F to 185°F (50°C to 85°C) being used with these wheels and system layouts.

This study supports the addition of a desiccant subsystem to a TAHP for particular climates and favorable gas and electricity rates. First costs per unit of cooling do not increase and energy consumption is significantly reduced.

Future research direction should include: integrating component cost relations and a fuel and electricity cost structure into the model to allow optimization on a life-cycle cost or cost-of-service basis; performing sensitivity studies and comparing the effects of component improvements on overall system cost and performance; extending the systems model to allow evaluation of seasonal performance and perform evaluations with climatic data for several typical cities with actual utility rates and experimentally investigating hybrid cooling system performance to verify system model predictions.

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Renewable Energy Division of the U.S. Department
of Energy.
### TABLE 1  
Commercial Equipment Performances

<table>
<thead>
<tr>
<th>Component</th>
<th>Pressure Drop (in H₂O)</th>
<th>(Pa)</th>
<th>$\epsilon$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Direct evaporative cooler</td>
<td>0.56</td>
<td>160</td>
<td>0.86</td>
</tr>
<tr>
<td>Indirect evaporative cooler</td>
<td>0.56</td>
<td>140</td>
<td>0.80</td>
</tr>
<tr>
<td>Air-to-Air Sensible Heat Exchanger</td>
<td>0.16</td>
<td>40</td>
<td>0.79</td>
</tr>
<tr>
<td>Air-to-Air and Sensible Heat</td>
<td>0.16</td>
<td>40</td>
<td>0.79</td>
</tr>
<tr>
<td>Air-to-Water Sensible Heat Exchanger</td>
<td>0.20</td>
<td>50</td>
<td>0.79</td>
</tr>
<tr>
<td>Other Pressure Drops (ducting, delivery, filters, etc.)</td>
<td>1.00</td>
<td>250</td>
<td></td>
</tr>
<tr>
<td>Dehumidifier</td>
<td>0.60</td>
<td>150</td>
<td></td>
</tr>
<tr>
<td>Vapor Compression Evaporator and Condenser</td>
<td>0.24</td>
<td>60</td>
<td></td>
</tr>
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</table>

$c$ = effectiveness

### TABLE 2  
Building Load Characteristics

<table>
<thead>
<tr>
<th>Cooling Load</th>
<th>Ton</th>
<th>kW</th>
</tr>
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<tbody>
<tr>
<td>Building</td>
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<td></td>
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<tr>
<td>Sensible</td>
<td>7.88</td>
<td>27.7</td>
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<tr>
<td>Latent</td>
<td>2.66</td>
<td>9.3</td>
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<tr>
<td>Total</td>
<td>10.52</td>
<td>37.0</td>
</tr>
<tr>
<td>Ventilation</td>
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<td></td>
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<tr>
<td>Sensible</td>
<td>1.31</td>
<td>4.6</td>
</tr>
<tr>
<td>Latent</td>
<td>1.31</td>
<td>4.6</td>
</tr>
<tr>
<td>Total</td>
<td>2.62</td>
<td>9.2</td>
</tr>
<tr>
<td>Total Loads</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sensible</td>
<td>9.18</td>
<td>32.3</td>
</tr>
<tr>
<td>Latent</td>
<td>3.95</td>
<td>13.6</td>
</tr>
<tr>
<td>Total</td>
<td>13.13</td>
<td>46.2</td>
</tr>
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</table>

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### TABLE 3
Comparison of System Performance

<table>
<thead>
<tr>
<th>Performance</th>
<th>System 1</th>
<th>System 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>COPs</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Complete System</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas</td>
<td>1.62</td>
<td>1.75</td>
</tr>
<tr>
<td>Electric</td>
<td>13.77</td>
<td>7.97</td>
</tr>
<tr>
<td>Equivalent thermal</td>
<td>1.07</td>
<td>1.01</td>
</tr>
<tr>
<td>Vapor Compression Subsystem</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas</td>
<td>0.99</td>
<td>1.2</td>
</tr>
<tr>
<td>Electric</td>
<td>15.35</td>
<td>*</td>
</tr>
<tr>
<td>Equivalent thermal</td>
<td>0.82</td>
<td>*</td>
</tr>
<tr>
<td>Desiccant Subsystem</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas</td>
<td>11.9</td>
<td>3.3</td>
</tr>
<tr>
<td>Electric</td>
<td>11.9</td>
<td>*</td>
</tr>
<tr>
<td>Equivalent thermal</td>
<td>2.75</td>
<td>*</td>
</tr>
<tr>
<td>Flow Rates</td>
<td>(kg/s)</td>
<td></td>
</tr>
<tr>
<td>Dehumidifier Supply and Regeneration</td>
<td>0.36</td>
<td>2.47</td>
</tr>
<tr>
<td>Vapor Compression Evaporator and Condenser</td>
<td>2.67</td>
<td>2.47</td>
</tr>
<tr>
<td>Energy Inputs</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>Internal Combustion Engine (Gas)</td>
<td>30.9</td>
<td>19.5</td>
</tr>
<tr>
<td>Auxiliary Regeneration (Gas)</td>
<td>1.3</td>
<td>6.9</td>
</tr>
<tr>
<td>Fans (Electric)</td>
<td>3.3</td>
<td>5.8</td>
</tr>
<tr>
<td>Hardware Capacities</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>Vapor Compression Subsystem</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporator (all sensible cooling)</td>
<td>30.7</td>
<td>23.4</td>
</tr>
<tr>
<td>Condenser (heat rejection)</td>
<td>40.3</td>
<td>39.6</td>
</tr>
<tr>
<td>Compressor (work input)</td>
<td>9.6</td>
<td>7.2</td>
</tr>
<tr>
<td>Dehumidification Subsystem</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sensible</td>
<td>1.6</td>
<td>8.9</td>
</tr>
<tr>
<td>Latent</td>
<td>13.9</td>
<td>13.9</td>
</tr>
<tr>
<td>Total</td>
<td>15.5</td>
<td>22.8</td>
</tr>
<tr>
<td>Total Hardware</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sensible</td>
<td>32.2</td>
<td>32.2</td>
</tr>
<tr>
<td>Latent</td>
<td>13.9</td>
<td>21.9</td>
</tr>
<tr>
<td>Total</td>
<td>46.2</td>
<td>44.2</td>
</tr>
</tbody>
</table>

*COPs for System 2 are not broken down because of using a single process airstream.*
<table>
<thead>
<tr>
<th>Dehumidifier Subsystem Operating Specification (System 1)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cooling Capacity</strong></td>
</tr>
<tr>
<td><strong>Airflow Rate</strong></td>
</tr>
<tr>
<td><strong>Electrical Power</strong></td>
</tr>
<tr>
<td><strong>Water Consumption</strong></td>
</tr>
<tr>
<td><strong>Waste Heat Recovery</strong></td>
</tr>
<tr>
<td><strong>Auxiliary Gas Consumption</strong></td>
</tr>
<tr>
<td><strong>Auxiliary Gas COP</strong></td>
</tr>
<tr>
<td><strong>Waste Heat Thermal COP</strong></td>
</tr>
<tr>
<td><strong>Overall Thermal COP</strong></td>
</tr>
<tr>
<td><strong>Combined Equivalent Thermal COP</strong></td>
</tr>
<tr>
<td><strong>Dehumidifier Wheel Diameter</strong></td>
</tr>
</tbody>
</table>

*Based on auxiliary gas input only.
**Including and waste heat input.
***Auxiliary and waste heat input.
<table>
<thead>
<tr>
<th>Component</th>
<th>Estimated Cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Desiccant Dehumidifier</td>
<td>460</td>
</tr>
<tr>
<td>Counterflow Heat Exchanger</td>
<td>200</td>
</tr>
<tr>
<td>Regeneration Heater</td>
<td>150</td>
</tr>
<tr>
<td>Direct Evaporative Cooler</td>
<td>165</td>
</tr>
<tr>
<td>Dehumidifier, Heat Exchanger Drive Motors and Speed Reducers</td>
<td>70</td>
</tr>
<tr>
<td>Fans (2) and Fan Drive Motor</td>
<td>270</td>
</tr>
<tr>
<td>Evaporative Cooler Pump</td>
<td>7</td>
</tr>
<tr>
<td>Sheet Metal Enclosure</td>
<td>215</td>
</tr>
<tr>
<td>Air Filters and Dampers</td>
<td>45</td>
</tr>
<tr>
<td>Electronic Controller</td>
<td>50</td>
</tr>
<tr>
<td>Miscellaneous Actuators and Mechanical Hardware</td>
<td>40</td>
</tr>
<tr>
<td>Assembly Labor</td>
<td>120</td>
</tr>
<tr>
<td>Subtotal for Desiccant Subsystem Packaged Unit</td>
<td>1812</td>
</tr>
<tr>
<td>Exhaust Heat Recovery Unit (Mounted on Internal Combustion Engine)</td>
<td>350</td>
</tr>
<tr>
<td>Estimated Total Manufacturing Cost</td>
<td>21.62</td>
</tr>
<tr>
<td>$/Ton of cooling (additional cost/additional capacity)</td>
<td>491</td>
</tr>
</tbody>
</table>

TABLE 5
Desiccant Cooling Subsystem Cost
(Nominal 4.4 Ton Capacity)
Figure 1. Psychrometric schematic of supply air states in a vapor compression cooling system

Figure 2. Psychrometric schematic of supply air states in a desiccant cooling system
Figure 3. Psychrometric schematic of supply air states in a simple hybrid cooling system.
Figure 4a. Schematic of a parallel-plate rotary dehumidifier
Figure 4b. Front view of Microbead®-silica-gel parallel-plate rotary dehumidifier

Figure 5. Comparison of wheel performance with airflow rates of 1120 scfm (0.6 kg/s) and an inlet regeneration temperature of 185°F (85°C)

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Figure 6. Schematic and psychrometric chart representation of System 1
Figure 7. Schematic and psychrometric chart representation of System 2
Figure 8. Integrated system schematic showing subsystem interfaces