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PERFORMANCE ASSESSMENT OF A DESICCANT COOLING SYSTEM IN A CHP APPLICATION INCORPORATING AN IC ENGINE

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ABSTRACT

Performance of a desiccant cooling system was evaluated in the context of combined heat and power (CHP). The baseline system incorporated a desiccant dehumidifier, a heat exchanger, an indirect evaporative cooler, and a direct evaporative cooler. The desiccant unit was regenerated through heat recovery from a gas-fired reciprocating internal combustion engine. The system offered sufficient sensible and latent cooling capacities for a wide range of climatic conditions, while allowing influx of outside air in excess of what is typically required for commercial buildings. Energy and water efficiencies of the desiccant cooling system were also evaluated and compared with those of a conventional system. The results of parametric assessments revealed the importance of using a heat exchanger for concurrent desiccant post cooling and regeneration air preheating. These functions resulted in enhancement of both the cooling performance and the thermal efficiency, which are essential for fuel utilization improvement. Two approaches for mixing of the return air and outside air were examined, and their impact on the system cooling performance and thermal efficiency was demonstrated. The scope of the parametric analyses also encompassed the impact of improving the indirect evaporative cooling effectiveness on the overall cooling system performance.

1 INTRODUCTION

Recent advancements in desiccant dehumidification and evaporative cooling technologies signal the incipience of a new generation of thermally activated HVAC products that further enhance the technological portfolio of distributed energy resources (DER). This is a breakthrough for combined heat and power (CHP), distributed generation (DG), solar or waste heat-driven cooling, and even stand-alone applications. Through innovative system configuration and integration, such systems can facilitate effective temperature and humidity control for buildings with the most stringent ventilation requirements in a vast domain of climatic conditions. In CHP applications, thermally activated desiccant cooling systems provide a significant energy-saving advantage over conventional systems. Implementation of such cooling technologies can also lead to significant downsizing of on-site power generators. In the interest of consistency, systems incorporating desiccant dehumidification and evaporative cooling will be referred to as "desiccant cooling systems" hereafter.

Because of these attributes, desiccant cooling technologies are positioned to help meet challenges surrounding critical issues, including grid congestion, energy price volatility, and emissions. The environmental benefits of these and similar concepts promoting use of waste heat constitute the basis for the recent initiatives for promoting output-based emission standards/regulations. Another point to be made is that, in addition to providing a viable alternative cooling approach, desiccant cooling can also operate in parallel with conventional systems for more efficient and/or economical fuel utilization. One example is a case in which the export of excess on-site power generation may not be cost-effective. Such circumstances may promote a combination of both desiccant and conventional cooling technologies.

It is the objective of this paper to demonstrate the thermodynamic advantage of a desiccant cooling system in the context of CHP. A reciprocating internal combustion (IC) engine is implemented in the CHP system, which provides a heat recovery opportunity for regeneration of the desiccant dehumidifier. The specific objectives to be achieved in this study are as follows:

- Demonstrate the cooling and energy performance of the system and its capability to provide comfort for a wide range of climatic conditions
- Perform parametric analyses for assessment of different operating strategies/system configurations
- Evaluate the system water consumption

For humidity and altitude considerations, the system under study is assumed to be located in Atlanta, Georgia, representing a hot and humid region in the United States. However, the range of climatic conditions adopted for analysis accommodates hot and dry conditions as well. In characterizing the space design load, return- and supply-air conditions for the building (indoor space) are assumed in congruity with those typically considered in commercial HVAC design and specifications. Although the results of this study are based on the supply air flow rate of

0.866 kg/s (46.6 m^3/min at standard conditions), normalization of the results is permissible to evaluate proportionately larger systems. (This flow rate was dictated by the availability of a commercial indirect evaporative cooling system and its laboratory performance results.) The study also compares the desiccant cooling system with a typical air-cooled conventional unit in terms of energy efficiency and water consumption.

Because this study focuses only on the system design performance, further studies are needed to quantify the annual energy-saving potential of similar systems for different building types and climatic zones. Examination of a hybrid system that integrates desiccant, evaporative, and conventional cooling technologies, in conjunction with energy and water efficiency, is also a part of the plan for the future studies. The future potential of various gas-fired DER technologies [1] is an incentive for consideration of other power generators as well.

2 SYSTEM DESCRIPTION

2.1 Desiccant Cooling System

As depicted in Figure 1, the system under consideration consists of a power generation IC engine, a thermally activated rotary desiccant wheel (DES), an air-to-air heat exchanger (HX), an indirect evaporative cooler (IEC), and a direct evaporative cooler (DEC). The subsystems of the desiccant cooling system can operate in concert to satisfy the required conditions of the supply air entering the indoor space.

The thermal energy output of the engine, from the jacket water and exhaust gas, is recovered for regeneration of the desiccant component to dehumidify either the incoming outside air (O.A.) or a mixture of O.A. and return air (R.A.). The flow rate of O.A. not only has to meet the ventilation requirement, but it has to compensate for the exhaust stream of the IEC unit for a mass balance. The return air is allowed to mix with either the preconditioned O.A. at the IEC inlet or the unconditioned outside air at the DES inlet. In the latter case, because of an increase in the airflow rate of the dehumidified air, the heat exchanger does not operate in a balanced-flow mode, unless the secondary airflow is augmented by an additional amount of outside air (Figure 1). These O.A./R.A. mixing options enhance the system operation flexibility in effectively targeting the cooling load and its latent/sensible composition.

The heat exchanger downstream of the desiccant wheel is designed for sensible cooling of the dehumidified air, which is important for IEC performance improvement. The exhaust flow leaving the IEC unit forms the secondary stream of the heat exchanger (rotary or fixed core). As another alternative, in lieu of the IEC exhaust, outside air is used for the post-cooling process as well. The outside air leaving the heat exchanger is then used as a preheated regeneration inlet air for the desiccant wheel. At the inlet of the IEC, as the mixture of O.A. and R.A. flows through the unit in the supply-flow direction, a portion of the air is discharged into wetted and perpendicularly oriented channels in successive stages. The product and exhaust air streams leaving the unit can attain temperatures well below the IEC inlet wet-bulb (WB) temperature, but higher than the corresponding dew-point temperature. It should be noted that the humidity ratio of the product air does not change because the air is only sensibly cooled as it flows through the IEC unit. Because of its high moisture content, the exhaust stream is discharged into the atmosphere after passing through the heat exchanger or upon leaving the IEC unit, depending on the operating mode. Therefore, the IEC exhaust stream leaving the HX does not lend itself to DES regeneration.

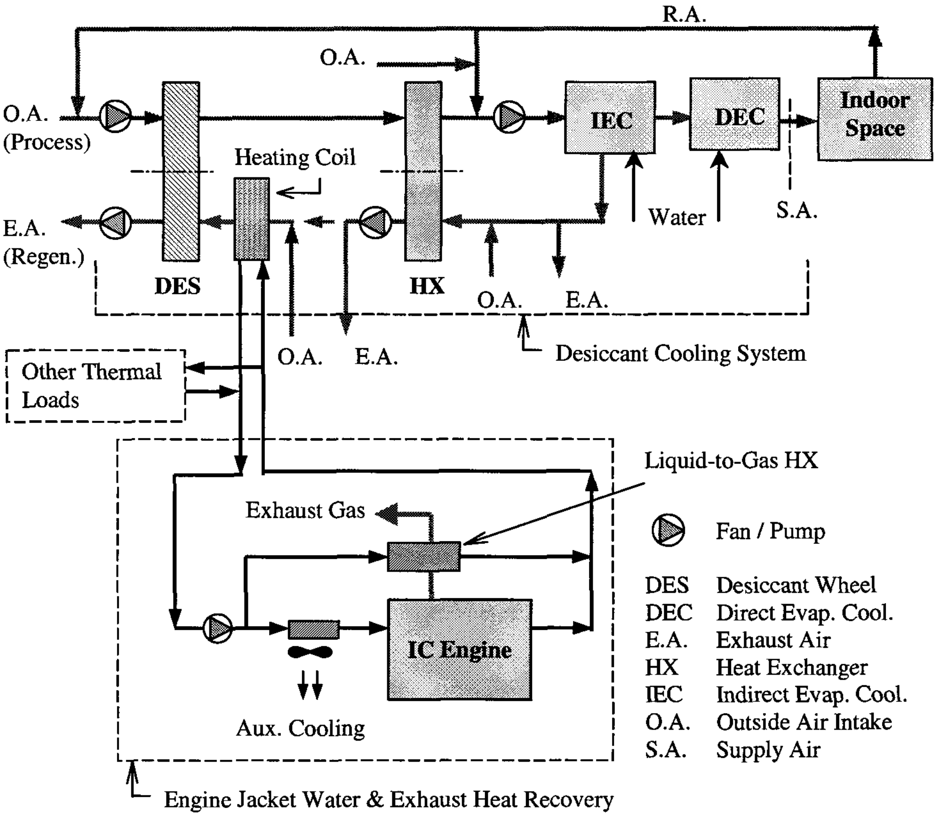

Figure 1: *System schematic*

The selected IEC unit operates at an exhaust-to-product (supply) flow ratio of about 0.9, requiring an inlet mass flow rate of about 1.9 times that of the product (supply) air entering the indoor space. This implies that about 47% of the IEC inlet air has to be composed of O.A., which is typically more than sufficient for building ventilation. In light of this, and in the interest of simplicity, no additional O.A. intake is considered, eliminating the need for a relief air stream from the indoor space. (Note that an additional O.A. intake is desirable to maintain the indoor space at a slightly positive pressure to control infiltration.)

2.2 Engine Heat Recovery

As depicted in Figure 1, the heating coil used for regeneration of the desiccant wheel is energized by the heat recovery from the engine jacket water and exhaust gas. No lubricant heat recovery from the engine is considered in this study. With all forms of heat recovery in place, the overall thermal efficiency of the engine can increase from about 30% for electricity generation alone to 75% [2]. Then, the full heat recovery is equivalent to 1.5 kWh per kWh of electrical energy output. Various configurations for engine heat recovery are available [2,3] and can lend themselves to optimum heat recovery. Considering that the desiccant regeneration air temperature is taken to be 90°C, a temperature of around 100°C for the working liquid entering the heating coil can be sufficient. A considerably higher regeneration operating temperature is also achievable with the heat recovery arrangement of Figure 1. However, an increase in the operating temperature reduces the amount of heat recovery per kWh of engine electrical output – a concern that has to be addressed in any CHP design process.

2.3 Modes of Operation

To evaluate the performance impact of each subsystem, the desiccant cooling system (Figure 1) is allowed to operate in a number of modes as described in Table 5.1.

Other perceivable modes not addressed in this study include the following:

- Use of only DEC unit (i.e., bypassing DES, HX, and IEC) as the ambient conditions permit
- Implementation of economizers and/or enthalpy exchangers to minimize the system load by maximizing direct use of O.A. to achieve a higher integrated/seasonal energy efficiency.

3 METHOD OF ANALYSIS

3.1 Cooling Characteristics

Critical to the system performance evaluation is characterization of the space and system cooling loads. The indoor space load is characterized by the following design supply- and return-air conditions that are typically used in design and specifications of conventional HVAC systems:

- Supply air (S.A.) at a temperature of 15°C and humidity ratio of 9.2 g/kg of dry air (13.3°C WB)
- Return air (R.A.) at a temperature of 25°C and humidity ratio of 10.2 g/kg of dry air (17.7°C WB).

This approach is important for the assessment and performance comparison of the desiccant cooling with conventional systems.

The sensible heat factor, SHF, of the indoor space load is determined by the following equation:

$$SHF_{Indoor\ Space} = \left(\frac{\dot{Q}_{CL,\ Sens.}}{\dot{Q}_{CL,total}} \right)_{Indoor\ Space} \cong \frac{(\dot{m}_a c_p)_{S.A.}(T_{R.A.} - T_{S.A.})}{(\dot{m}_a c_p)_{S.A.}(T_{R.A.} - T_{S.A.}) + \dot{m}_{a,S.A.}\, h_{fg}(W_{R.A.} - W_{S.A.})} \tag{1}$$

The following equations describe the SHF for the system load and capacity.

$$SHF_{System\ Load} = \left(\frac{\dot{Q}_{CL,\ Sens.}}{\dot{Q}_{CL,total}} \right)_{System} \cong \frac{(\dot{m}_a c_p)_{S.A.}(T_{Mixed} - T_{S.A.})}{(\dot{m}_a c_p)_{S.A.}(T_{Mixed} - T_{S.A.}) + \dot{m}_{a,S.A.}\, h_{fg}(W_{Mixed} - W_{S.A.})} \tag{2}$$

$$SHF_{System\ Capacity} = \left(\frac{\dot{Q}_{CC,\ Sens.}}{\dot{Q}_{CC,total}} \right) \cong \frac{(\dot{m}_a c_p)_{S.A.}(T_{Mixed} - T_{S.A.,\ actual})}{(\dot{m}_a c_p)_{S.A.}(T_{Mixed} - T_{S.A.,\ actual}) + \dot{m}_{a,S.A.}\, h_{fg}(W_{Mixed} - W_{S.A.,\ actual})} \tag{3}$$

In these equations, the subscript "$S.A.$" denotes the required supply-air conditions and "$S.A., actual$" the supply-air conditions achievable by the system. The subscript "$Mixed$" represents the properties of the O.A./R.A. mixture. Regardless of the mixing scenario, these mixture properties are determined using the conditions of the constituent streams entering the system, before any conditioning. With the S.A. flow rate of 0.866 kg/s (46.6 m^3/min at standard conditions), the sensible and total cooling loads of the indoor space become 8.54 kW and 10.6 kW, respectively, yielding a sensible heat factor of about 0.80. (Note that the SHF is independent of the supply airflow rate, as long as the prescribed R.A. and S.A. conditions are intact.) However, because of handling a large amount of outside air intake (47%), the SHF for the system load can be considerably less than 0.80, depending on the ambient humidity level, as will be addressed later.

3.2 System Efficiency

To evaluate the cooling system efficiency, two forms of coefficient of performance (COP) are considered to separately account for electrical and thermal energy efficiencies.

$$COP_{electrical} = \frac{\dot{Q}_{CC}}{\dot{E}_{in,\, elect.}} \tag{4}$$

$$COP_{thermal} = \frac{\dot{Q}_{CC}}{\dot{Q}_{DES,\, regen.}} = \frac{\dot{Q}_{CC}}{\left(\dot{m}_{m.a.}\ c_p\right)_{regen.}\left(\Delta T\right)_{H.C.}} \tag{5}$$

The electrical power input, $\dot{E}_{elect.}$, accounts for the total power consumption of the fans and pumps (Figure 1). At a given ambient temperature, the temperature increase across the regeneration heating coil, $(\Delta T)_{H.C.}$, depends on whether the incoming ambient air is preheated by the HX or not.

An overall CHP system efficiency is defined as the ratio of the sum of the net electrical power output ($\dot{W}_{elect.}$) and the cooling capacity ($\dot{Q}_{CC}$) to the rate of fuel input ($\dot{Q}_{fuel}$):

$$\eta_{CHP} = \frac{\dot{E}_{out,\, elect.} + \dot{Q}_{CC}}{\dot{Q}_{fuel}} = \frac{\dot{E}_{out,\, elect.} + \dot{Q}_{DES,\, regen.} \times COP_{thermal}}{\dot{Q}_{fuel}} \tag{6}$$

This equation reflects a positive correlation between η_{CHP} and $COP_{thermal}$. (The definition of Equation (6) is among the commonly used overall efficiency indices [4].)

3.3 DES Performance

The selected desiccant wheel is equally split (50/50) between the process and regeneration air streams and rotates at an optimum or near-optimum speed (18 to 24 RPH). The process-air velocity is maintained between 2.8 m/s and 3 m/s at standard conditions (15°C and 101.039 kPa). The desiccant is assumed to be regenerated at 90°C, representing the regeneration air temperature downstream of the heating coil (Figure 1). This relatively low regeneration temperature improves the heat recovery from the reciprocating engine, leading to a higher overall CHP efficiency. In evaluating the performance of this commercially available unit, the manufacturer's performance software has been used. Figures 2a and 2b provide, respectively, the moisture removal capacity and the process exit temperature for a wide range of inlet conditions.

The pressure drop across the wheel ranges from 290 Pa to 328 Pa for the process side and from 328 to 363 Pa for the regeneration side. Assuming an overall efficiency of 60% for the fans, the maximum fan power input is 0.88 kW for the process side and 1.3 kW for the regeneration. Although the results of Figures 2a and 2b are for the Atlanta elevation (305 m), the same performance curves are valid at other elevations, as long as the mass flow rate and inlet air humidity ratio are in conformity [5]. However, maintaining the same mass flow rate at lower pressures (higher altitudes) increases the fan power consumption because of an increase in the air velocity.

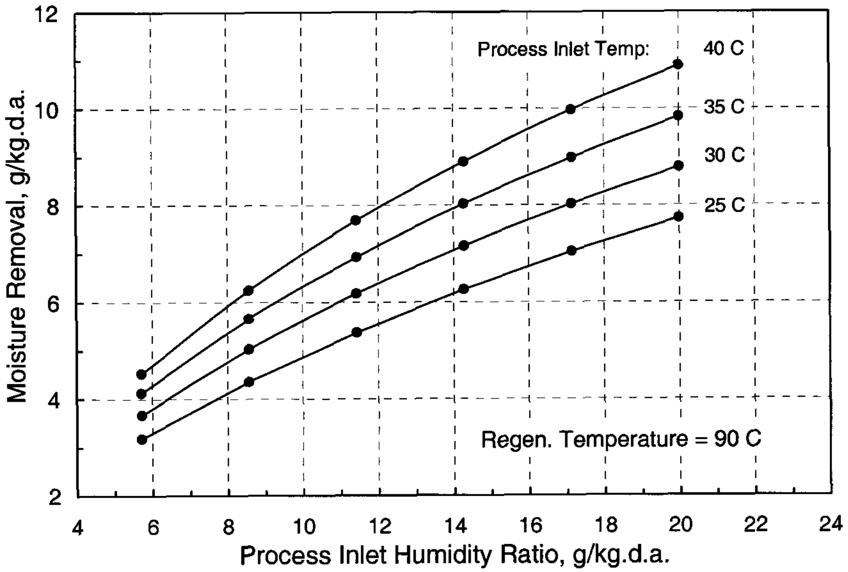

Figure 2a: Desiccant wheel (DES) moisture-removal performance

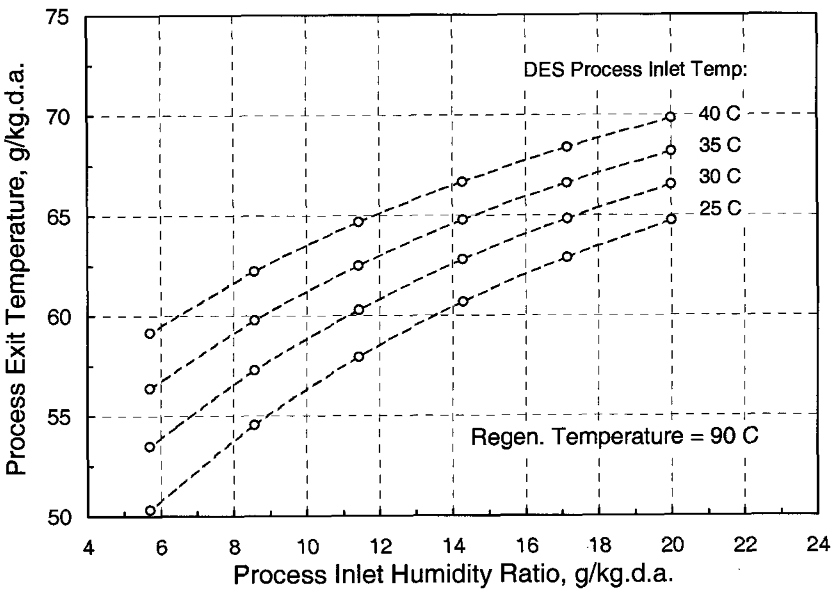

Figure 2b: Desiccant wheel (DES) process outlet temperature

3.4 IEC Performance

The selected IEC unit utilizes a portion of the entering air for the secondary air stream (Figure 1). For performance assessment of this unit, the following wet-bulb based effectiveness definition has been adopted [6]:

$$\varepsilon_{IEC} = 100\left(\frac{T_{IEC,inlet} - T_{IEC,outlet}}{T_{IEC,inlet} - T^{*}_{IEC,inlet}}\right) \tag{7}$$

where $T^{*}_{IEC,inlet}$ is the inlet wet-bulb temperature, while the rest of the variables are dry-bulb temperatures.

Laboratory tests have shown that the wet-bulb effectiveness curves of this unit for a wide range of inlet temperatures virtually collapse for different inlet humidity levels. Shown in Figure 3 are the effectiveness empirical correlations for a commercially available IEC and a prototype unit. For the parametric analysis presented here, the performance of the actual product (model A) and a hypothetical product (model B - a compromise between the commercial and prototype units) are used. The pressure drop across the unit, which is a function of the face velocity, is 622 Pa. With an overall fan efficiency of 60%, this translates into a power input of 1.67 kW for the flow rate considered (1.65 kg/s at the inlet).

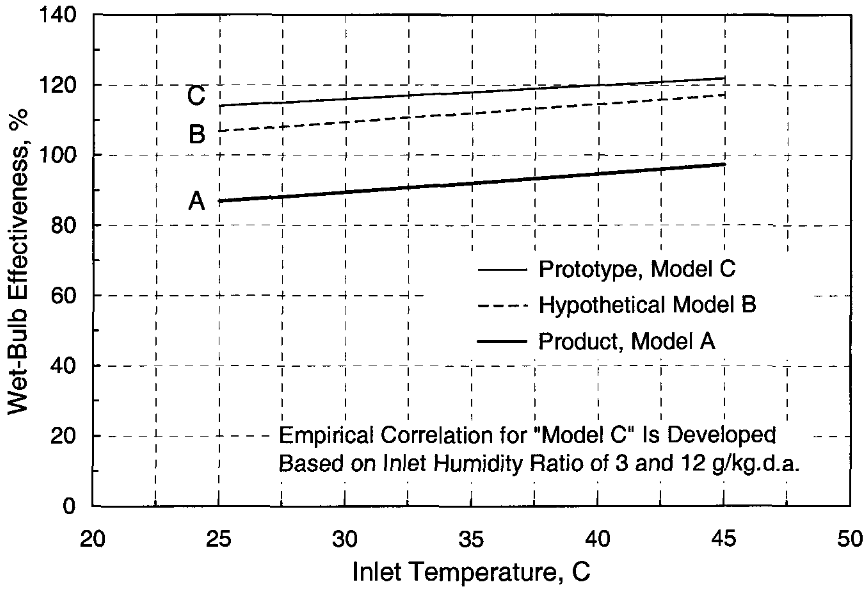

Figure 3: *Empirical correlations for IEC performance*

The properties of the exhaust stream, the secondary stream leaving the IEC, are determined from the following mass- and energy- balance equations:

$$[\dot{m}_a(1+W)]_{IEC,inlet} + \dot{m}_w = [\dot{m}_a(1+W)]_{IEC,outlet} + [\dot{m}_a(1+W)]_{IEC,exhaust} \tag{8}$$

$$(\dot{m}_a h)_{IEC,inlet} + \dot{m}_w h_w = (\dot{m}_a h)_{IEC,outlet} + (\dot{m}_a h)_{IEC,exhaust} \tag{9}$$

The humidity ratio at the exhaust stream is determined based on the notion that the air is virtually saturated (~ 100% relative humidity). The specific enthalpy of moist air is a function of dry-bulb temperature and humidity ratio, i.e., $h = h(T,W)$.

3.5 Heat Exchanger

The effectiveness of the heat exchanger for a balanced-flow mode, which is the case in this study, is determined by the following equation:

$$\varepsilon_{HX} = 100\left(\frac{T_{HX,inlet} - T_{HX,outlet}}{T_{HX,inlet} - T'_{HX,inlet}}\right) \tag{10}$$

In this equation, $T'_{HX,inlet}$ represents the inlet temperature of the HX secondary air. The temperature $T'_{HX,inlet}$ is equal to $T_{IEC,exhaust}$, $T_{amb.}$, or T_{Mixed} when the IEC exhaust, ambient air, or a mixture of the two is used for post cooling. The HX effectiveness is assumed to be 70%. Note that when the IEC exhaust or a mixture with ambient air is used for DES post cooling, there are two unknown variables: $T'_{HX,inlet}$ and $T_{HX,outlet}$. This necessitates simultaneous solution of Equations (8) and (10).

In this study, the effect of leakage between the two streams of the heat exchanger is neglected. Although this is a reasonable assumption for well-designed fixed-core heat exchangers, it may not be valid for rotary types, depending on the design characteristics and rotational speed [7].

4 RESULTS AND DISCUSSION

This section presents the results of the analytical study, encompassing system performance, parametric evaluation of system configuration and operating modes, engine sizing, and water consumption.

4.1 Cooling System Performance

Figure 4 illustrates the processes involved in the desiccant cooling. The supply air flows at a rate of approximately 0.866 kg/s and is composed of 53% return air and 47% outside air, as dictated by the exhaust-to-supply flow ratio of the IEC unit. In the system of Figure 4, the R.A. is mixed with O.A. at the inlet of the desiccant wheel, and the heat exchanger only performs as a DES post cooler (no regeneration preheating). (In the absence of relief air, the return airflow rate equals that of the supply air.) With a DEC downstream of the IEC, the humidity ratio of the supply air is adjusted to maintain the design-level humidity while cooler supply air is achieved. The actual supply air temperature in Figure 4 is less than the required supply temperature of 15°C (indicating excessive sensible cooling capacity), while the humidity ratio is right on the target. The stream exhausted from the IEC unit is virtually saturated. (It should be noted that, in Figure 4, the path of the IEC secondary-flow process, from the HX outlet to IEC exhaust, is symbolic and not representative of the actual path. Furthermore, in the interest of clarity, the effects of fans on the principal states are not reflected.)

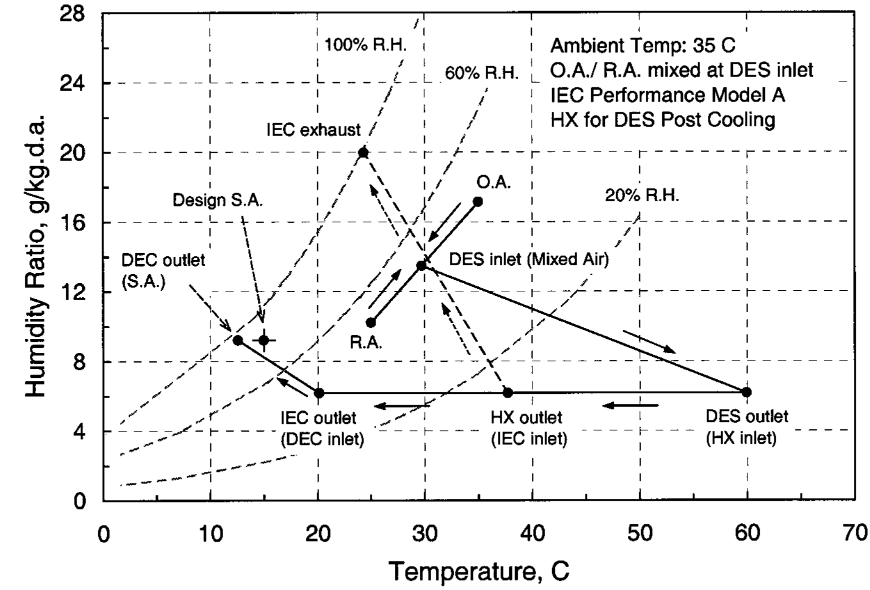

Figure 4: Psychrometric chart for desiccant cooling processes

Figure 5 provides the system cooling capacity and the sensible heat factor as functions of ambient humidity ratio when the ambient temperature is 35°C. The system of Figure 5 incorporates a heat exchanger that cools the dehumidified air leaving the DES and preheats the regeneration air upstream of the heating coil. The mixture of O.A. and R.A. takes place at the DES inlet.

Figure 5 indicates that the total cooling capacity of the system, 26 kW to 27.5 kW, is not highly sensitive to the ambient humidity level, whereas the sensible cooling capacity is quite sensitive, as the SHF trend suggests. In contrast, the desiccant cooling system offers a cooling capacity of more than twice that of standard air-cooled air-conditioning systems at about the same supply air flow rate for a wide range of ambient humidity ratios. Furthermore, while the typical SHF for conventional systems is between 0.7 and 0.8, the desiccant cooling system offers a lower value (higher latent capacity) at high ambient humidity levels. It is important to note that the performance of the conventional system is based on 17% outside air intake, compared to 47% for the desiccant cooling system. These findings point to the effectiveness of the proposed system in hot and humid climates. In addition, for the cooling capacities shown in Figure 5, a conventional system would typically require supply air flow rate of about 1.6 kg/s, which is 87% higher than that of the desiccant cooling system. Even with a less S.A. flow rate, the 47% O.A. intake for the desiccant cooling unit translates into an amount of approximately 50% higher than that of the conventional system. The maximum cooling capacity registered for this desiccant cooling system is about 28.2 kW, which occurs an ambient temperature of 40°C and humidity ratio of 20 g/kg of dry air with the system of Figure 5.

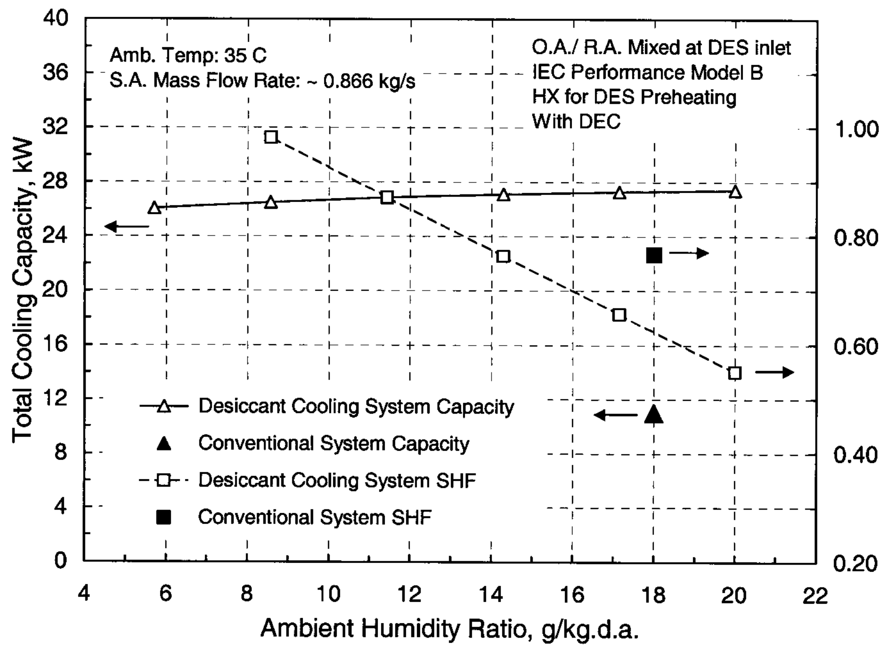

Figure 5: *Desiccant cooling performance*

In Figure 6, the electrical and thermal energy COP of the system of Figure 5 is plotted as a function of ambient temperature for two humidity levels: 17.14 and 8.57 g/kg of dry air. At 35°C ambient temperature, the desiccant cooling system operates at an electrical energy COP of about 5.3, which is about 60% more efficient than typical conventional systems, despite handling a larger amount of outside air. The thermal energy COP of the desiccant cooling system improves with increasing ambient temperature and humidity ratio. The impact of the ambient conditions on the electrical energy COP is much less pronounced. The weak positive correlation observed between the electrical COP and the ambient temperature stems from the total cooling capacity trend seen in Figure 5. Contrary to the desiccant cooling system, the electrical COP of conventional systems is adversely affected by the increasing ambient temperature.

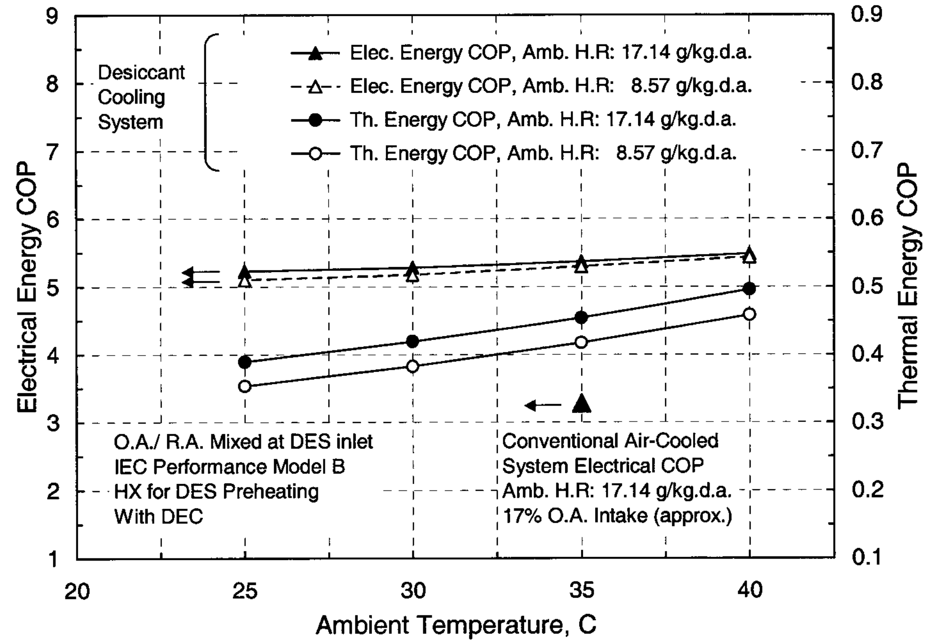

Figure 6: *Cooling system efficiency evaluation*

4.2 Parametric Evaluations

4.2.1 Evaporative Cooling

Figures 7 and 8 show the impact of the IEC effectiveness on the overall system performance, with and without incorporating a DEC unit, for a wide range of ambient humidity ratios at 35°C. Shown in these figures are the cooling capacity-to-load ratios for latent and sensible cooling and the thermal energy COP. The systems of Figures 7 and 8 incorporate performance models A and B, respectively, for the IEC effectiveness. Other attributes of the systems are identical: incorporating a heat exchanger for desiccant post cooling and mixing of O.A. and R.A. streams at the inlet of the desiccant wheel.

As seen in Figure 7, in the absence of a DEC, the system is incapable of meeting the sensible load, although it provides excessive latent (dehumidification) capacity for the entire range of ambient humidity ratios considered. However, by converting the latent load, the DEC boosts the sensible capacity to an extent that is sufficient at even high ambient humidity levels. In contrast, when the more effective IEC model B is used (Figure 8), a noticeably improved sensible cooling performance is observed, although a DEC will still be required at high humidity levels. An improvement with the thermal energy COP is also registered with the inclusion of a DEC. (Note that the thermal energy COP curves for the scenarios involving DEC virtually collapse.)

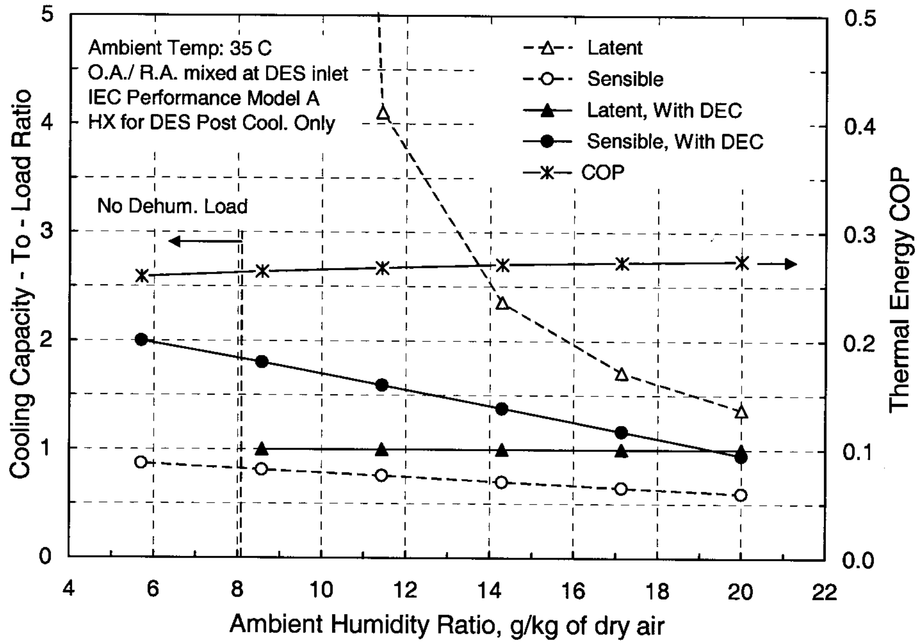

Figure 7: *System performance with "model A" for IEC*

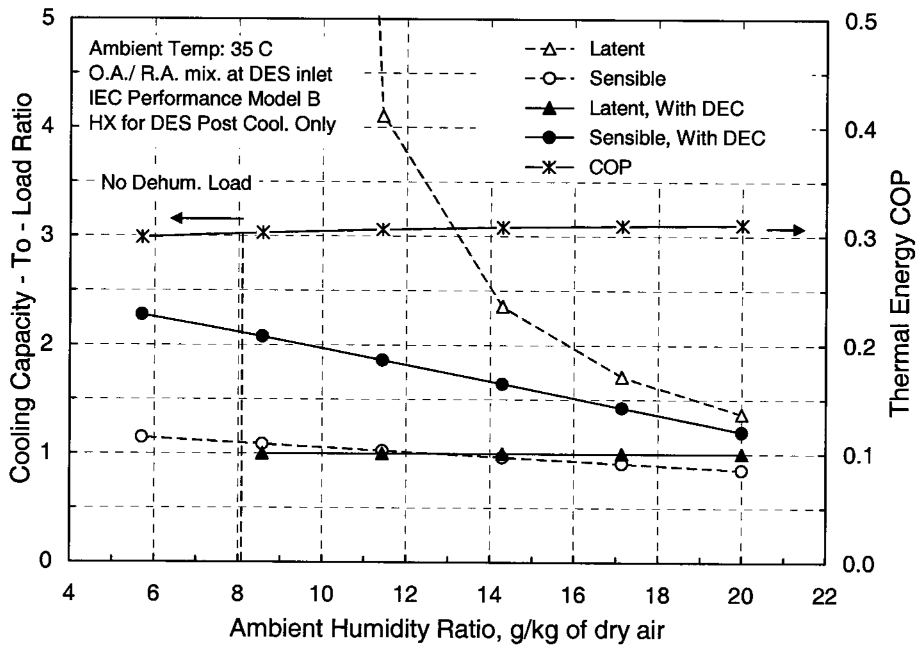

Figure 8: *System performance with "model B" for IEC*

4.2.2 Heat Exchanger

Figure 9 examines the performance of the system of Figure 7 under the same operating conditions but in the absence of the heat exchanger. A comparison of Figures 7 and 9 reveals the adverse impact of eliminating this component (HX). Without the heat exchanger, the insufficiency of the sensible cooling capacity at high ambient humidity levels (in excess of about 16 g/kg of dry air) when the ambient temperature is 35°C is evident. The HX effectiveness is assumed to be 70%.

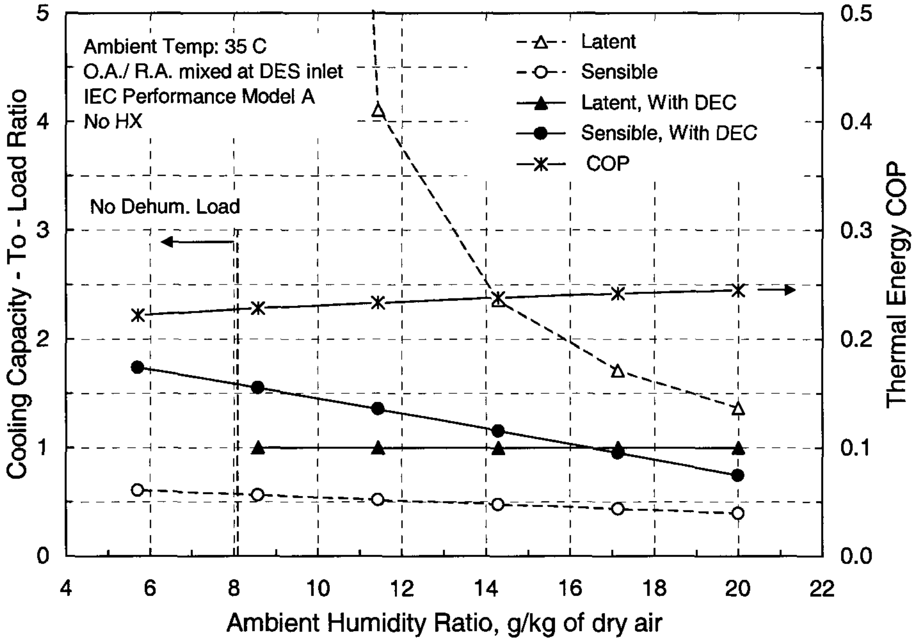

Figure 9: *System performance with no HX, "model A" for IEC*

For further insights on the significance of operating a heat exchanger, the cooling system performance is evaluated under two scenarios with respect to the heat exchanger applications. These scenarios are (1) desiccant post cooling via the exhaust stream of the IEC unit and (2) preheating the regeneration air drawn from the ambient, concurrent with desiccant post cooling. Figure 10 illustrates variation of the normalized performance parameters with the ambient temperature when the corresponding humidity ratio is 17.14 g/kg of dry air. As seen in this figure, although the degradation of the sensible cooling performance resulting from the second scenario is relatively insignificant, its positive impact on the thermal energy consumption is rather well pronounced. By allocating the heat exchanger to preheating of the regeneration air, about 40% to 45% reduction in the thermal energy consumption is realized as a result of the COP improvement. This finding has a profound implication regarding fuel utilization/efficiency for both CHP and stand-alone applications. Additional discussion on this topic is provided when the operational mode involving the mixture of O.A. and R.A. streams is addressed.

4.2.3 O.A./R.A. Mixture

The preceding results and discussions were based on the assumption that the mixing of the R.A. and O.A. streams takes place at the DES inlet.

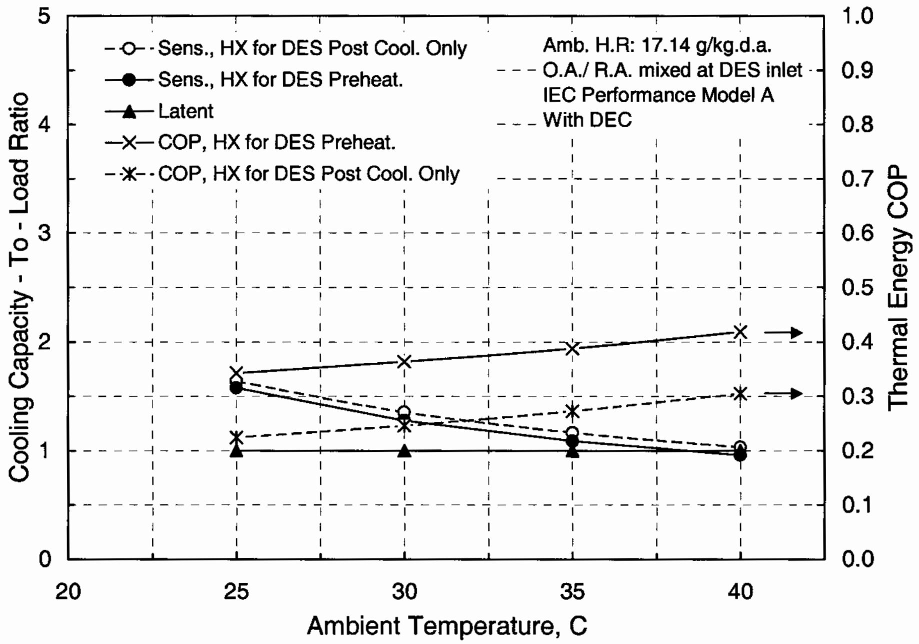

Figure 10: *Impact of HX operating mode*

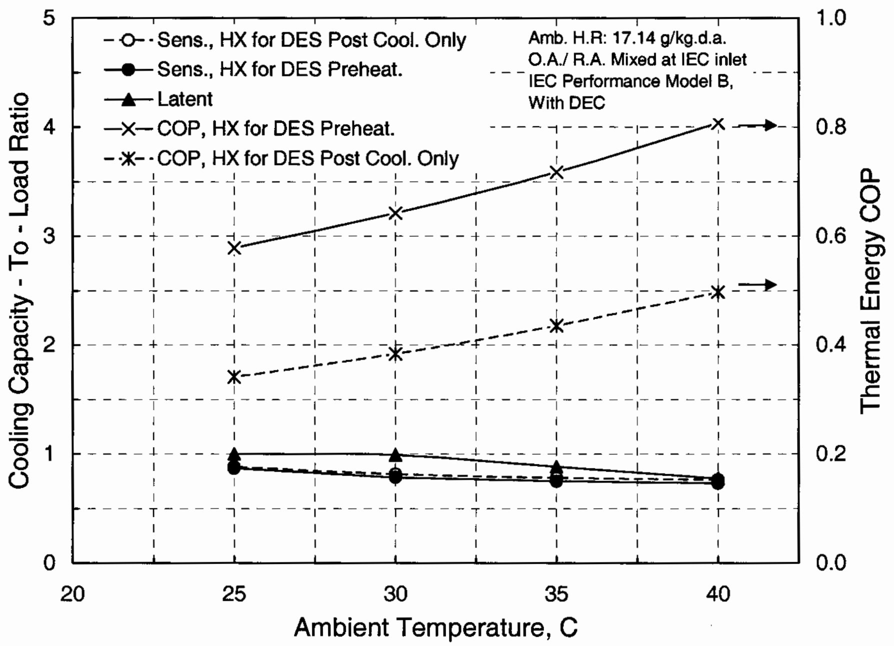

Figure 11: *System performance with O.A./R.A. mixing at IEC inlet*

Figure 11 examines the impact of moving the mixing point to the inlet of the IEC unit on the system performance. The results of Figure 11 are discussed in light of the two alternatives considered for application of the heat exchanger (i.e., DES post cooling and regeneration preheating vs. only DES post cooling.) This figure suggests that the latent and sensible cooling capacities become insufficient at high ambient temperatures when the humidity ratio is 17.14 g/kg of dry air-in spite of using a higher performing IEC unit (model B). The heat exchanger operating mode virtually has no effect on the cooling capacities but has a significant impact on the thermal energy COP - a notion consistent with the earlier finding. An evaluation of the COP trends in Figures 10 and 11 highlights the substantially higher energy efficiency of the scenario involving mixing of O.A. and R.A. at the IEC inlet.

Figures 10 and 11 have demonstrated that one of the two O.A./R.A. mixing scenarios yields excess cooling capacities, while the other one degrades the performance but offers a higher thermal efficiency. This observation points to the need for allowing a combination of the two mixing approaches for an effective and efficient indoor quality control, without imposing an excessive system on/off frequency.

4.3 Engine Sizing and Overall CHP Performance

An appropriate engine size with respect to the aforementioned desiccant cooling system depends on whether an electrical- or thermal-load following CHP model is to be adopted. For the latter case, the maximum required thermal energy for regeneration of the desiccant material directly dictates the engine size, provided that this required heat represents the annual peak thermal demand. The preceding discussions on the heat exchanger revealed the importance of desiccant regeneration preheating in reduction of the thermal energy input. With this feature in place, the required heat input ranges from 55 kW to 73 kW for the range of ambient conditions considered in this study. Assuming a heat recovery of 1.2 kWh per kWh of output electricity (66% overall engine thermal efficiency) and an electrical power generation efficiency of 30%, a 60-kW engine is required, at the minimum. A 50-kW engine would be adequate if an overall thermal efficiency of the engine increases to 73%. Further system downsizing can be realized by improving the thermal COP of the desiccant cooling system in the thermal-load-following model, as can be seen by examining Equation (6).

Engine sizing is also of an economic decision that has to be based on the annual system performance and the life-cycle cost. Therefore, consideration of an even smaller engine size may be appropriate, depending on the full- and part-load frequencies/operational duration. In this case, use of auxiliary burners for regeneration would be imperative. This points to the importance of building energy simulation models in optimum selection and sizing of CHP systems.

In the thermal-following mode, the overall efficiency of the CHP system incorporating a 60-kW engine is determined (Equation 6) to vary from about 42.5% to 44.5% at an ambient temperature of 35°C for the range of the humidity ratio considered. At the lower ambient temperature of 25°C, the CHP efficiency drops by less than 2 percentile points. These efficiency estimates are based on the overall cooling capacities at the given conditions. The actual efficiencies will depend upon the actual operating loads.

For the case of the electrical-load following CHP model (i.e., sizing the engine to meet the entire or a part of the building electrical peak load), the thermal output may not meet the thermal demand of the desiccant cooling system. Therefore, for this model, auxiliary burners may be needed. A study performed on a commercial building indicated that thermal-load following CHP systems tend to be more efficient than electrical-load following ones but require larger on-site power generators [8,9].

4.4 Water Consumption

Water consumption of desiccant cooling systems can conceivably be a major concern, especially in arid regions where water resource management is critical. Although electric, air-cooled, conventional cooling systems do not use water at the sites, their electrical energy input requires water consumption at the power generation plants. A study has shown that the national aggregate water use, combining thermoelectric and hydroelectric power plants, is about 7.6 L/kWh [10]. Based on this finding, Figure 12 provides a comparison between the total water use of the evaporative cooling system studied here and a conventional one. The water usage in this figure is normalized based on the total system cooling capacity. For the conventional system, an average COP of 3.5 is assumed.

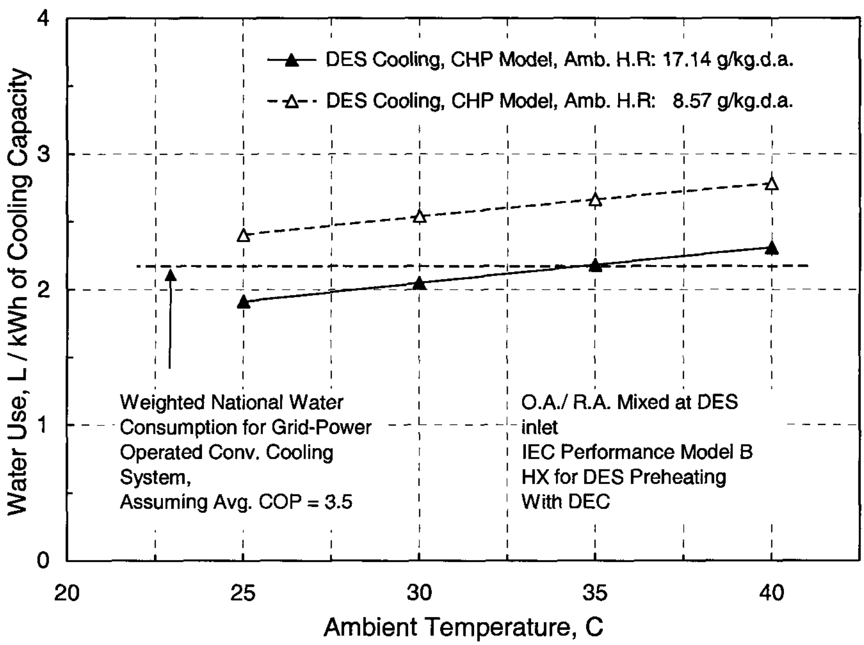

Figure 12: System water consumption

Referring to Figure 12, the water consumption of the desiccant cooling system increases with temperature but decreases with the increasing humidity level. At the ambient conditions of 35°C and 17.14 g/kg, the water use of this system equals that of the conventional. The desiccant cooling water consumption reported in Figure 12 is only associated with the evaporative cooling units, IEC and DEC. No water use is attributed to the electrical energy input of this system, assuming the required electricity is drawn from the on-site generator, which does not consume water.

Future studies will address potential benefits of desiccant/conventional cooling hybrid systems with respect to energy and water efficiency. An alternative strategy is integration of a conventional cooling system downstream of the IEC unit on the exhaust side to recover a portion of the exhaust water and to provide additional cooling. This option can be particularly beneficial when the on-site power generation exceeds the building electrical demand and export of electricity to the local grid is not economically favorable.

5 CONCLUSIONS

This paper presented and discussed the performance results of a thermally activated desiccant cooling system in a CHP application incorporating a reciprocating IC engine. The baseline cooling system consisted of a desiccant wheel, a heat exchanger, an indirect evaporative cooler, and a direct evaporative cooler. Implementation of the direct evaporative cooling enabled conversion of excess latent capacity to sensible, whenever necessary. As one of the main objectives, a parametric analysis was performed to examine the performance impact of various operating scenarios with regard to the subsystems.

It was demonstrated that the desiccant cooling system not only could handle relatively high system latent loads at high ventilation rates, they could also concurrently satisfy the sensible loads for a wide range of climatic conditions. Of particular interest was demonstrating the effectiveness of the system in humid and dry climates despite its lower supply-air flow rate and higher amount of outside air intake compared to conventional systems. The maximum cooling capacity of the desiccant cooling system was about 28.5 kW, occurring at high temperature and humidity conditions. Achieving this capacity required installation of a 60-kW engine for a thermal-load-following CHP model. The electrical COP of the system was determined to be greater than 5, compared to less than 3.5 for standard conventional systems. Furthermore, the COP was shown to slightly improve with the increasing ambient temperature, a notion contrary to the performance behavior of conventional systems.

The results of the parametric analyses were in favor of incorporating a heat exchanger to simultaneously accomplish desiccant post cooling and regeneration preheating. The system thermal efficiency improvement observed with this operating mode is of great importance with respect to CHP fuel utilization. The system performance showed a strong sensitivity to the method of mixing outside air with the return air. For nearly the entire domain of ambient conditions, the cooling capacity of the system exceeded the load when the mixture occurred at the inlet of the desiccant wheel. Moving the mixing point to the immediate upstream of the indirect evaporative cooler degraded the cooling capacity to levels below the requirements at even moderate temperatures, but it offered a significantly higher thermal COP (and a higher CHP efficiency, consequently). A discussion was made in promotion of a combination of the two mixing strategies for optimum operation of the system.

Finally, the water consumption of the desiccant cooling system was addressed and discussed in conjunction with the notion of indirect water use associated with the central plant electricity in non-CHP applications. Unlike the common perceptions, the water use of the desiccant cooling system, in the context of CHP, was not substantially different from the indirect water use of conventional electric systems.

The current study is a prelude to a more comprehensive future research in this area. Among the topics under consideration include (1) annual performance evaluation of the desiccant cooling system under different climatic conditions and applications and (2) exploration of desiccant/conventional cooling system in pursuit of maximum energy and water efficiency.

***Table 5.1**: Modes of Desiccant Cooling System Operation*

Device	*Status*	*Function*	*Remarks*
DES	*Yes*	*Dehumidification*	*Deactivated and bypassed at low ambient humidity levels.*
HX	*Yes*	*DES post cooling only*	*A combination of IEC exhaust and O.A. used for DES post cooling.*
		DES post cooling and regeneration preheating	*Only O.A. for DES regeneration, IEC exhaust not used.*
	No	--------	*No role for IEC exhaust stream.*
IEC	*Yes*	*Sensible cooling*	*Exhaust stream may be used for DES post cooling.* *Two performance curves considered for analysis: Models A and B.*
DEC	*Yes*	*Sensible/latent capacity adjustment*	*For fine-tuning of S.A. temperature*
	No	-------	
R.A. Circulation	*Yes*	*Upstream of DES*	*A combination of two may be necessary for optimum operation.*
		Upstream of IEC	

5.1 Nomenclature

c_p Specific heat under constant pressure, kJ/kg K

COP Coefficient of performance for cooling

$\dot{E}_{in,\ elect.}$ Electrical power input, kW

$\dot{E}_{out,\ elect.}$ Electrical power output, kW

h	Specific enthalpy of moist air, kJ/kg of dry air
h_{fg}	Enthalpy of vaporization, kJ/kg
h_w	Enthalpy of water vapor, kJ/kg
$\dot{m}_a$	Mass flow rate of dry air, kg/s
$\dot{Q}_{CC}$	Cooling capacity, kW
$\dot{Q}_{CL}$	Cooling load, kW
$\dot{Q}_{DES}$	Desiccant regeneration heat input, kW
T	Dry-bulb temperature, C
T^*	Wet-bulb temperature, C
T'	Secondary-air dry-bulb temperature, C
W	Humidity ratio, g/kg dry air
$(\Delta T)_{H.C.}$	Temperature increase across regeneration heating coil, C
ε_{HX}	Effectiveness of heat exchanger
ε_{IEC}	Effectiveness of indirect evaporative cooler
η_{CHP}	Overall efficiency of CHP system

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