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ADVANCED DEVELOPMENT AND MARKET PENETRATION OF DESICCANT-BASED AIR-CONDITIONING SYSTEMS

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EXECUTIVE SUMMARY

In 1998, a cooperative research and development agreement was signed between UT-Battelle under its U.S. Department of Energy contract and Fresh Air Solutions by Engelhard/ICC to develop the enabling technologies which permit the widespread field application and successful commercial development of non-CFC comfort cooling systems based on advanced desiccant materials and desiccant air conditioning methodologies.

Desiccant air conditioning systems can be used as alternatives for conventional air conditioning equipment in any commercial or residential building. A recent breakthrough in desiccant materials technology and the creation of new markets by Indoor Air Quality (IAQ) issues make desiccant-based air conditioning equipment practical for many space-conditioning applications. Barriers to broad acceptance of this technology are: (1) absence of computerized algorithms that allow convenient incorporation of desiccant modules in heating, ventilating, and air conditioning (HVAC) simulation programs used by evaluation and application engineers; (2) lack of suitable "metrics" to comparatively evaluate desiccant-based system performance against conventional systems; and (3) a perception of inefficiency from earlier research on desiccant space-conditioning systems. The project consisted of three main phases to address these issues.

PHASE 1: PERFORMANCE MODELING

A detailed computer model was developed by Fresh Air Solutions to evaluate design modifications such as wheel speed, regeneration temperature, and air flow rate to the desiccant system. The model was verified against experimental data at UT-Battelle to determine the accuracy of predictions for latent capacity and COP. The results indicate that the model tends to overpredict both measures of performance. This is most likely the result of the model's inability to account for carryover of moisture from the regeneration stream to the process air stream.

PHASE 2: OPTIMIZATION STRATEGIES FOR ENERGY CONSERVATION

A draft metric was evaluated for its accuracy in comparing desiccant system performance. Laboratory test data were used to compare the algorithms in the proposed metric to ensure their accuracy. Included in the metric are standard conditions for testing and calculation procedures for determining the performance for desiccant systems under different modes of operation.

PHASE 3: NEW COMPONENT DEVELOPMENT AND SYSTEM REDESIGN

Work was performed to increase system efficiency through improvements in the regeneration heat transfer process to achieve higher regeneration temperatures and increase the moisture removal from the desiccant. An evaluation was made of the impact of product configuration recommendations, such as wheel speed, air flow rates, and desiccant loading on the performance of the desiccant system. In addition, carryover effects from the regeneration side

to the process air stream were investigated. Finally, the effects of changes in the product configuration were evaluated at off-design conditions that are more representative of actual field usage.

ABSTRACT

Industry needs and goals presently focus on promoting the mainstream acceptance of desiccant systems and overcoming barriers to widespread commercialization, which include:

- developing algorithms that accurately predict the performance characteristics resulting from design improvements;
- establishing a method for system testing and rating criteria that assess product quality and building design capability; and
- developing a clear analysis of the energy use and potential for energy savings when using desiccant systems to lower the humidity in building ventilation and air conditioning systems.

At present, there is no HVAC industry accepted basis for adequately comparing the moisture removal efficiency of a desiccant system to that of vapor compression equipment. With the introduction of ASHRAE standard 62, ventilation rates are increased by a factor of 4 (from 5 to 20 cubic feet per minute) for most occupancy classifications. Under humid outdoor conditions, conventional electric vapor compression cooling systems will not be able to remove the moisture without first cooling the air below an acceptable comfort level and then reheating. This method would result in excessive energy requirements and higher utility demand charges. Utilizing desiccant systems to pretreat the air and remove moisture upstream of the conventional cooling system would enable conventional vapor compression systems to meet the new operating requirements without incurring severe energy penalties.

1. INTRODUCTION

Desiccant systems are growing in popularity because of their ability to independently control humidity levels (latent loads) in buildings, thereby allowing conventional air-conditioning systems to primarily control temperature (sensible loads). In hospital operating rooms as an extreme example of a critical building heating, cooling and ventilation application, humidity and temperature can be controlled separately, allowing the surgeon and operating staff to work comfortably under intense lighting while wearing several layers of protective clothing. There is no need to compromise conditions in one operating suite because of the demands of a procedure being performed in another served by the same system. On a less dramatic scale, this same freedom from compromised comfort and indoor air quality can be extended to offices, schools, nursing homes, theaters, retail stores, assembly halls, etc.

ASHRAE Standard 62-1989 covering ***Ventilation for Acceptable Indoor Air Quality*** now requires much more fresh air for building ventilation than the previous standard. This requirement for increased levels of ventilation air has dramatically increased the *latent* over the *sensible* portion of building HVAC loads. New ASHRAE weather data has been introduced in the ***1997 ASHRAE Handbook—Fundamentals*** which is used by engineers to size and select HVAC equipment for building air-conditioning. This new data identifies regional design dew point temperatures (peak outside air humidity conditions) in addition to peak sensible or temperature conditions that building air-conditioning equipment should be sized to accommodate. The large humidity loads from outside air in non-arid climates is now readily recognized and quantified as design conditions. Outdoor air pre-conditioning (primarily dehumidification) technologies like desiccant systems are being used in combination with conventional HVAC operations down stream in order to deliver the required amount of fresh air and control indoor humidity.

In a recent ASHRAE Journal publication that dealt exclusively with ventilation air load requirements, it was shown that latent loads are always higher than sensible loads for every region in the United States with the exception of desert climates (Harriman, et. al., 1997). It is also true that better thermal insulation in buildings, high-efficiency lighting, and more efficient, plug-load appliances have reduced the electrical power use per unit area (watt/ft^2) in buildings resulting in a proportional increase in the importance of latent load over sensible. Desiccant systems remove moisture from air directly rather than cooling air below its dew point. In a typical desiccant system, the moisture (latent load) is removed by the desiccant, and then the temperature (sensible load) of the dry air is reduced to the desired comfort conditions by conventional sensible coolers (e.g., heat exchangers, cooling coils, evaporative coolers, etc.). In this manner the latent and sensible loads are handled separately and more efficiently by hardware components specifically designed to remove that load.

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Most thermostatically controlled, conventional air conditioners can only dehumidify when there is a corresponding sensible load. Even equipment that is correctly sized to handle the combined latent and sensible load, will cycle on and off, allowing humidity to float out of control at "off design," "low load" conditions. This inability to control humidity will occur for the majority of the cooling season. But, desiccant dehumidification equipment when combined with conventional cooling systems permit improved control at off-design conditions resulting in controlled humidity levels at preferred temperatures. The energy required to reactive the desiccant material of a desiccant system and permit it to remove the latent air conditioning load may be supplied by building exhaust air or, in the case of thermally-regenerated active-desiccant systems, by natural gas, propane, hot water, steam and/or waste heat reclaim instead of the electricity used in conventional vapor-compression equipment. This allows dramatic improvements in energy savings or utility load shifting. Existing gas supplies, recovered or solar-generated hot water, low-pressure steam, or steam generated at part-load conditions are usually adequate for active system desiccant regeneration.

2. PHASE 1: PERFORMANCE MODELING

A proprietary desiccant system modeling program provided by Fresh Air Solutions, Inc. was used to obtain comparisons of modeled system performance to experimentally measured performance. The Participant uses this Windows™ based modeling tool primarily as a convenient, in-house method to estimate system performance at design and off-design conditions. It is also used as a marketing and sales tool by Fresh Air Solutions' sales engineers to easily demonstrate equipment performance to prospective customers.

This is a very convenient, versatile program which can:

- Simulate pre- and post-cooling of the process air steam by other HVAC components,
- Simulate several options and temperature regions for thermal regeneration of the desiccant,
- Allow optimization of the system for latent or total (sensible + latent) cooling,
- Be set to a design load or off-design load run mode,
- Simulate use- or non-use of an evaporative cooler on the regeneration side,
- Provide for outside (ambient) and/or building return air at any flow rate, temperature, and humidity condition at the process inlet,
- Provide for outside (ambient) and/or building exhaust air at any flow rate, temperature, and humidity condition at the regeneration inlet,
- Allow for different weight % loadings of desiccant on the desiccant wheel,
- Allow for different desiccant wheel and heat exchanger wheel rotation rates and optimize these rotation rates for any set of conditions,
- Accommodate a percentage of regeneration air by-pass,
- Simulate all the standard wheel sizes manufactured by Fresh Air Solutions,
- Show intermediate conditions between each major system component,
- Calculate the thermal energy required for desiccant regeneration, and, of course,
- Calculate the process air outlet conditions.

All of the input data needed for these calculations are accessible in pull-down menus or parameter "windows" superimposed on a system graphic. A printing option allows generation of hard copy results. In this report, output from this model was checked against experimentally generated laboratory results and compared to predictions made with empirically-generated algorithms generated from curve-fits of ORNL laboratory results.

3. PHASE 2: OPTIMIZATION STRATEGIES FOR ENERGY CONSERVATION

Properly applied, desiccant ventilation air pretreatment systems have the potential to reduce building energy consumption, decrease greenhouse CO_2 emissions to the atmosphere, and significantly improve the indoor air quality experienced by building occupants. One of the impediments to mainstream acceptance of desiccant air conditioning products has been the absence of a uniform methodology for laboratory testing and product rating of desiccant-based systems. An industry-accepted method of test (MOT) and product certification rating system for desiccant products would allow consumer comparisons of products from different manufacturers, rate the relative performance of these systems to other-more conventional products, and facilitate integration of desiccant-based options with conventional building heating, ventilating, and air conditioning (HVAC) equipment.

A draft metric was evaluated for its ability to compare desiccant system performance. A method of testing is included in the proposed metric which specifies standard rating conditions and methods for calculating the performance and efficiency. Several modes of operation for the desiccant system are prescribed: 1) ventilation/exhaust; 2) ventilation; 3) recirculation/exhaust; and 4) recirculation. The metric will apply to packaged, air-to-air, dehumidification systems that:

a) use desiccants for the dehumidification of process air,
b) utilize heated air for the regeneration of the desiccant material,
c) include air-moving means for the process air
d) include air-moving means for the regeneration air,
e) may include ancillary components that provide pre-cooling and/or post-cooling of the process air,
f) may include ancillary components that provide heating of the process air, and
g) may include ancillary components that provide humidification of the process air.

The metric is presented in the form of a test standard with recommendations for an industry-sponsored rating standard for equipment comparisons.

The performance of the desiccant system is determined for one or more of the operating modes in terms of the following:

a) total cooling capacity,
b) sensible cooling capacity,
c) latent cooling capacity,
d) fuel input rate for regeneration, and
e) electric input rate for regeneration, motors, and ancillary equipment.

These performance measures are in terms used for rating air conditioning systems and are familiar to HVAC designers. For some dehumidification systems, the sensible cooling capacity may actually be negative due to the heat added to the process air stream from the dehumidification process. Overall efficiency terms, such as COP and EER, are not included as rating parameters due to the difficulty presented by having a combination of fuel and electric inputs.

Initial comparisons against laboratory data at the conditions specified in the metric revealed an error in the power measurement for the fans. This enabled the data to be corrected that resulted in a higher efficiency for the experimental desiccant system than previously reported. Thus, the metric accomplished one of it purposes in identifying errors in system performance.

4. PHASE 3: NEW COMPONENT DEVELOPMENT AND SYSTEM REDESIGN

4.1 Phase 3A: Characterization of Heat Recovery Wheels in Thermally Regenerated Desiccant Systems Utilizing Evaporative Cooling

The dynamics of thermal regeneration via a rotating wheel coupled with evaporative cooling in a gas-fired, desiccant, dehumidification system are explored in relation to system efficiency and capacity. Implementation of these features reduces the sensible cooling load of the supply air, but also diminishes the dehumidification (latent) capacity of the system due to moisture transfer to the dehumidified air. The conflicting nature of these attributes necessitates examination of the system performance parameters with respect to the rotational speed of the thermal recovery wheel and the effect of the evaporative cooling.

The performance parameters considered in this study are dehumidification (latent) capacity, net capacity, COP for the latent capacity, and an overall COP based on the net capacity. By incorporating the net effect of the latent capacity and the sensible load, the net capacity and the related COP offer an appropriate means for a comprehensive examination of the system performance.

The results of this study indicate that, for the inlet air conditions considered, the thermal regenerator in conjunction with evaporative cooling of regeneration air leads to enhancement of the net capacity and the energy efficiency of the system. The significance of this enhancement varies with the wheel speed and the inlet air conditions.

EXPERIMENTAL PLAN

As shown in Figure 1, two air streams, process and regeneration, are involved in the desiccant dehumidification process. As the process air flows through the desiccant wheel, the moisture is removed from the air stream. The desiccant is restored to its dry state by exposure to the heated regeneration air stream as it rotates. Due to the heat transfer from the regeneration side via the rotating wheel and to the latent heat effect, the process air stream is heated during the dehumidification process that necessitates post cooling for comfort. To reduce the energy consumption associated with the post cooling, the process air exiting the desiccant wheel is partially cooled via a heat recovery wheel, which turns in the opposite direction of the desiccant wheel. As the heat recovery wheel rotates, heat is transferred from the process stream to the incoming regeneration air. Consequently, the relatively cool incoming regeneration air is preheated which reduces the energy required to heat the regeneration air stream. The effectiveness of the wheel is dependent on its rotational speed. In many systems, a direct evaporative cooler is also incorporated on the regeneration side, upstream of the heat recovery wheel (Figure 1). When the evaporative cooling unit is activated, the incoming regeneration air is cooled to a temperature at or near the wet bulb temperature before entering the recovery wheel. This improves the cooling performance but diminishes the preheating effect of the wheel. The processes involved in the desiccant dehumidification system incorporating a thermal recovery wheel and evaporative cooling for the regeneration air are illustrated via the psychrometric diagram in Figure 2.

Heat pipes may also be used in desiccant systems in lieu of thermal recovery wheels. Unlike the rotary regenerators whose speed can be fine-tuned, the heat pipes are passive and do not offer any controllability. However, thermal recovery wheels result in moisture migration from the regeneration to the process stream due to the air exchange facilitated by the porous wheel matrix. As shown in Figure 2, the humidity ratio of the process air increases and that of the regeneration air decreases (paths p2-p3 and r2-r3), both of which affect the overall performance of the system. This effect is further magnified when evaporative cooling is used to condition the incoming regeneration air.

The performance of this system, from the standpoint of dehumidification capacity and energy efficiency, depends upon a number of factors including the type of desiccant material, regeneration temperature set point, process and regeneration air flow rates, and the rotational speeds and physical characteristic of the two wheels. Optimum operation of these systems necessitates exploring the complex inter-relationships among these variables. Papers by Vineyard et al. (2000) and Jalalzadeh-Azar et al. (2000) presented parametric analyses relating performance of commercially available, rotary, gas-fired, desiccant systems to a number of operational variables. Van den Bulck et al. (1986) analytically evaluated the optimum values of the desiccant wheel rotational speed and the regeneration mass flow rate in minimizing the thermal energy required for regeneration. The main focuses of these studies, however, did not involve the effects of the parameters associated with the thermal regeneration.

This study is intended to characterize the impact of the recovery wheel on the overall system efficiency considering different inlet conditions. In doing so, other design/operating variables including the desiccant wheel speed, the air flow rates, and the regeneration temperature are kept constant at the design values recommended by the manufacturer. The potential benefits and the adverse effects of this evaporative cooler/heat recovery wheel combination on the overall desiccant system performance are also discussed.

NOMENCLATURE

COP	coefficient of performance
$\bar{c}_p$	average specific heat of moist air
$d.a.$	dry air
$\delta \dot{Q}_s$	added sensible cooling load
h	specific enthalpy per unit mass of dry air
$\dot{m}$	mass flow rate
$\dot{Q}$	capacity (latent or net)
$\dot{Q}_{reg.}$	Time rate of thermal energy transfer to regeneration air
T	dry bulb temperature
w	humidity ratio

Subscripts

a	dry air
l	pertaining to latent capacity
p	process air
r	regeneration air
Z	refers to the end point of a constant-temperature dehumidification process (Figure 2).

PERFORMANCE ANALYSIS

a) Regenerator effectiveness

Since, for the system under consideration, the regeneration and process air mass capacity rates are equal [i.e., $(\dot{m}c_p)_{process} \approx (\dot{m}c_p)_{regen.}$], and the residence time of the rotary matrix in both streams is the same (the process and regeneration angles are equal, i.e., 180°), the regenerator is classified as balanced and symmetric (Suryanarayana, 1995). Therefore, the thermal effectiveness, ε, can be defined as:

$$\varepsilon = \frac{h_r 3 - h_r 2}{h_p 2 - h_r 2} \approx \frac{T_r 3 - T_r 2}{T_p 2 - T_r 2} \approx \frac{T_p 2 - T_p 3}{T_p 2 - T_r 2} \tag{1}$$

where h and T , respectively, denote the specific enthalpy and dry bulb temperature of moist air.

b) System efficiency

The energy efficiency indices considered in this study are the coefficient of performance based on the equivalent latent cooling capacity associated with dehumidification, COP_l, and that based on the net system capacity, COP, as defined below:

$$COP_l = \frac{\dot{Q}_l}{\dot{Q}_{reg.}}, \tag{2}$$

$$COP = \frac{\dot{Q}_{net}}{\dot{Q}_{reg.}}. \tag{3}$$

The equivalent latent cooling capacity, $\dot{Q}_l$, is determined as:

$$\dot{Q}_l = \dot{m}_a \left[h_p 1(T_p 1, w_p 1) - h_z(T_p 1, w_p 3) \right] \tag{4}$$

where $h_p 1$ is the specific enthalpy of the moist air at the process inlet, and h_z is the enthalpy evaluated at the conditions corresponding to the temperature of the process inlet and the humidity ratio of the process exit (Figure 2). (More details can be found in ASHRAE 1997.) The net cooling capacity, $\dot{Q}_{net}$, in equation (3) takes into account the sensible load stemming from the deviation of the process exit temperature from that of the inlet as follows:

$$\dot{Q}_{net} = \dot{Q}_l - \delta \dot{Q}_s = \dot{m}_a \left[h_p 1(T_p 1, w_p 1) - h_p 3(T_p 3, w_p 3) \right]. \tag{5}$$

The sensible load can be approximated as:

$$\delta \dot{Q}_s \approx \dot{m}_a \bar{c}_p (T_p 3 - T_p 1) \tag{6}$$

where $\bar{c}_p$ is the average constant-pressure specific heat of the process air. Although $\delta \dot{Q}_s$ is typically positive which has a diminishing effect on $\dot{Q}_{net}$, it can contribute to the net cooling capacity when the evaporative cooling on the regeneration side is activated and the incoming air is sufficiently dry.

In equations (2) and (3), $\dot{Q}_{reg.}$ is the time rate of the thermal energy supply to the regeneration air stream which is required to regenerate the desiccant material. This quantity is approximated as:

$$\dot{Q}_{reg.} \approx \dot{m}_a \bar{c}_p (T_r4 - T_r3) \quad (7)$$

where T_r3 and T_r3 represent the mean temperatures at the inlet and exit of the heating coil.

QUALIFICATION OF EXPERIMENTAL DATA

The uncertainties of the measurements at the inlet and exit test planes primarily stem from the instrumental uncertainties. The inlet properties of the process and regeneration air streams are uniform. The spatial nonuniformity of the properties at the process exit is virtually eliminated due to the presence of mixing baffles and duct fittings upstream of the instrumented plane. However, the dry bulb and dew point temperature measurements made downstream of the desiccant wheel on the process side (state p2 in Figure 2) and those made downstream of the recovery wheel on the regeneration side are highly subject to spatial nonuniformity. (Spatial nonuniformities of the fluid properties for a similar system have been evaluated by Jalalzadeh-Azar, et al, 1998). The main source of such nonuniformities can be attributed to the cyclic operation of the desiccant and regeneration matrices. Since insertion of mixing baffles in the corresponding spaces would interfere with the normal system operation and performance, an analytical approach is used to determine the fluid properties at these locations.

For each set of inlet conditions and the operational status of evaporative cooling unit ("on" or "off"), an experimental run is conducted with the thermal recovery wheel deactivated (0 rpm). The properties of the process air exiting the desiccant wheel (state p2) when the wheel is activated, are set to be equal to those of the process air exiting the system (state p3) when the recovery wheel is not turning for the corresponding inlet conditions and evaporative cooling status. The justification for this assumption is presented later in this paper. Any uncertainty stemming from this assumption is considerably less than that associated with the spatial nonuniformities encountered by direct measurements of these properties. In fact, in some cases, violation of mass balance is observed with the water vapor constituent of the process air when the actual measurements at the aforementioned locations are incorporated into calculations.

Similarly, the temperature data for the regeneration air leaving the heat recovery wheel (state r3) are also subject to a great deal of uncertainty and are disregarded as well. Consideration of these measurements in the energy transfer calculations lead to violation of the first law of thermodynamics. These data are replaced with those resulting from application of an energy balance for the recovery wheel based on the more accurately measured process exit and regeneration inlet temperatures and on the temperature determined for the process air entering the recovery wheel as discussed before.

RESULTS AND DISCUSSIONS

Figures 3 to 7 present the results obtained for different scenarios stemming from consideration of 1) different inlet conditions, 2) "on" vs. "off" operation of the evaporative cooling unit on the regeneration side, and 3) various rotational speeds for the thermal regenerator, 0, 5, and 10 rpm. The regeneration temperature set point and the rotational speed of the desiccant wheel are maintained, respectively, at 88°C and 80 rph in all experimental runs. The conditions at the regeneration inlet are the same as those at the process inlet for any given experimental run.

Figure 3 presents the effectiveness of the thermal recovery wheel for various process and regeneration air inlet conditions considering two speeds for the heat recovery wheel. This figure suggests that the impact of increasing the speed of the wheel from 5 rpm to 10 rpm is greater on the effectiveness than the change in the inlet air temperature from 27°C to 35°C. The effectiveness of the heat recovery wheel decreases with the increasing inlet humidity ratio. However, this variation is well within the prescribed uncertainty range of about ±5% for the effectiveness, and no correlation between this parameter and the inlet humidity ratio can be inferred solely based on these results. The average thermal effectiveness is about 0.78 for a 5 rpm wheel speed and about 0.84 for the one at 10 rpm.

Figures 4a and 4b show the impact of operating the evaporative cooling unit on the humidity ratio of the exiting process air when the recovery wheel is not rotating. Increasing the humidity level of the regeneration air due to activation of the evaporative cooling diminishes the rate of moisture release from the desiccant wheel. This, in turn, results in a decreased dehumidification capacity on the process side, translating into an increase in the process exit humidity ratio. However, the change in the humidity ratio of the process exit is relatively small compared to the amount of the moisture added to the regeneration air. Moisture addition to the incoming regeneration air by evaporative cooling increases with decreasing inlet humidity ratio and with increasing inlet air temperature. Also shown in these figures is the moisture removal capacity for the case without the evaporative cooling. These results are indicative of increasing dehumidification capacity with the humidity ratio at the inlet. Slight increase in the capacity is also observed when the inlet temperature decreases from 35°C to 27°C.

An important implication of the discussion surrounding Figures 4a and 4b is that, for the inlet conditions considered here, the properties of the process air downstream of the desiccant wheel are rather insensitive to moisture change in the regeneration air caused by the rotating thermal recovery wheel. As the thermal recovery wheel rotates into the regeneration air stream, air is exchanged between the regeneration and process streams, leading to a decrease in the humidity level of the regeneration air upstream of the heater and to a corresponding increase in the moisture content of the exiting process air. The highest possible change in the regeneration air humidity ratio is the difference between the average humidity ratio of the two streams and that of the regeneration inlet [i.e, w sub r2-(w sub p2+w sub r2)/2] which is less than 5 g/kg d.a. for the conditions considered here (see Figures 4a and 4b). The actual change in the humidity level of the regeneration air entering the heater is significantly less than this maximum value and, hence, much less significant than that induced by the evaporative cooling. Therefore, the process humidity ratio upstream of the thermal recovery wheel is expected to be rather insensitive to the effect of the wheel rotation. In light of this and since the regeneration temperature set point is fixed, the temperature at this location also should virtually remain unaffected by the wheel rotation. (This is the justification for assuming that, regardless of the recovery wheel speed, 5 or 10 rpm, the properties of the process air downstream of the desiccant wheel are the same as those of the exiting process air when the recovery wheel is deactivated.)

Figures 5a and 5b provide the normalized latent capacity of the system, as a function of inlet air humidity ratio and temperature, wheel rotation rate, and "on" or "off" status of the evaporative cooling component. The normalized values are obtained by dividing the latent capacity (equation 4) by the reference value corresponding to the inlet conditions of 35°C and 11 g/kg d.a. for the case without the evaporative cooling and no rotation of the heat recovery wheel. These figures indicate that 1) the latent capacity diminishes with the increasing recovery wheel speed, 2) activation of the evaporative cooling unit has an adverse effect on the latent capacity especially at the higher inlet dry bulb temperature. In light of the earlier finding that the recovery wheel does not significantly affect the properties of the process air exiting the desiccant wheel, moisture migration from the regeneration air stream to the process air stream is the primary cause of the decrease in the latent capacity as the thermal recovery wheel speed increases (Figures 5a and 5b). The variation in the latent capacity at a given wheel speed is the resultant of two opposing forces, the increasing moisture transfer via the recovery wheel and the decreasing humidity ratio of the exiting process air with the increasing inlet air humidity ratio (Figures 4a and 4b). As seen in Figure 5a, for the case with a wheel speed of 10 rpm without the evaporative cooling, the latent capacity increases as the inlet air humidity ratio increases from about 11 g/kg d.a. to about 14 g/kg d.a., but starts decreasing when the inlet air humidity ratio increases to about 17.5 g/kg d.a. A plausible explanation for the diminished latent capacity at this inlet humidity ratio is the dominance of the moisture migration over the improved dehumidification capacity.

Despite the adverse effect on the latent capacity, the evaporative cooling can significantly reduce the sensible load (equation 6) inherited from desiccant dehumidification and, hence, can improve the net capacity (equation 5) of the system. Figures 6a and 6b are indicative of this potential benefit. (The reference value used for normalization in these figures corresponds to the same inlet conditions used for Figures 5 but with a heat recovery wheel speed of 10 rpm and with the evaporative cooling activated.) Virtually, for all the inlet conditions considered in this study, the scenario corresponding to a wheel speed of 10 rpm with active evaporative cooling offers the highest net capacity. The importance of the rotating recovery wheel is realized by observing the negative values obtained for net capacity when the wheel is shut off. As the humidity level increases at the regeneration inlet, the effectiveness of the evaporative cooling diminishes, particularly at lower inlet temperatures as shown in Figure 6b.

Comparison of Figures 6a and 6b also indicates that, at the higher inlet temperature, a higher net capacity is realized when the evaporative cooling is activated. However, it is important to realize that the net capacity is an overall capacity adjusted for the sensible load needed to cool the process exit air to the temperature of the inlet air and not to a supply temperature required for maintaining indoor comfort level. In reality, the overall equivalent cooling (the sum of dehumidification load and sensible cooling) required to condition the inlet process air to meet the applicable comfort criteria may be less for the case with the lower inlet temperature.

Figure 7a reveals that, at an inlet air temperature of 35°C, the overall COP of the system based on the net capacity improves with the thermal recovery wheel speed and with the operation of evaporative cooling for the regeneration air. At the inlet temperature of 27°C, the improvement on the COP due to the evaporative cooling is not only considerably less pronounced but also can be negatively impacted as seen in Figure 7b for the 10 rpm rotation rate with the inlet humidity ratio of 17.5 g/kg d.a.

The uncertainties in the experimental results are dominated by the systematic uncertainties associated with the instrumentation and the spatial variations of the properties. The random uncertainties are also incorporated in the prescribed uncertainties of the results. Table 1 lists the results and their uncertainties for the experimental run of the reference point used in Figure 6a. (Uncertainty analyses pertaining to desiccant dehumidification are provided by Jalalzadeh-Azar et al., 1996, and Slayzak and Ryan, 1998.)

CONCLUSIONS FROM PHASE 3A RESEARCH

This study examined the impact of the rotary thermal regenerator and evaporative cooling of the regeneration air on the performance of a gas-fired desiccant system. While the speed of the desiccant wheel, the regeneration temperature, and the air-flow rates remained fixed, a number of experimental runs were made for different scenarios. Variation of the thermal recovery wheel speed and consideration of "on" or "off" status for the evaporative cooler constituted these scenarios. For each case, experimental data were taken when the system was at or near steady state for a given set of inlet conditions.

This study reveals that moisture transfer from the regeneration air stream to the process side caused by the rotation of the thermal recovery wheel is significant and increases with the wheel speed. Activation of the evaporative cooler at the regeneration inlet leads to even higher amount of moisture migration when coupled with the rotating regeneration wheel. Significant reduction in the system latent capacity is observed due to the moisture transfer. This effect is magnified with the increasing dry bulb temperature and decreasing humidity level at the inlet. Unlike the latent capacity, the net capacity of the system, which accounts for the sensible load induced by the desiccant dehumidification, demonstrates a tremendously favorable response to an increase in the wheel speed. (This notion is in line with the improvement seen with the regenerator effectiveness as the wheel speed increases.) Similar observation is made regarding the operation of evaporative cooling although its advantage appears to diminish as the inlet temperature decreases. Evaluation of the overall COP (based on the net capacity) also points at the superiority of scenarios involving a higher wheel rotation speed coupled with evaporative cooling.

4.2 Phase 3B: Parametric Analysis of Variables That Affect the Performance of a Desiccant Dehumidification System

Desiccant dehumidification systems, which are used to reduce the moisture (latent load) of the conditioned air in buildings, are typically specified on the basis of grain depression (pounds of water removed per hour) for a given volumetric flow rate of air at a specified dry bulb and humidity ratio. While grain depression gives some indication of the performance of the system, it does not adequately describe the efficiency of the moisture removal process. Several operating parameters such as desiccant wheel speed, regeneration temperature, volumetric air-flow rate, wheel thickness, sector angles, and desiccant loading affect the ability of the desiccant dehumidification system to remove moisture. There are so many design parameters that influence the operation of a desiccant system that it is difficult to quantify the impact from the interactions on system performance. The purpose of this study is to investigate the impact of varying some of these operating parameters on the performance of a desiccant dehumidification system and to report the results using more quantitative measures, such as latent capacity and latent coefficient of performance (latent COP), that better describe the efficiency of the moisture removal process. The results will be used to improve the understanding of the operation of desiccant systems and to optimize their performance by changing certain operating parameters or improving components.

Two desiccant loadings were tested; one at normal production level and the other with 25% more desiccant applied to the wheel. For both desiccant loadings, the latent capacity and COP increased as desiccant wheel speed increased. As expected, latent capacity improved significantly as air-flow rates increased. It is noted, however, that the efficiency (latent COP) was quite sensitive to air flow rate and showed a maximum at a particular flow rate that best matched the other system operating/design conditions. Finally, higher regeneration temperatures resulted in significant increases in latent capacity for both desiccant loadings, with little or no change in latent COP. Therefore, cost-effective means of achieving higher regeneration temperatures should be investigated.

EXPERIMENTAL PLAN

The desiccant dehumidification system process, shown in Figure 8, brings in outside air (1) and passes it through a desiccant material, removing moisture and increasing the temperature of the air in the process. Next, the air flows through a thermal wheel (2) to lower the temperature of dry air to acceptable occupant levels. The dry air is then introduced to the supply air stream (3) or is passed through a cooling or heating coil for further conditioning. The desiccant material used in the dehumidification process can be a liquid or solid applied or formed in situ on a porous matrix. The desiccant wheel is constantly turning to allow the desiccant to move between process and regeneration air streams. In order to regenerate the desiccant, outside air or exhaust air from the building is brought in and passed through an evaporative cooler (4) which provides cooling air for the thermal wheel (5) to cool the process air stream. As the regeneration air stream passes through the regenerative heat exchanger, it picks up heat from the process air stream prior to entry into the regeneration heat exchanger (6). Air passing through the rotating thermal wheel rapidly increases in temperature prior to entering the desiccant wheel (7) where it drives off the absorbed moisture. The warm, moist air is then discharged to the atmosphere (8).

There are three basic modes of operation for desiccant systems: 1) **recirculation mode** where the unit treats return air from the building and uses outdoor air to regenerate the desiccant; 2) **ventilation mode** where the unit conditions outdoor air before it is introduced to the space and regenerates the desiccant with outdoor air; and 3) **ventilation with heat recovery mode** where the unit conditions outdoor air being supplied to the space and uses exhaust air from the building to regenerate the desiccant (CDH Energy 1995). All the testing performed on the unit in this study was in the **ventilation mode** since it represents the majority of applications for desiccant dehumidification systems. An additional benefit from this study is that the results can be compared to numerous field tests presently being conducted to evaluate problems with poor performance.

The test plan was developed to analyze the impact of the main operational design variables that affect system performance. The test matrix (Table 2) was designed to minimize the number of data points required to obtain adequate information on the impact of each operational parameter. As shown in Table 2, the regeneration temperature varied from 180 to 230°F (82.2 to 110°C), air-flow rate for both the process and regeneration air streams ranged from 1500 to 3300 ft^3/min (42.5 to 93.4 m^3/min), and desiccant wheel speed was varied from 33 to 76 revolutions per hour (rph). In addition, two different desiccant loadings (concentrations) were tested; one at the normal production level and the other with 25% more desiccant applied to the wheel.

In separate tests, the carryover effect (moisture transfer from the regeneration side to the process side) resulting from the use of the evaporator pad was evaluated by running the system with the evaporator pad on and off while varying the thermal wheel speed from 0 to 20 revolutions per minute (rpm) and holding the desiccant wheel in a static (0 rph) position. Previous tests have shown that the effectiveness of the evaporative pad decreases with increasing relative humidity levels of the incoming air on the regeneration side (Jalalzadeh 1998). The purpose of this test was to determine if losses in latent capacity resulting from evaporative pad operation are justified by increased sensible cooling of the process airstream when the desiccant dehumidification system is run in the ventilation mode at high relative humidities.

TEST FACILITY

The test facility is a modified production desiccant dehumidification system and equipped with instrumentation to measure temperature, air- and water-flow rates, dew point temperature, and electrical energy input. Sensors used for these measurements and their associated accuracies are shown in Table 3. Requirements for accuracy of test instrumentation are in accordance with ASHRAE Standard 139-1998 (ASHRAE 1998). Air-side measurements for the process and regeneration air streams include inlet and outlet dry bulb and dew point measurements and air-flow rates. A chilled mirror hygrometer was used to measure dew point temperature on the basis of previous research showing improved accuracy for evaluation of the absolute humidity ratio via dew point temperature as compared to relative humidity and dry bulb temperature measurements (Jalalzadeh 1996). The dry bulb and dew point temperatures were used to determine the enthalpy and absolute humidity ratio. From these properties, the latent capacity and latent COP can be determined.

The inlet dry bulb temperature for the process air stream was maintained at +/- 0.5°F (0.3°C) by using a 10 kW heater along with exhaust air from the regeneration outlet. For the regeneration air stream, a 30 kW heater was used to regulate the dry bulb temperature. Wet bulb temperatures on both air streams were maintained at +/- 0.5°F (0.3°C) by introducing steam from process lines.

Wheel speeds for both the desiccant and thermal wheels were determined by marking the perimeter of the rotors and counting the revolutions while measuring time with a stopwatch accurate to 1/100 second. The sheet metal panel on the process section between the desiccant and thermal wheel was replaced with a plexiglas panel to allow viewing of the wheels to ensure proper rotation and convenient measurement of wheel speeds.

TEST PROCEDURES

Tests were conducted to determine the impact from varying several design parameters on the latent capacity and latent COP of the desiccant dehumidification system. The latent capacity is derived from the following equation:

$$\text{Latent Capacity} = m_{AIR} \times (\omega_{IN} - \omega_{OUT}) \times h_{WATER} \times 60 \qquad (8)$$

where m_{air} represents the mass flow rate of air, ω represents the absolute humidity ratio, and h is the heat of vaporization for water. The latent COP, a measure of the overall system efficiency, is determined by dividing the latent capacity by the total power input to the process, including the boiler, fans, and motors. All tests were performed at the first condition listed in ARI Standard 940-98 (Table 4) which calls for the process and regeneration air inlet conditions to be controlled at 95°F (35°C) dry bulb temperature and 75°F (23.9°C) wet bulb temperature (ARI 1998). This condition was selected since it is close to the process and regeneration conditions at which the unit was initially tested by the manufacturer and it represents the peak design condition for desiccant dehumidification systems in the ventilation mode. The test unit was operated in a steady-state mode (no boiler cycling) to accurately measure the effects for all the different parameters. In field installations, the gas boiler normally cycles 4-5 times per hour resulting in large fluctuations in the regeneration temperature.

Prior to beginning the tests, the production desiccant dehumidification system was modified to allow a parametric analysis of several of the components. The original unit was designed for an air-flow rate of 2600 ft^3/min (73.6 m^3/min) and utilized a 160,000 Btu/h (input rate) gas boiler to heat the regeneration air stream. The desiccant wheel rotated at a rate of 76 rph and the thermal wheel turned at 10 rpm. The regeneration temperature was 190°F (87.8°C). Modifications to the production unit included the following: 1) the existing heat exchanger for regenerating the desiccant was replaced with a larger unit to allow higher regeneration temperatures and steady-state conditions to be achieved; 2) desiccant and thermal wheel single-speed motors were replaced with inverter-driven motors to allow for wheel speeds to be varied; 3) process and regeneration fan motors were replaced with larger inverter-driven fan motors to allow the air-flow rates to be varied and to achieve higher air flows than the production unit; and 4) a 60 kW electric boiler was added to run in stand-alone mode or in series with the gas boiler to increase the heat input for higher regeneration temperatures and for improving steady-state temperature control.

EXPERIMENTAL RESULTS

The testing is designed to assess desiccant dehumidification system design modifications that improve unit capacity and efficiency and identify improvements for "next generation" products and applications. The experiments will also aid in supporting rating/certification standards for desiccant-based products. Areas of interest include evaporative cooler effectiveness, desiccant loading, and variations in desiccant wheel speed, air-flow rate, and regeneration temperature.

Evaporative Cooler Effectiveness

Tests were conducted with the evaporative pad on and off while varying the speed of the thermal wheel and holding the desiccant wheel in a static position to determine the amount of moisture transfer (latent load) to the process air stream. The results, shown in Table 5, indicate that the thermal wheel transfers a difference of 6.2 grains/$lb_{DRY\ AIR}$ at 10 rpm (normal wheel speed) and 10.7 grains/$lb_{DRY\ AIR}$ at 20 rpm. The penalty from using the evaporative pad for the high relative humidity and temperature test condition ranges from 13,800 to 19,500 Btu/h. Since this translated to such a significant decrease in the latent capacity, the remainder of the system tests were run without the evaporative pad on. For future tests at lower temperature and relative humidity conditions where the cooling gain should more than offset the decreased latent capacity, the effectiveness of the evaporative pad will be reevaluated.

Desiccant Loading

The amount of desiccant material loaded onto a wheel affects the amount of moisture removal at different system design and operating conditions. Optimal desiccant loading can allow multiple possibilities for system enhancements such as a lower regeneration temperature, slower wheel speeds, or higher air-flow rates. This could provide better energy conservation, air-flow circulation, and system flexibility. To test the effects of desiccant loading, two different wheels were investigated; one with the manufacturer's specifications for the amount of desiccant and the other with 25% more desiccant material. All tests were performed for both wheels at the same operating parameters. The results for both loadings are shown in Figures 9 through 14. In general, the heavily loaded wheel yielded better latent COPs than the lightly loaded wheel for all the parameters being investigated. More detailed results are given in the following sections.

Desiccant Wheel Speed

Ideally, the desiccant wheel is rotated at a speed where the desiccant will be near total saturation at a point just before it rotates out of the process air stream into the regeneration stream. Similarly, the regeneration temperature, regeneration sector angle, and desiccant wheel rotation rate should be designed so that dried desiccant is rotated out of the regeneration air path and into the process path right at the point where the last few molecules of absorbed water are removed from the desiccant loaded on the wheel. Wheel speeds that are too fast do not utilize all of the active desiccant for process-side water removal or allow for total desiccant regeneration which results in lower capacity and thermal cycling with sub-optimal moisture removal (low efficiency). Desiccant wheel speeds that are too slow allow saturated desiccant to remain in the process air stream too long and excess heating of already activated desiccant which also translate into capacity and efficiency losses.

Figures 9 and 10 summarize the data obtained from experimental runs in which the desiccant wheel speed is varied from 33 to 76 rph while the other system operating parameters are held to the baseline conditions in Table 2. Quite obviously, the latent capacity and latent efficiency for the heavily loaded desiccant wheel remain higher at lower wheel speeds when compared to the lightly loaded wheel. This is a reasonable result because more desiccant is available for drying air on the heavily loaded wheel at the longer process-side residence times (slower rotation rates). The efficiency of the lightly loaded wheel may suffer at the slower wheel speeds because too much time is spent in the regeneration air path for the amount of desiccant on the wheel. The heavily loaded wheel can better utilize time in the regeneration air stream because it has more desiccant to regenerate. At a rotation rate of 58 rph (manufacturer's desiccant wheel speed), there is essentially no benefit from higher desiccant loadings at these test conditions. Faster rotation rates than those explored in this work may actually show a ***disadvantage*** for the heavily loaded wheel because excess desiccant will not be effectively utilized and the higher heat capacity of the heavier wheel will promote more thermal cycling losses between the process and regeneration air streams (Collier 1997).

Air-Flow Rate

By their design, desiccant dehumidification systems move relatively large volumes of air to achieve higher capacities. This dependence on air movement increases the relative size of desiccant-based equipment when compared to refrigerant/vapor-compression systems of similar capacities and it places a major emphasis on the parasitic pressure-drop losses that result from moving large volumes of air through a sequence of filters, wheels, and heat exchanger coils. Each active component in the desiccant system must be designed and operated with a targeted air-flow rate in order to achieve optimal unit performance. Experimental results from variations of air flow help define the sensitivity of unit performance to this important operational design variable and establish an optimal flow rate that best suits the capabilities of individual components used to build the unit. A cautionary note is required, however, in that the laboratory test unit used to obtain this data was operated in a manner where the desiccant regeneration temperature was maintained at a fixed temperature with the aid of added boiler capacity and a larger regeneration heat exchanger, Figure 8. In a production unit, increased air-flow rates with a limited boiler capacity and conventionally sized regeneration heat exchanger will eventually result in lower regeneration temperatures which will have an additional, secondary effect on the unit's capacity and efficiency. In addition, process and regeneration airflow rates were balanced (equal) for all the tests.

Figures 11 and 12 summarize the results of variations in process and regeneration air-flow rates on the latent capacity and COP of the desiccant unit under test. Figure 11 shows that moisture removal capacity increases with air-flow rates, as would be expected. This increase occurs in roughly the same proportion for both desiccant loadings and most likely results from the regeneration temperature being maintained at 190°F (87.8°C). A modest increase in capacity is shown for the heavily loaded wheel. Intuitively, a large enough air-flow rate which would totally saturate the desiccant at a given set of inlet conditions, wheel rotation rate, regeneration temperature, and desiccant loading would eventually cause the curves in Figure 11 to flatten out. Limitations in the capabilities of the test unit prevent reaching this flow rate.

The curves in Figure 12 indicate that the efficiency peaks at a certain flow rate. This is probably caused by the balanced process and regeneration airflow rates in combination with all the other variables being kept constant. Some process/regeneration flow rate ratio other than 1/1 is probably optimal over such a wide range of air throughput (Van den Bulck 1992). Note that the maxima appears to shift from approximately 2450 to 2600 ft^3/min (69.4 to 73.6 m^3/min) for the more heavily loaded desiccant wheel as compared to the lightly loaded wheel. This maxima shift is in the right direction since higher desiccant loadings should favor higher air-flow rates.

Regeneration Temperature

An external heat source is required in active desiccant dehumidification systems to drive absorbed water off the desiccant in a regeneration process so it will be in a state where it can absorb more water from the process air stream. At best, the COP of this absorption/desorption process on the desiccant wheel can approach 1.00, because at least as much thermal energy is required to desorb the water from the desiccant as the latent Btu/lb benefit obtained in removing water from building ventilation air. Naturally, equipment designers want to minimize regeneration energy input in order to increase the operating efficiency of the overall desiccant dehumidification system.

Variations in system latent capacity and COP for lightly and heavily loaded desiccant wheels as a function of regeneration temperature are shown in Figures 13 and 14, respectively. As in previously described parametric studies, the other system operating conditions were held to the baseline values shown in Table 2. Figure 13 shows steadily increasing capacities for both wheels as regeneration temperatures are increased. Data plotted in Figure 13 indicate that the latent capacity is slightly higher for the heavily loaded desiccant wheel at regeneration temperatures below 200°F (93.3°C). Of the two desiccant loadings, the more heavily loaded wheel is obviously better suited to the air-flow rate, wheel rotation rate, and process/regeneration sector split at lower temperatures.

It is interesting to note the crossover of the heavily loaded wheel versus the lightly loaded wheel curves at regeneration temperatures greater than 200°F (93.3°C). This may due to the greater heat capacity of the heavily loaded wheel resulting in higher thermal cycling losses and lower latent capacities at the higher regeneration temperatures. For the desiccant wheels tested in this study, regeneration temperatures between 200 - 220°F (93.3 - 104.4°C) appear to give the best combination of increased capacity at near optimal efficiency.

CONCLUSIONS FROM PHASE 3B RESEARCH

The performance of a solid desiccant dehumidification system depends on a complicated interplay of system operating and design parameters. Optimal system capacity and operating efficiency depend on balancing these operational components against the design limitations of individual equipment components used by the manufacturer to construct the unit. Parametric studies like the one presented here help provide a more intuitive understanding of how these desiccant-based systems operate, a means of verifying modeled results, and a guide for future design improvements.

Some compromises in component selection are driven by an attempt to provide an affordable product with versatility so that it is applicable to several different modes of operation. The unit tested was designed with a direct evaporative cooler at the entrance to the regeneration side so it could be easily applied in a **"ventilation with heat recovery"** mode where relatively cool and dry (low enthalpy) building exhaust air can be used to effectively cool the process air stream and regenerate the desiccant. When the unit is used in the **ventilation mode**, test results indicated that the evaporative cooler has marginal utility. Therefore, it was not used in testing variations in the other operational parameters.

Since the primary intention of desiccant-based equipment is to provide "latent air conditioning", it is inappropriate to compare it to other, more conventional approaches based on total cooling - latent plus sensible capacity. One confusing aspect of using a total cooling criteria for rating desiccant equipment results from the sensible heating of air which occurs as it is dried by the desiccant wheel. It is possible with a desiccant-based dehumidification unit to provide dried process air at a higher temperature than inlet conditions indicating a negative sensible capacity and a total capacity for the unit that is less than its latent capacity. Desiccant systems should be rated in terms of latent cooling provided. Similarly, the coefficient of performance should be based on the latent cooling output/energy input. Any inter-comparisons between thermally driven and electrically driven dehumidification technologies should be on the basis of primary energy and should include consideration of the sensible heat ratio performance of the respective equipment.

Parametric analysis revealed that the evaporator pad was marginally effective and actually resulted in a capacity decrease when used at the test conditions. Therefore, water was not circulated through the pad in subsequent testing. A comparison of two different desiccant loadings indicated that a heavier loading was generally more efficient (higher latent COP) than a lighter loading. The higher loading showed increased latent capacity versus the lighter loading at lower desiccant wheel speeds and regeneration temperatures.

For both desiccant loadings, the latent capacity and COP increased as desiccant wheel speed increased. Limitations in the test rig prevented testing beyond 76 rph. As expected, latent capacity improved significantly as air flow rates increased. It is noted, however, that the efficiency (latent COP) was quite sensitive to air flow rate and showed a maximum at a particular flow rate that best matched the other system operating/design conditions. Finally, increasing the regeneration temperatures resulted in significant increases in latent capacity for both desiccant loadings, with little or no change in latent COP. Therefore, cost-effective means of achieving higher regeneration temperatures should be investigated.

4.3 Phase 3C: Performance Characteristics at Off-Design Conditions for a Thermally-RegeneratedDesiccant System

The purpose of this study is to investigate the performance of a thermally-regenerated desiccant system at off-design inlet air conditions and to report the results using more quantitative measures, such as latent capacity and latent coefficient of performance (COP), that better describe the efficiency of the moisture removal process. The testing focused on two inlet dry bulb temperatures; the design dry bulb temperature, 95°F (35°C), and 80°F (26.7°C), which represents a higher absolute humidity condition. In addition to testing at different inlet air conditions, several operating parameters, such as desiccant wheel speed, regeneration temperature, and volumetric air-flow rate were varied to quantify their impact on system performance. The results will be used to improve the understanding of the operation of desiccant systems and to optimize their performance by changing certain operating parameters or improving components.

Desiccant system designs are typically specified for a 95°F (35°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature outdoor ambient condition. However, most desiccant systems operate at this condition less than 2% of the year. A more meaningful indication of desiccant system performance is shown by testing at a design condition that is experienced for a greater number of hours, such as the 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature. Because a desiccant system is competing against conventional vapor compression equipment as an alternative means of removing moisture, this ambient condition is also a better comparison since the efficiency of air-conditioning equipment is determined at 82°F (27.8°C).

For both inlet air conditions, the latent capacity increased significantly as desiccant wheel speed, regeneration temperature, and air-flow rate increased. The results also indicate that latent capacity increased significantly more for the 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition as these variables increased. This suggests that the moisture removal of the desiccant system is much improved over that indicated by testing at the design condition. The results for the latent COP were much different than those for the latent capacity. As desiccant wheel speed increased, the latent COP increased. However, as the regeneration temperature increased, the latent COP showed a slight decrease. The latent COP was quite sensitive to air-flow rate and showed a maximum at a particular air-flow rate that best matched the other system operating/design conditions. Latent COPs for the 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition were slightly less than those for the design condition. This result is most likely caused by the increased energy that must be supplied on the regeneration side of the process to heat the air due to the lower entering temperature of the outside air.

EXPERIMENTAL PLAN

The test plan was developed to analyze the impact of the main operational design variables that affect system performance. The test matrix (Table 6) was designed to minimize the number of data points required to obtain adequate information on the impact of each operational parameter. As shown in Table 6, the regeneration temperature was varied from 180 to 230°F (82.2 to 110°C), air-flow rate for both the process and regeneration air streams ranged from 1500 to 3300 ft^3/min (42.5 to 93.4 m^3/min), and desiccant wheel speed was varied from 33 to 76 revolutions per hour (rph). In addition to the baseline tests performed at a 95°F (35°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature, tests were also performed at an 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition to evaluate the system performance at an off-design inlet air condition.

EXPERIMENTAL RESULTS

The testing is designed to assess desiccant dehumidification system design modifications that improve unit capacity and efficiency and identify improvements for "next generation" products and applications. The experiments will also aid in supporting rating/certification standards for desiccant-based products. Areas of interest include variations in entering ambient conditions, desiccant wheel speed, air-flow rate, and regeneration temperature.

Ambient Conditions

Baseline tests were performed at a 95°F (35°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature since that is the condition at which desiccant systems are designed and specified. In addition, system tests were also performed at an 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition to evaluate the system performance at an off-design condition. Typically, the system only operates at the design condition less than 2% of the year. The majority of the operation takes place at off-design conditions, such as the 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition. Thus, a more accurate assessment is indicated by testing at an off-design inlet air condition. Also, the 80°F (26.7°C) dry bulb temperature, 75°F (23.9°C) wet bulb temperature condition represents a situation where conventional vapor compression systems are unable to run long enough to remove enough moisture from the conditioned space without overcooling. Thus, it offers a comparative data point for evaluating the performance of desiccant systems to conventional vapor compression systems in providing overall space conditioning (latent and sensible).

Desiccant Wheel Speed

Figures 15 and 16 summarize the data obtained from experimental runs in which the desiccant wheel speed is varied from 33 to 76 rph at two inlet air conditions while the other system operating parameters are held to the baseline conditions in Table 6. Quite obviously, the latent capacity for both inlet air conditions increases as wheel speed increases. However, the latent capacity at the 80°F (26.7°C) inlet condition is increasing at a much faster rate than for the 95°F (35°C) ambient condition. At the lower wheel speed, too much time is spent in the regeneration air path for the amount of desiccant on the wheel. This results in less moisture removal on the process side at both ambient temperatures since the desiccant becomes saturated before the wheel has completed the full rotation through the process side. At the higher wheel rotation rates, the latent capacity is much higher for the lower ambient temperature condition primarily because the partial pressure of the water vapor, which is the driving force, is higher.

The results for latent COP (Figure 16) show a similar effect as the latent capacity in that latent COP increases as the wheel speed increases. However, the latent COP for the higher ambient temperature is higher than that of the lower ambient temperature, a reversal of the latent capacity results. This is the result of a lower amount of heat input required at the higher ambient temperature to reach the specified regeneration temperature. The latent COPs converge for the 95°F (35°C) and 80°F (26.7°C) inlet air temperature curves at 76 rph as the result of the greater increase in capacity at the lower inlet air temperature.

Air-Flow Rate

Figures 17 and 18 summarize the results of variations in process and regeneration air-flow rates on the latent capacity and COP of the desiccant unit under test. Figure 17 shows that moisture removal capacity increases with air-flow rates, as would be expected. This increase is more pronounced at the 80°F (26.7°C) dry bulb temperature due to the higher moisture content in the incoming airstream. Intuitively, a large enough air-flow rate which would totally saturate the desiccant at a given set of inlet conditions, wheel rotation rate, regeneration temperature, and desiccant loading would eventually cause the curves in Figure 17 to flatten out. Limitations in the capabilities of the test unit prevent reaching this flow rate. It should be mentioned that the extent of dehumidification, commonly referred to as grain depression, decreases as the air flow rate increases. Therefore, the higher latent capacity results from a larger amount of air being less completely dried (higher grains/lb at the desiccant wheel exit).

The curves in Figure 18 indicate that the efficiency peaks at approximately 2100 ft^3/min (59.5 m^3/min). Note that the latent COP drops off more rapidly on either side of the maxima for the 80°F (26.7°C) inlet air condition. At air-flow rates below 2100 ft^3/min (59.5 m^3/min), the desiccant material is underutilized for both conditions. However, more energy is required to heat the entering air for the 80°F (26.7°C) inlet air compared to the 95°F (35°C) dry bulb temperature. As the air-flow rate approaches 2100 ft^3/min (59.5 m^3/min), the latent capacity is increasing at a faster rate for the 80°F (26.7°C) inlet air temperature, compared to the higher ambient temperature, resulting in equivalent latent COPs for both inlet air conditions. Above 2100 ft^3/min (59.5 m^3/min), the energy required to heat the entering air for the 80°F (26.7°C) dry bulb temperature condition is increasing at a higher rate than the increase in latent capacity, resulting in a drop-off in the latent COP, compared to the 95°F (35°C) inlet air condition. It is noted that all tests were run with equal flow rates on both the regeneration and process side. A previous study indicates that reducing the air flow rate for the regeneration air relative to the process air can produce even higher latent COPs (Jalalzadeh-Azar et al. 2000). However, decreasing the regeneration air-flow rate would also reduce the post-cooling advantage offered by the heat exchanger wheel, resulting in a higher exiting dry bulb temperature.

Regeneration Temperature

Variations in system latent capacity and COP as a function of regeneration temperature for both entering dry bulb temperatures are shown in Figures 19 and 20, respectively. As in previously described parametric studies, the other system operating conditions were held to the baseline values shown in Table 6. Figure 19 shows steadily increasing capacities for both conditions as regeneration temperatures are increased. Data plotted in Figure 19 indicate that the latent capacity is much higher for the 80°F (26.7°C) dry bulb temperature condition. This result follows the trend established in the wheel speed and air-flow rate tests.

It is interesting to note the results for latent COP (Figure 20) for the two ambient dry bulb temperature conditions. The latent COP at 80°F (26.7°C) shows almost no decrease as the regeneration temperature increases. This is the result of the latent capacity increasing at almost the same rate as the energy required for regeneration. The latent COP at 95°F (35°C) gradually decreases, however, to the point where it is equivalent to that for the lower ambient temperature condition. This occurs as the result of the latent capacity for the 95°F (35°C) condition increasing at a lesser rate than for the 80°F (26.7°C) condition.

CONCLUSIONS FROM PHASE 3C RESEARCH

For both dry bulb inlet temperatures, the latent capacity significantly increased as desiccant wheel speed, air-flow rate, and regeneration temperature increased. Tests at the 80°F (26.7°C) temperature showed a greater improvement mainly because of the higher partial pressure of the water vapor at that temperature, which is the driving force for absorption of water on the desiccant material. This conclusion indicates that the performance of desiccant systems is markedly improved at off-design conditions, which represent the majority of operating hours. If it is the intent to optimize performance for latent capacity alone, then the design should increase desiccant wheel speed, air-flow rate, and regeneration temperature to the highest levels possible

The results for latent COP were much different than those for latent capacity. In all the tests, the 95°F (35°C) results had higher latent COPs than those for the lower inlet dry bulb temperatures. This was mainly due to the increased energy required for heating the lower temperature inlet air up to the regeneration temperature. As desiccant wheel speed increased, the latent COP for both inlet dry bulb temperature test conditions increased dramatically. In contrast, the latent COPs for air-flow rate variations reached a maxima at approximately 2100 ft^3/min (59.5 m^3/min) before beginning a rapid descent. This behavior is attributed to the increased energy required to raise the inlet air up to the regeneration temperature. Finally, as regeneration temperature increased, the latent COPs decreased; with the most significant decrease occurring at the 95°F (35°C) test condition. This suggests that latent capacity can be increased significantly at the off-design conditions with little or no decrease in efficiency.

One final note on the results is that designers must determine the requirements of desiccant systems in order to apply the lessons learned from this study. For instance, if energy costs are low, then latent COP can be sacrificed by raising the wheel speed, air-flow rates, and regeneration temperature to increase latent capacity. However, if energy costs are high, a balance needs to occur in choosing the operating characteristics to optimize both latent capacity and COP. In addition, the comfort levels in the conditioned space also need to be considered when designing for an optimum latent capacity. As mentioned in a previous section, increasing air-flow rates results in an increased latent capacity. However, the grain depression across the wheel is lower which results in air being delivered to the conditioned space at a higher relative humidity. This can reduce the comfort levels that the desiccant system is intended to improve.

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TABLE 1
Experimental Data for Reference Point in Figure 13a

Description	Symbol	Results and uncertainties
Dry bulb temperature at process inlet	T_p1	35.1 +/- 0.5°C
Dry bulb temperature at process inlet	T_p3	30.8 +/- 0.7°C
Humidity ratio at process inlet	w_p1	11.0 g/kg d.a. +/- 3%
Humidity ratio at process exit	w_p3	6.9 g/kg d.a. +/- 5%
Moisture removal capacity per unit mass of dry air	Δw	4.1 g/kg d.a. +/- 6%
Effectiveness of regeneration wheel	ε	0.84 +/- 5%
Process and regeneration air mass flow rate	$\dot{m}$	4250 m^3/h +/- 2%
Latent capacity	$\dot{Q}_l$	15.1 kW +/- 6%
Net capacity	$\dot{Q}_{net}$	21.3 kW +/- 6%
Regeneration heat input	$\dot{Q}_{reg.}$	42.9 kW +/- 5%
COP for latent capacity	COP_l	0.35 +/- 7%
COP based on net capacity	COP	0.50 +/- 7%

TABLE 2
Baseline and Parametric System Operation Values

Operational Parameters	Baseline Values	Parametric Variations
Process/Regeneration Dry Bulb Temperature	95°F (35°C)	----
Process/Regeneration Wet Bulb Temperature	75°F (23.9°C)	----
Desiccant Wheel Speed	58 rph	33 - 76 rph
Thermal Wheel Speed	10 rpm	----
Process/Regeneration Air Flow Rate	3000 ft^3/min (85 m^3/min)	1500 - 3300 ft^3/min (42.5 - 93.4 m^3/min)
Regeneration Temperature	190°F (87.8°C)	180 - 230°F (82.2 - 110°C)

TABLE 3
Desiccant Test Instrumentation

Measurement	Sensor	Precision/Accuracy
Temperature	Averaging RTD	=/- 0.24% at 70°F (21.1°C) Range = -50 to 275° F (-45.6 to 135°C)
Air Flow	Fan Evaluator	+/- 2% Range 500 - 5000 ft^3/min (14.2 - 141.6 m^3/min)
Water Flow	Turbine Meter	+/- 0.5% of reading Range 2.5 to 29.0 gal/min (9.5 to 109.8 L/min)
Dew Point Temperature	Chilled Mirror	Dew Point: +/- 0.2°C (-80 to 95°C)
Power	Watt Transducer (2)	+/- 0.5% of full scale Range = 0 to 40000 watts 0 to 500 watts

TABLE 4
ARI Standard Rating Conditions *

Condition Number	Process Air Inlet Condition Dry Bulb	Process Air Inlet Condition Wet Bulb	Regeneration Air Inlet Condition Dry Bulb	Regeneration Air Inlet Condition Wet Bulb
1	95°F (35°C)	75°F (23.9°C)	95°F (35°C)	75°F (23.9°C)
2	80°F (26.7°C)	75°F (23.9°C)	80°F (26.7°C)	75°F (23.9°C)
3	80°F (26.7°C)	67°F (19.4°C)	95°F (35°C)	75°F (23.9°C)
4	45°F (7.2°C)	45°F (7.2°C)	80°F (26.7°C)	75°F (23.9°C)

* The tolerance for all temperatures during the test is +/- 0.5°F (+/- 0.3°C)

TABLE 5
Thermal Wheel Carryover Effects

Wheel Speed (rpm)	Carryover (grains/$lb_{DRY\ AIR}$) Evaporative Pad OFF	Carryover (grains/$lb_{DRY\ AIR}$) Evaporative Pad ON
0	0.0	0.0
10	0.0	6.2 (13,800 Btu/h)
20	3.1 (6,984 Btu/h)	10.7 (19,500 Btu/h)

TABLE 6
Baseline and Parametric System Operation Values

Operational Parameters	Baseline Values	Parametric Variations
Process/Regeneration Dry Bulb Temperatures	95°F (35°C)	80°F (26.7°C)
Process/Regeneration Wet Bulb Temperature	75°F (23.9°C)	----
Desiccant Wheel Speed	58 rph	33 - 76 rph
Thermal Wheel Speed	10 rpm	----
Process/Regeneration Air Flow Rate	3000 ft^3/min (85 m^3/min)	1500 - 3300 ft^3/min (42.5 - 93.4 m^3/min)
Regeneration Temperature	190°F (87.8°C)	180 - 230°F (82.2 - 110°C)

TABLE 7
Desiccant Test Instrumentation

Measurement	Sensor	Precision/Accuracy
Temperature	Averaging RTD	=/- 0.24% at 70°F (21.1°C) Range = -50 to 275° F (-45.6 to 135°C)
Air Flow	Fan Evaluator	+/- 2% Range 500 - 5000 ft^3/min (14.2 - 141.6 m^3/min)
Water Flow	Turbine Meter	+/- 0.5% of reading Range 2.5 to 29.0 gal/min (9.5 to 109.8 L/min)
Dew Point Temperature	Chilled Mirror	Dew Point: +/- 0.2°C (-80 to 95°C)
Power	Watt Transducer (2)	+/- 0.5% of full scale Range = 0 to 40000 watts 0 to 500 watts

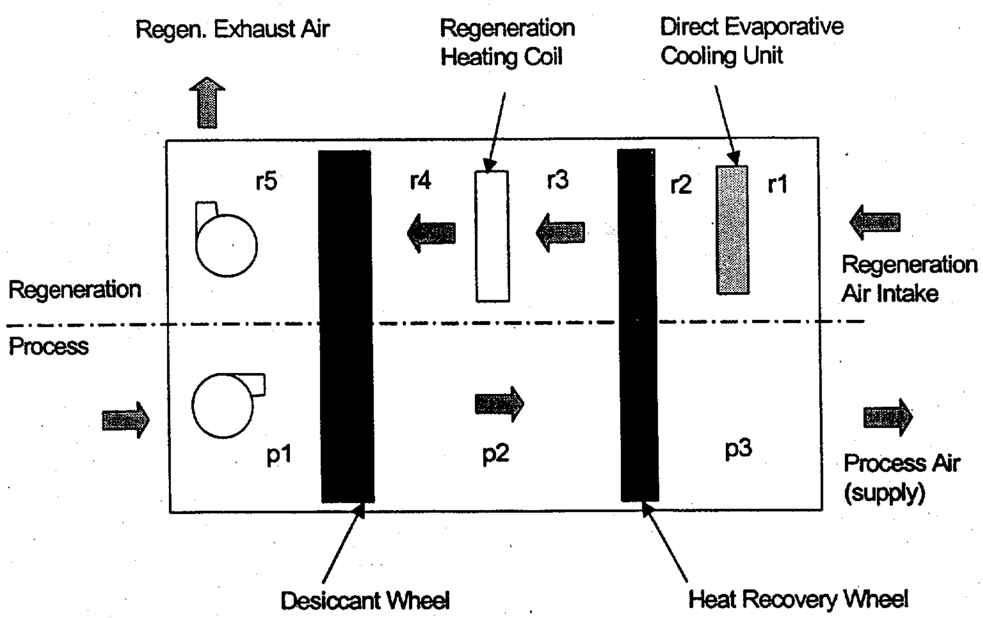

Figure 1. Schematic of desiccant system.

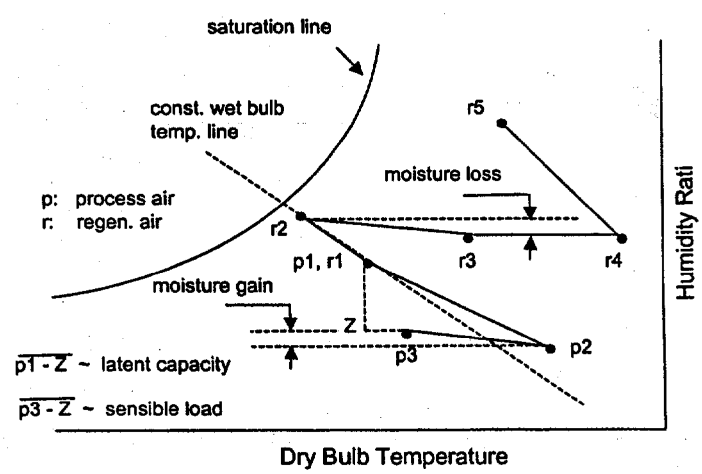

Figure 2. Psychrometric diagram illustrating processes in desiccant dehumidification.

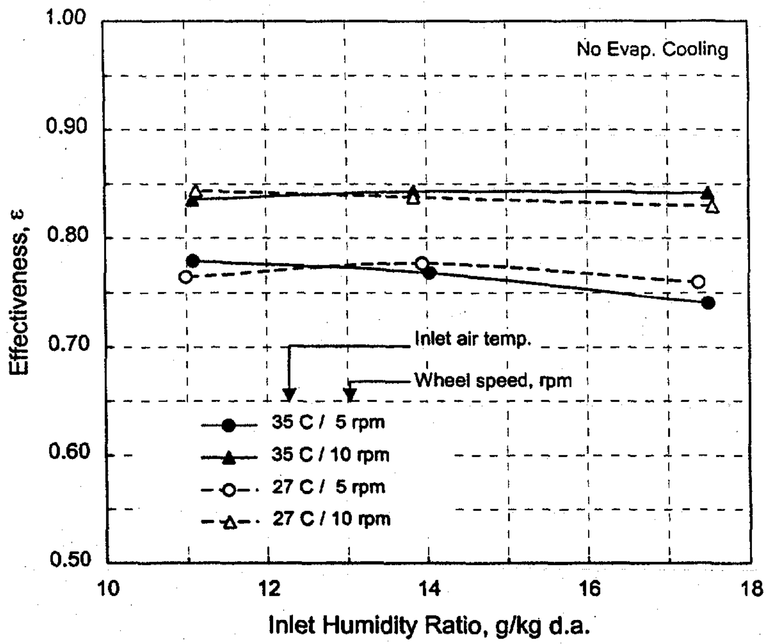

Figure 3. Effectiveness of thermal regenerator

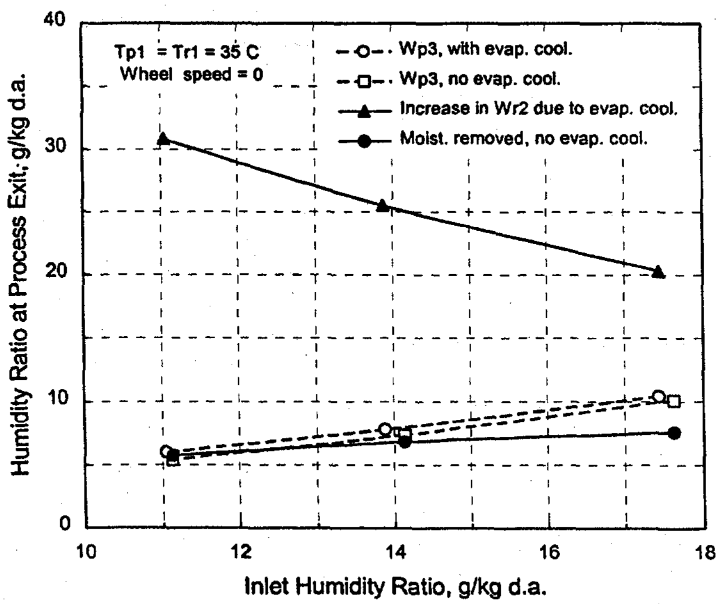

Figure 4a. Effect of evaporative cooling on dehumidification performance for 35°C inlet air temperature.

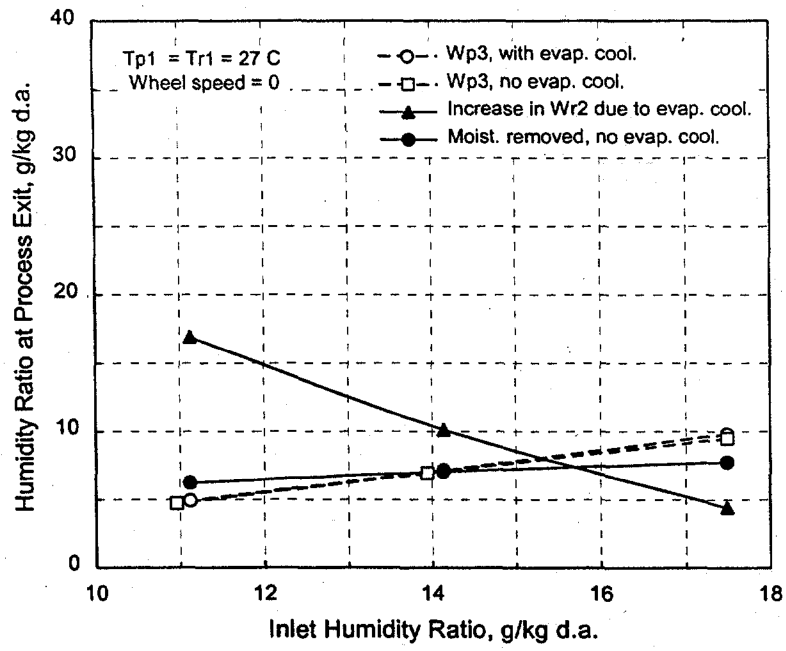

Figure 4b. Effect of evaporative cooling on dehumidification performance for 27°C inlet air temperature.

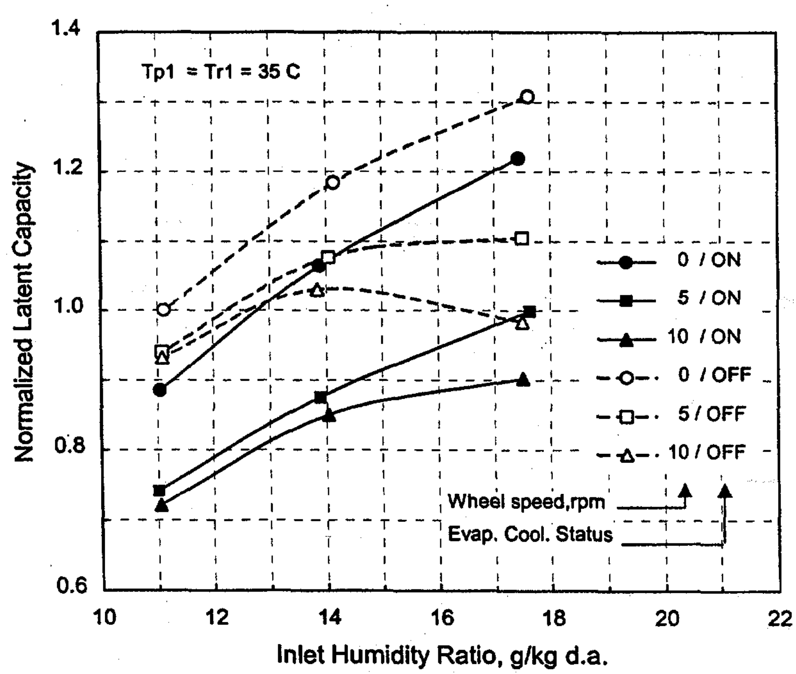

Figure 5a. Latent capacity variation for 35°C inlet air temperature.

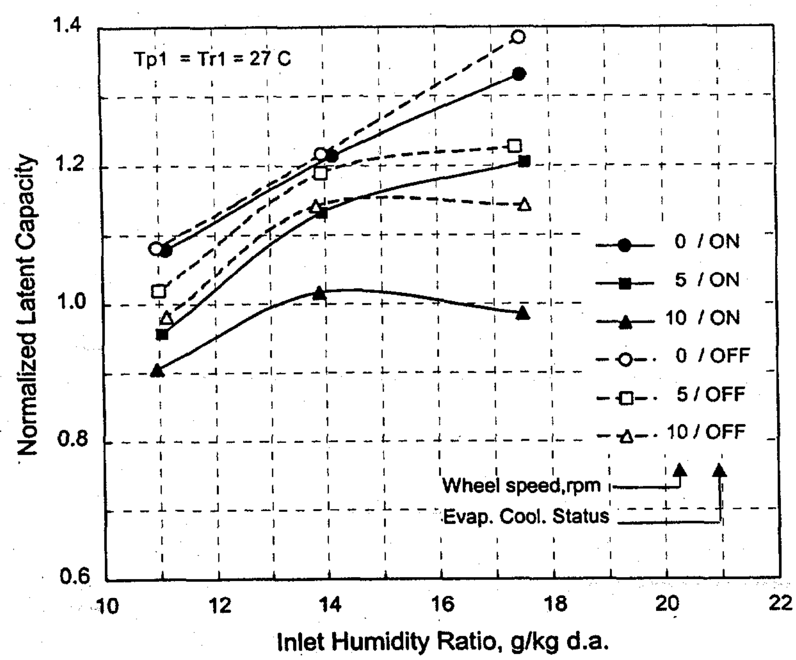

Figure 5b. Latent capacity variation for 27°C inlet air temperature.

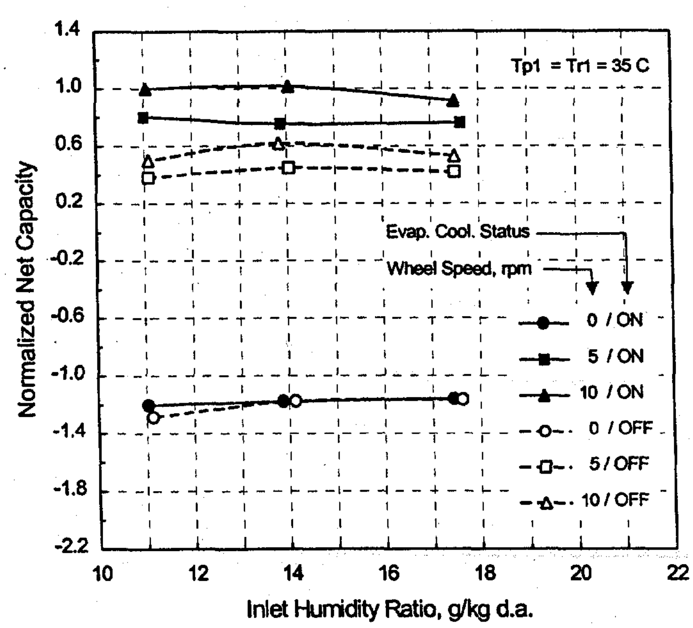

Figure 6a. Net capacity variation for 35°C inlet air temperature.

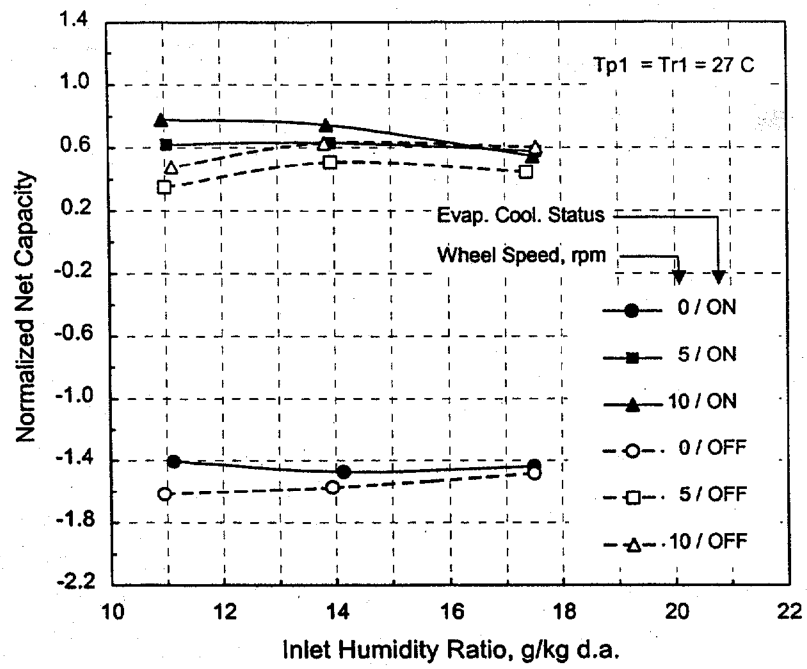

Figure 6b. Net capacity variation for 27°C inlet air temperature.

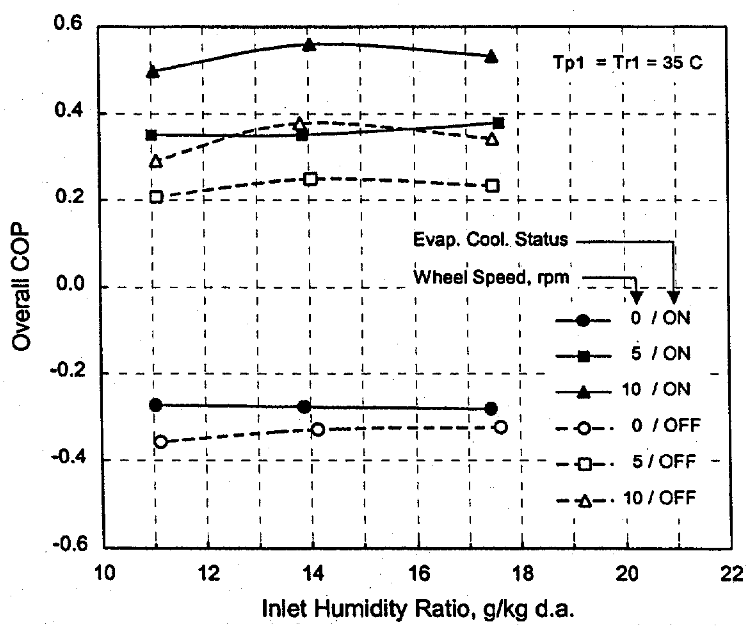

Figure 7a. COP variation for 35°C inlet air temperature.

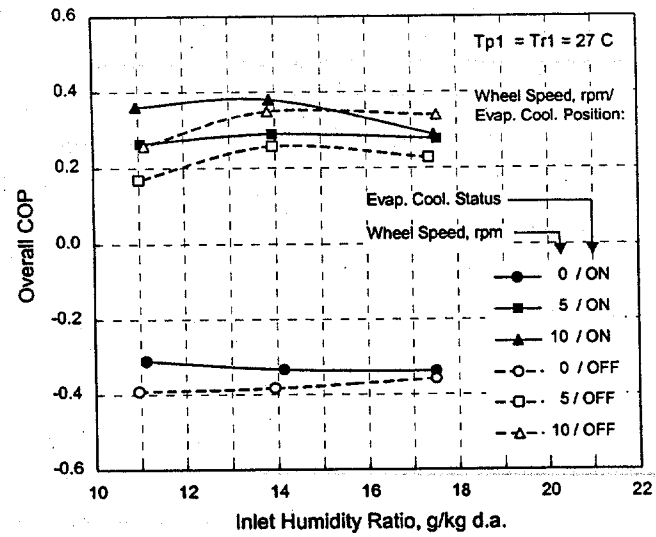

Figure 7b. COP variation for 27°C inlet air temperature.

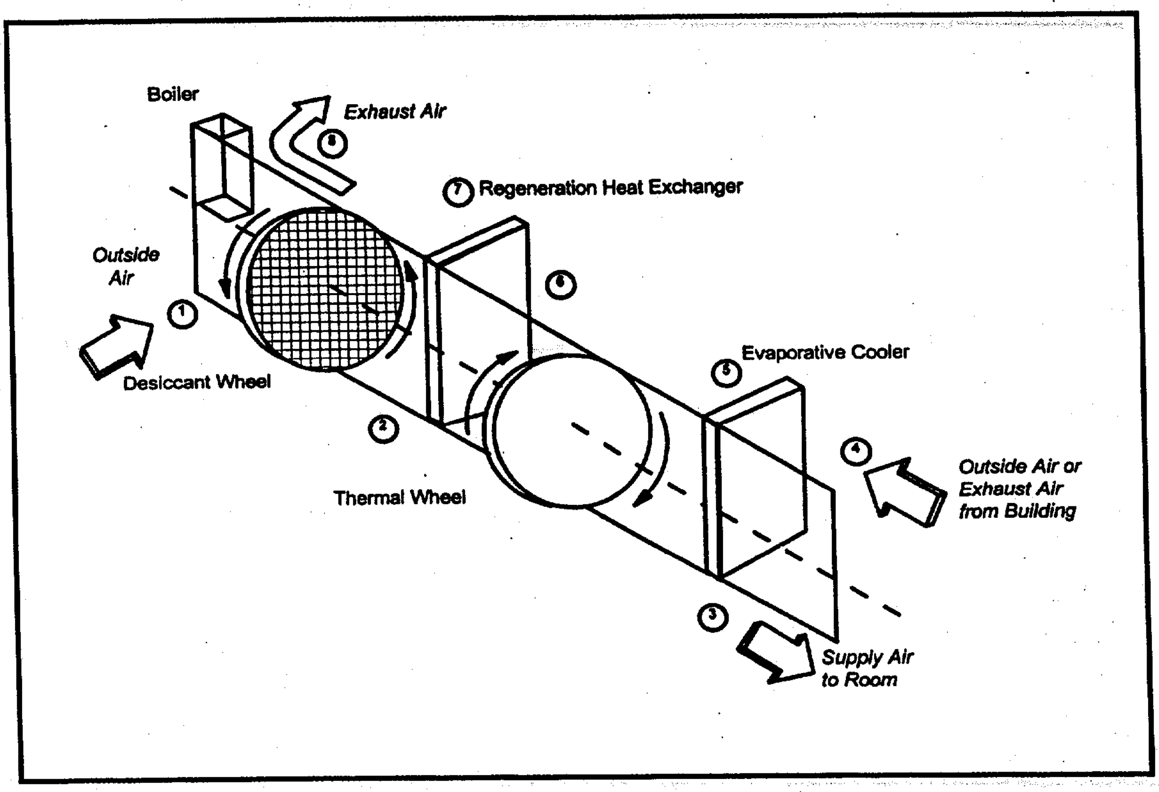

Figure 8. Desiccant dehumidification system schematic

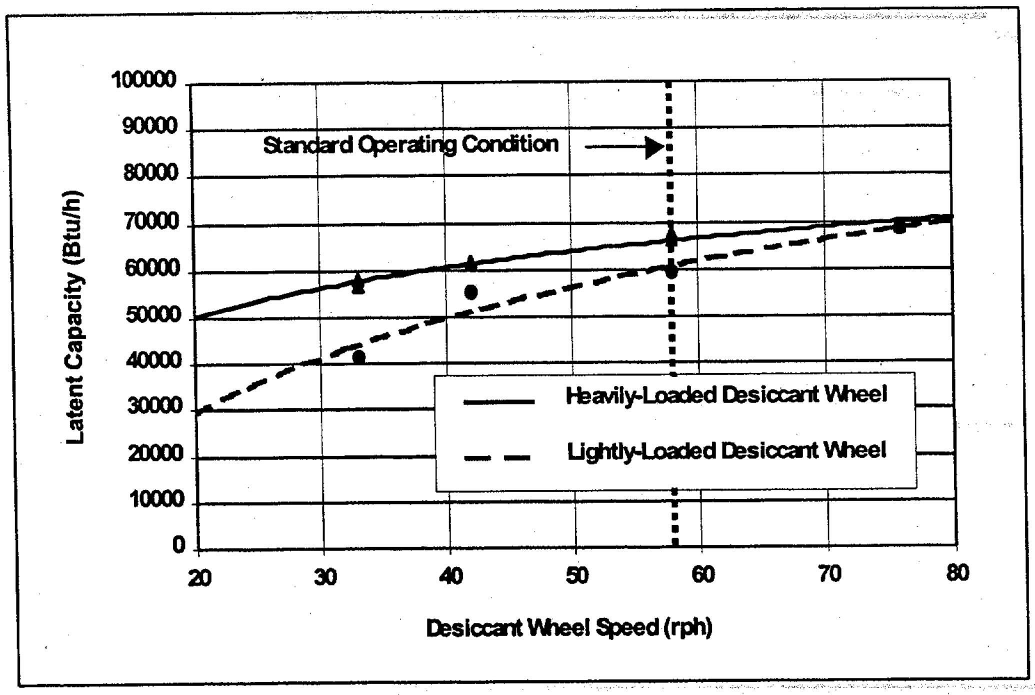

Figure 9. Desiccant wheel speed vs latent capacity

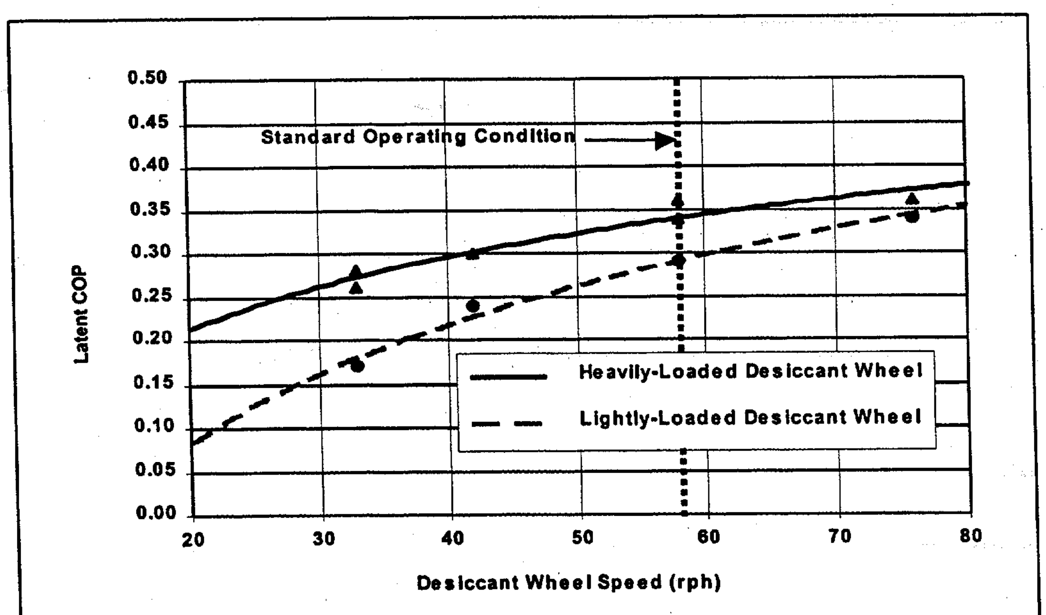

Figure 10. Desiccant wheel speed vs latent COP

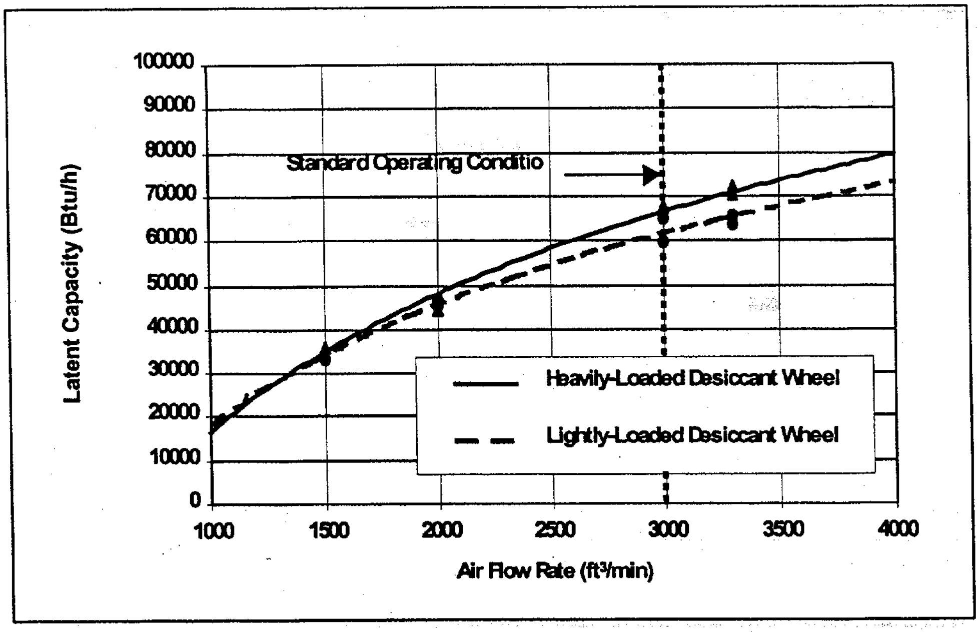

Figure 11. Air Flow rate vs latent capacity

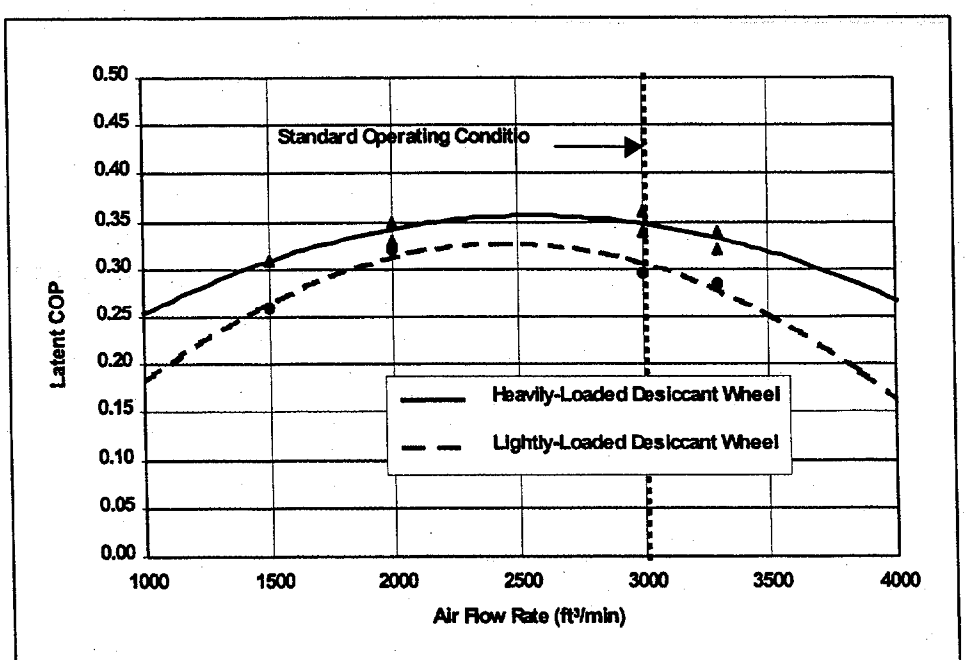

Figure 12. Air flow rate vs latent COP

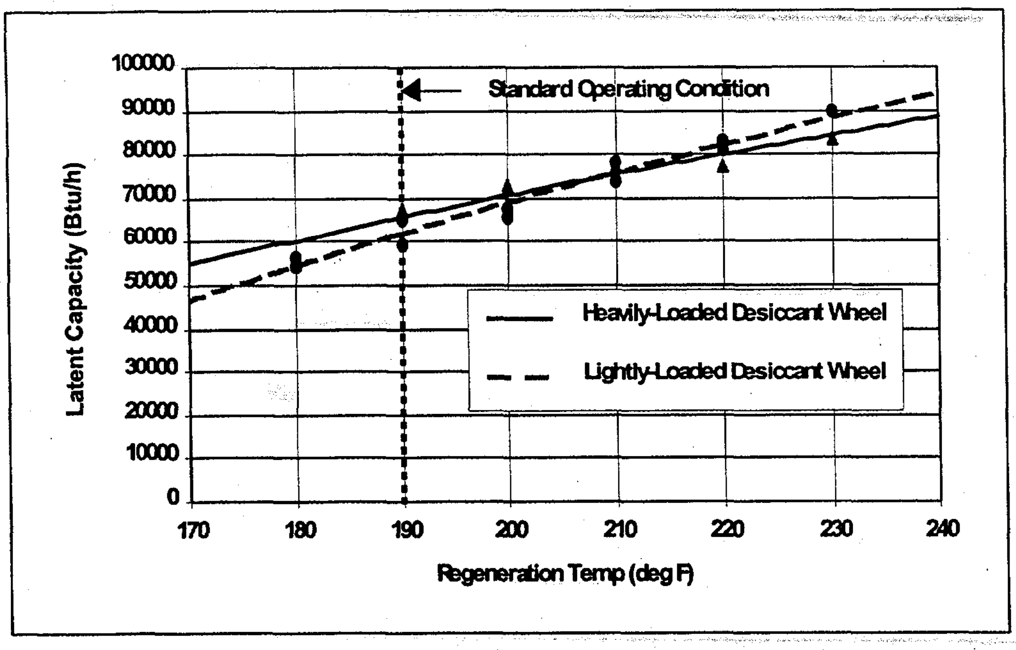

Figure 13. Regeneration temperature vs latent capacity

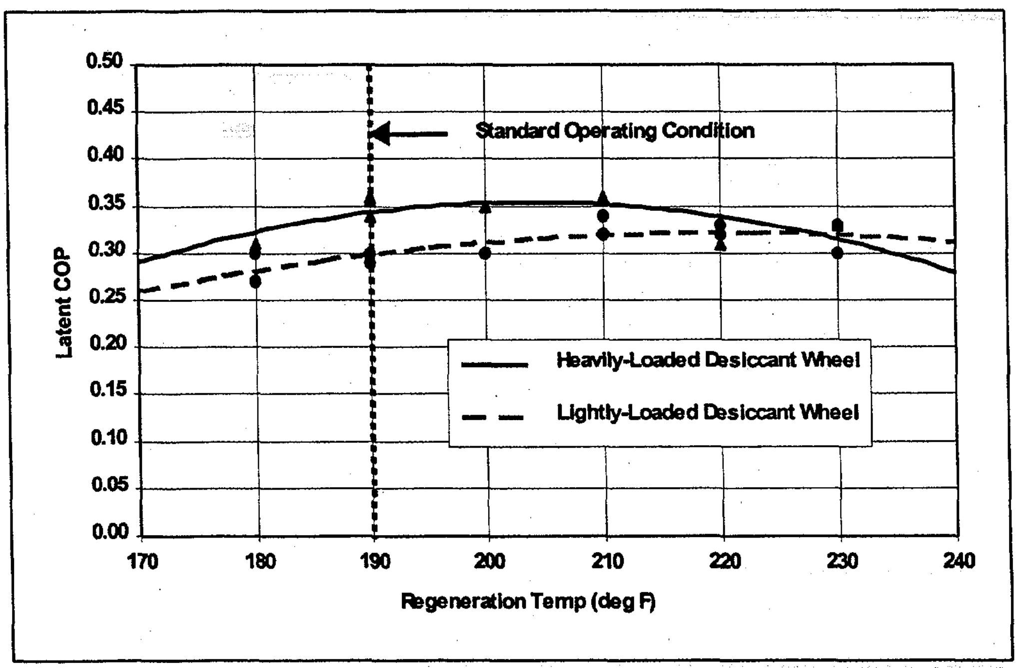

Figure 14. Regeneration temperature vs latent COP

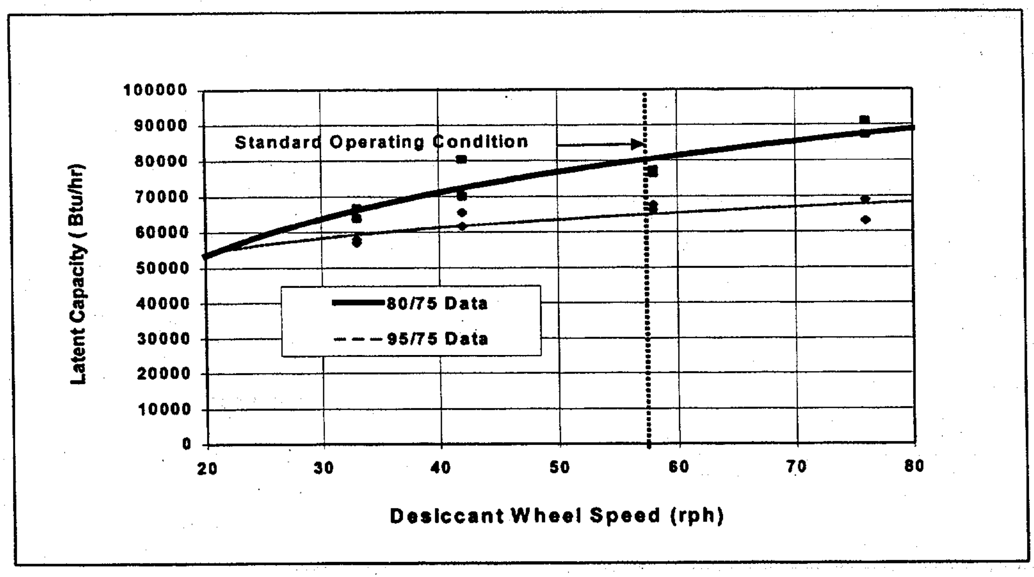

Figure 15. Desiccant wheel speed vs latent capacity

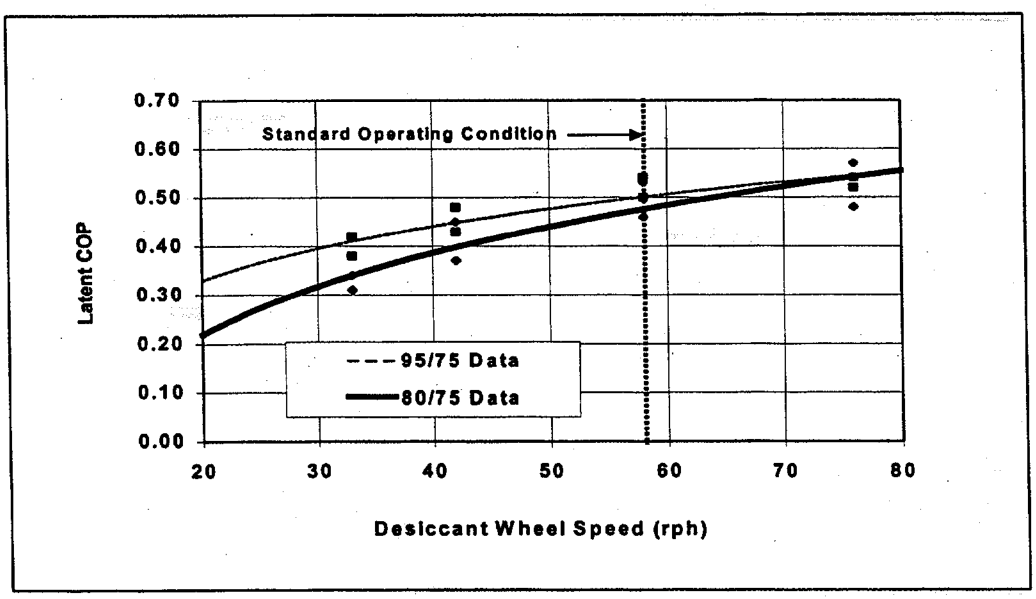

Figure 16. Desiccant wheel speed vs latent COP

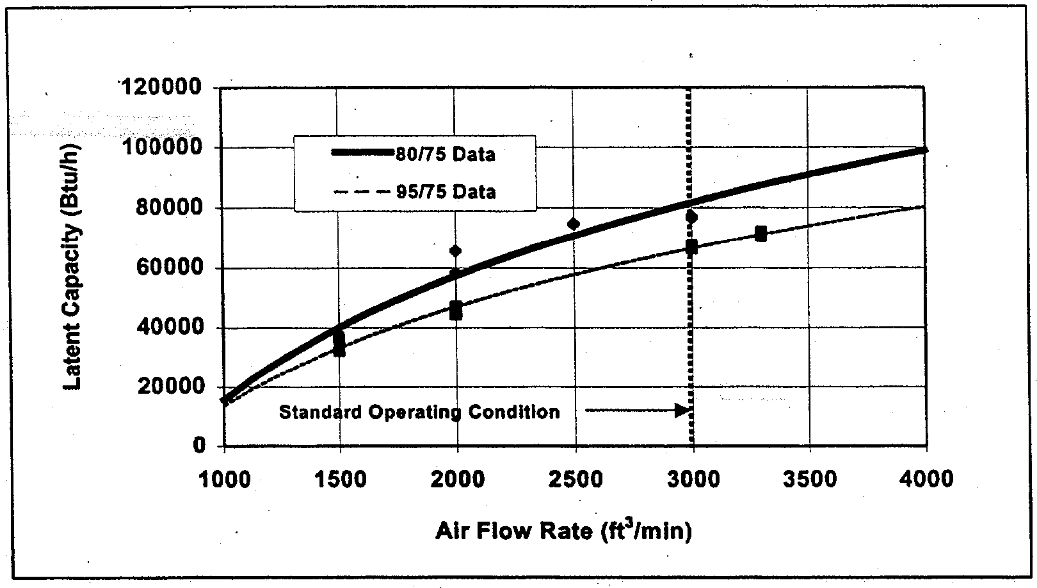

Figure 17. Air flow rate vs latent capacity

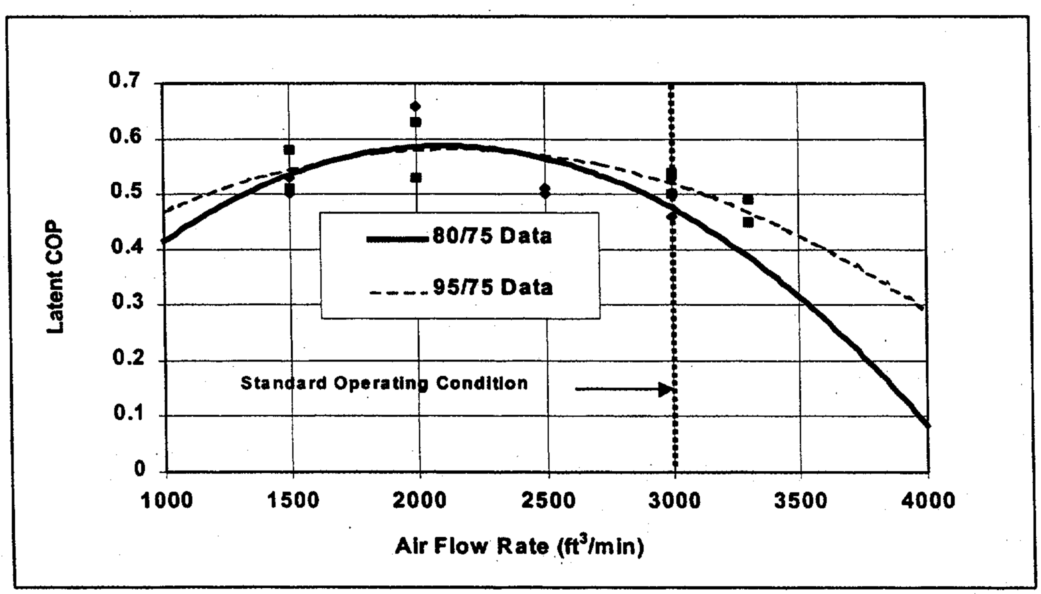

Figure 18. Air flow rate vs latent COP

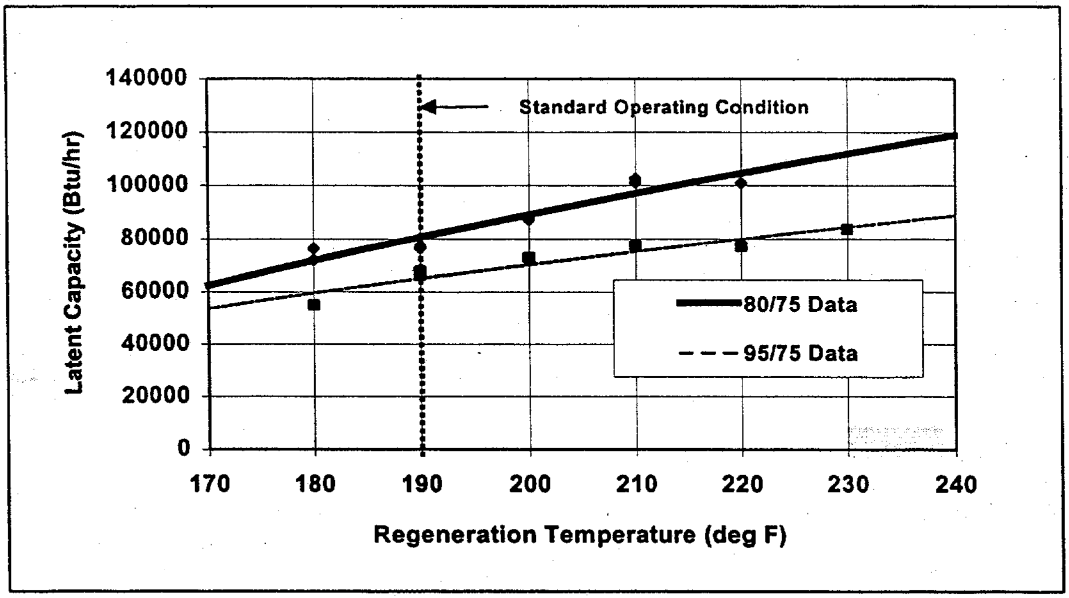

Figure 19. Regeneration temperature vs latent capacity

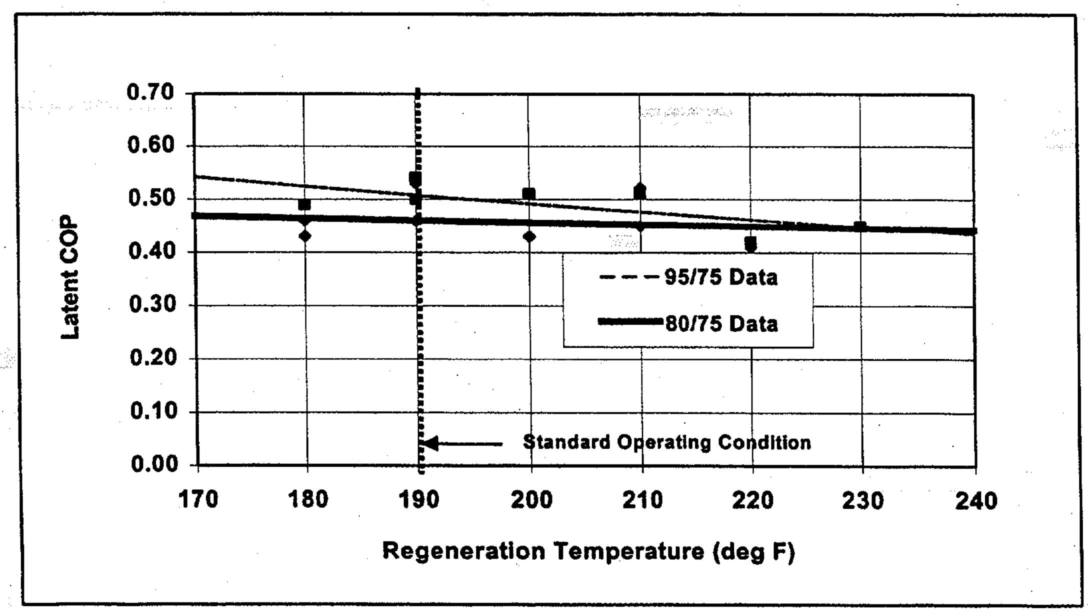

Figure 20. Regeneration temperature vs. latent COP

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