Performance Characteristics for a Desiccant System at Two Extreme Ambient Conditions

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ABSTRACT

The purpose of this study is to investigate the performance of a thermally regenerated desiccant system at two extreme ambient conditions (high sensible and high latent load) and to report the results using quantitative measures, such as latent capacity and latent coefficient of performance (COP), that better describe the efficiency of the moisture removal process. For this study, latent COP is defined as the latent capacity divided by the total energy input of the process. In addition to testing at two different inlet air conditions, several operating parameters, such as desiccant wheel speed, regeneration temperature, and face velocity, 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 space conditioning system designs are typically specified for a 95°F (35°C) dry-bulb temperature and 75°F (23.9°C) wet-bulb temperature outdoor ambient condition (condition 1 in ARI Standard 940-98). However, most systems operate at this condition less than 2% of the year. A more meaningful indication of desiccant system performance is shown by testing at an ambient condition that reflects a higher latent load and is experienced for a greater number of hours, such as condition 2 in ARI Standard 940-98, which specifies an 80°F (26.7°C) dry-bulb temperature and a 75°F (23.9°C) wet-bulb temperature. In order to evaluate the performance of a desiccant system against conventional vapor compression equipment, this ambient condition would also give a better comparison since the efficiency for air-conditioning equipment is determined at 82°F (27.8°C) (ARI 1998).

Thermally regenerated desiccant systems, which are used to reduce the moisture (latent load) of the conditioned air in buildings, are specified on the basis of grain depression (grains of water removed per pound of dry air) for a given face velocity at a specified dry bulb and humidity ratio. While grain depression gives an 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, face velocity, regeneration temperature, 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. Thus, this study seeks to quantify the impact of varying some of the most important operating design parameters individually so as to isolate the changes and report the results using terms, such as capacity and COP, that are common to the air-conditioning industry.

For both inlet air conditions, the latent capacity increased significantly as desiccant wheel speed, regeneration temperature, and face velocity 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 design condition 1. The results for the latent COP were much different from 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 face velocity

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and showed a maximum at a particular face velocity 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 higher temperature condition. This result is most likely caused by the increased energy that must be supplied on the regeneration side of the wheel to heat the air due to the lower entering temperature of the outside air.

INTRODUCTION

This work was performed under an industry/government cooperative research and development agreement (CRADA) established to evaluate and test design concepts for desiccant units using advanced materials and components. The objectives were to (1) significantly improve indoor air quality in buildings, (2) decrease the use of vapor-compression equipment that uses halogenated refrigerants, (3) reduce primary electrical energy consumption resulting in corresponding reductions in carbon dioxide emissions, (4) introduce new products and markets for U.S. industry through expanded utilization of desiccant dehumidification systems, and (5) use the data generated from the study to evaluate proposed rating/certification standards for desiccant systems.

Recently, design dew-point temperatures for 239 locations in the United States have been introduced by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) to be used by specifiers to size and select building heating, ventilation, and air-conditioning (HVAC) equipment (ASHRAE 1997). In 1990, ASHRAE made recommendations in *ANSI/ASHRAE Standard 62-1989* that indoor levels of relative humidity be maintained between 30% and 60% and that ventilation rates be increased to improve indoor air quality (ASHRAE 1990). Unfortunately, increasing the ventilation rates and specifying HVAC equipment on the basis of dew-point temperature increases the amount of latent load that must be handled by the building HVAC system. In response to this problem, desiccant dehumidification systems have been identified as an emerging technology to control the increased latent load in buildings. The conventional approach to controlling the latent load has been to use air-conditioning equipment to reduce moisture in the airstream to acceptable levels or to reduce the quantity of ventilation air. Using conventional air-conditioning equipment usually requires the airstream to be reheated since overcooling occurs—often an unacceptable alternative in light of present energy costs. Using desiccant dehumidification systems to reduce the latent load in combination with conventional air-conditioning equipment can lead to a significant downsizing of the air-conditioning unit since cooling is only required to reduce the sensible load. This particular arrangement yields an improved control of the relative humidity and can, depending on the source of energy for the regeneration process, reduce energy costs.

Advances in desiccant materials have enabled the technology to become more practical for space-conditioning applications. However, additional work needs to be done to improve system efficiency and expand the application opportunities for desiccant-based systems. Studies have shown that the cooling load accompanying ventilation air can be dominated by the latent component in some climates (Harriman et al. 1997). This suggests that applications can benefit tremendously from desiccant technology where the ventilation requirements are significantly increased by *ANSI/ASHRAE Standard 62-1989.*

BACKGROUND

Energy concerns in the late 1970s led researchers to investigate the possibilities of using desiccant dehumidification in residential and commercial air conditioning. Problems were initially encountered because desiccant systems required large amounts of fan power to compensate for the high pressure drops associated with the packed silica gel beds. Large pressure drops required different desiccant materials or system setup for use in air-conditioning applications (Slayzak et al. 1996). Companies in the early 1980s started using lithium chloride in fluted wheels as a possible answer to the problem. The fluted wheels allowed for lower pressure drops and the wheel design was better suited for air-conditioning applications. The only problem was that the lithium chloride started weeping at high relative humidities, requiring a new material to be chosen. Realizing that silica gel did not have this problem, companies developed a silica gel that could be formed into a fluted wheel. Optimization of the silica gel/fluted wheel design process for air-conditioning applications was pursued by companies through funding from the natural gas industry. Different formulations/formations of desiccant materials allowed for different amounts of moisture removal. Titanium silicate was proven to have a larger saturation capacity than regular silica gel and was selected for use in desiccant systems to be marketed in conjunction with air-conditioning units (Slayzak et al. 1996).

Several research organizations, are presently analyzing commercial desiccant systems to improve their performance. The federal government, along with the natural gas industry, is funding this research to help understand how desiccant systems operate in the field in different building applications at high relative humidity conditions and to determine the effects of different design parameters on their performance.

As shown in Figure 1, two airstreams, process and regeneration, are involved in the active desiccant dehumidification system. As the process air flows through the desiccant wheel, the moisture is removed from the airstream. The desiccant is restored to its dry state by exposure to the heated regeneration airstream as it rotates. Due to the heat transfer from the regeneration side via the rotating wheel and the latent heat effect, the process airstream is heated during the dehumidification process. This necessitates post cooling for comfort.

To reduce the energy consumption associated with the post cooling, the process air exiting the desiccant wheel can be partially cooled via a heat recovery wheel. As the heat recov-

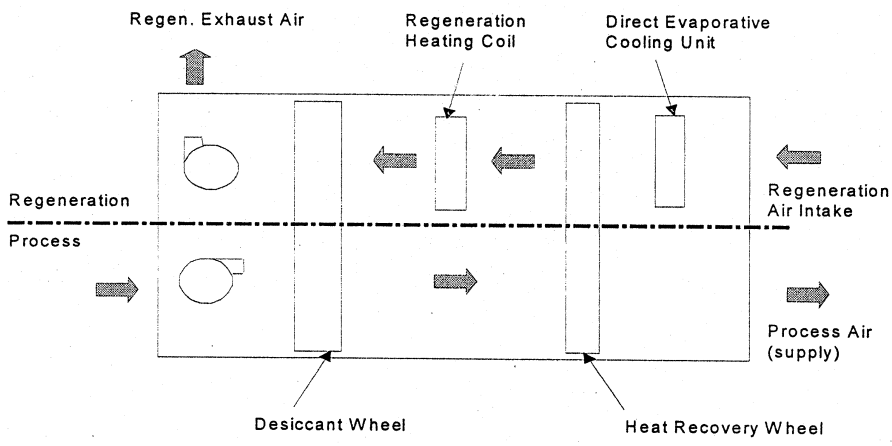

Figure 1 *Desiccant dehumidification system schematic.*

ery wheel rotates, heat is transferred from the process stream to the incoming regeneration air. Consequently, the relatively cool incoming regeneration air is preheated, thus reducing the energy required to heat the regeneration airstream. The amount of heat transferred by the heat recovery 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 of the entering air. This improves the cooling performance but diminishes the preheating effect of the wheel. For this study, the evaporative cooler was not used.

Heat pipes may be used in desiccant systems in lieu of thermal recovery wheels, which permit moisture migration from the regeneration to the process stream due to the air exchange facilitated by the porous wheel matrix. This moisture migration results in the humidity ratio of the process air increasing and that of the regeneration air decreasing, 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. Heat pipes do not share this limitation since there is no moisture migration from the airstreams.

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 setpoint, process and regeneration face velocities, 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. Recent studies have presented parametric analyses relating the performance of commercially available, rotary, gas-fired, desiccant systems to a number of operational variables (Vineyard et al. 2000; Jalalzadeh-Azar et al. 2000). An additional study 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 (Van den Bulck et al. 1986).

This study is intended to characterize system performance at a high sensible and high latent load condition while varying some of the operational parameters, such as desiccant wheel speed, face velocity, and regeneration temperature. Desiccant system performance at a higher humidity is more indicative of a normal ambient condition and provides useful information to HVAC system designers when comparing the performance of a desiccant system to conventional vapor compression equipment as an alternative means of removing moisture.

EXPERIMENTAL PLAN

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 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 1) 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 1, the regeneration temperature was varied from 180°F to 230°F (82.2°C to 110°C); face velocity for both the process

TABLE 1
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 face velocity	476.2 ft/min (145.1 m/min)	238.1–523.8 ft/min (72.6–159.7 m/min)
Regeneration temperature	190°F (87.8°C)	180°μ-230°F (82.2°•-110°C)

TABLE 2
Desiccant Test Instrumentation

Measurement	Sensor	Precision/Accuracy
Temperature	Averaging RTD	±0.24% at 70°F (21.1°C) Range = –50°F-275°F (–45.6°•~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 2.5-29.0 gal/min (9.5-109.8 L/min)
Dew-point temperature	Chilled mirror	Dew point: ±0.2°C (–80°•~95°C)
Power	Watt transducer (2)	±0.5% of full scale Range = 0-40000 watts 0-500 watts

and regeneration airstreams ranged from 238.1 ft/min to 523.8 ft/min (72.6 m/min to 159.7 m/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, to evaluate the system performance at a higher latent load condition.

TEST FACILITY

Experimental data were taken using a desiccant dehumidification system with the same features as shown in Figure 1, with the exception of the evaporative cooler, which was disengaged. The facility is equipped with the necessary instrumentation to measure air dry-bulb and dew-point temperatures at the principal points (Figure 1) and process and regeneration airflow rates. Requirements for accuracy of test instrumentation are in accordance with *ANSI/ASHRAE Standard 139-1998* (ASHRAE 1998). A chilled mirror was used to measure dew-point temperature on the basis of previous research, showing improved accuracy for evaluation of the humidity ratio via dew-point temperature as compared to relative humidity and dry-bulb temperature measurements (Jalalzadeh-Azar et al. 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 airstream 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 airstream, a 30 kW heater was used to regulate the dry-bulb temperature. Wet-bulb temperatures on both airstreams 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 plexiglass 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:

$$\text{Latent Capacity} = m_{air} \times (\omega_{in} - \omega_{out}) \times h_{water} \times 60 \qquad (1)$$

TABLE 3
ARI Standard Rating Conditions*

Condition Number	Process Air Inlet Condition		Regeneration Air Inlet Condition	
	Dry Bulb	Wet Bulb	Dry Bulb	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)

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 energy input of the process. The energy input was determined from measurements of the thermal energy supplied to the heat exchanger for the regeneration process and the electrical power to the fans and motors. Electrical power, measured in kilowatts, was converted to thermal energy using a conversion factor (3413 Btu/kWh). It is noted that an additional factor could be applied so that the electrical energy input is taken directly back to the power plant (primary energy). Since the majority of the input energy is the thermal energy for the heat exchanger, using primary energy for the fans and motors would only increase the total energy a small amount and result in a slight reduction in the latent COP.

Tests were performed at conditions 1 and 2 listed in ARI Standard 940-98 (Table 3), which calls for the process and regeneration air inlet conditions to be controlled at 95°F (35°C) and 80°F (26.7°C) dry-bulb temperatures and 75°F (23.9°C) wet-bulb temperature (ARI 1998). These conditions were selected since they are representative of high sensible and latent load conditions for desiccant dehumidification systems operating 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, a production desiccant dehumidification system was modified to allow a parametric analysis of several of the components. The original unit was designed for a face velocity of 412.7 ft/min (125.8 m/min) and utilized a 160,000 Btu/h (47 kW) gas boiler to heat the regeneration airstream. 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 hot water coil 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 airflow 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 variations in entering ambient conditions, desiccant wheel speed, face velocity, 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 ambient condition at which air-conditioning systems have traditionally been 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 a high latent load condition. Typically, the system only operates at the higher ambient condition less than 2% of the year. The majority of the operation takes place at a lower ambient conditions, such as the 80°F (26.7°C) dry-bulb temperature, 75°F (23.9°C) wet-bulb temperature condition. The 80°F (26.7°C) dry-bulb temperature, 75°F (23.9°C) wet-bulb temperature condition also 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

The desiccant wheel is ideally rotated at a speed where the desiccant is no longer removing moisture at an appreciable rate before it rotates out of the process airstream 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 when it is no longer practical or efficient to add more thermal energy to remove the last few molecules of absorbed water from the desiccant.

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 airstream too long and also overheat the already dry desiccant, which reduces capacity and efficiency.

Figures 2 and 3 summarize the data obtained from experimental runs in which the desiccant wheel speed is varied from 33 rph to 76 rph at two inlet air conditions while the other system operating parameters are held to the baseline conditions in Table 1. Quite obviously, the latent capacity for both inlet air conditions increases as wheel speed increases. It is noted that there will be a wheel speed at which the latent capacity will begin to drop. However, that point was not reached in the testing. Also, 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) 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 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 show a similar trend as the latent capacity in that latent COP increases as the wheel speed increases. However, the latent COP for the higher temperature

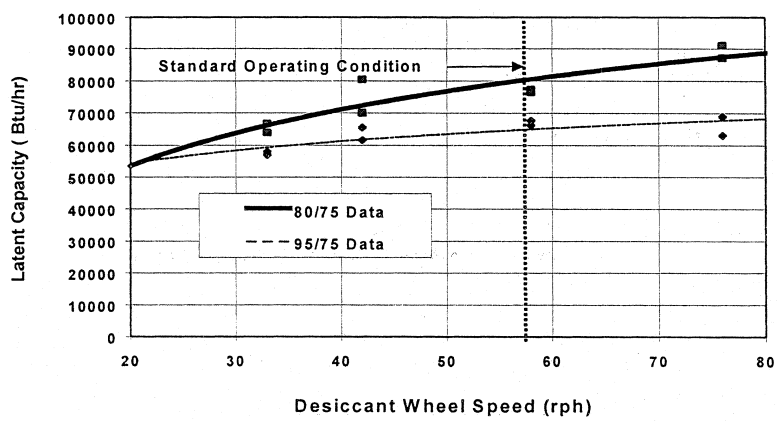

Figure 2 *Desiccant wheel speed vs. latent capacity.*

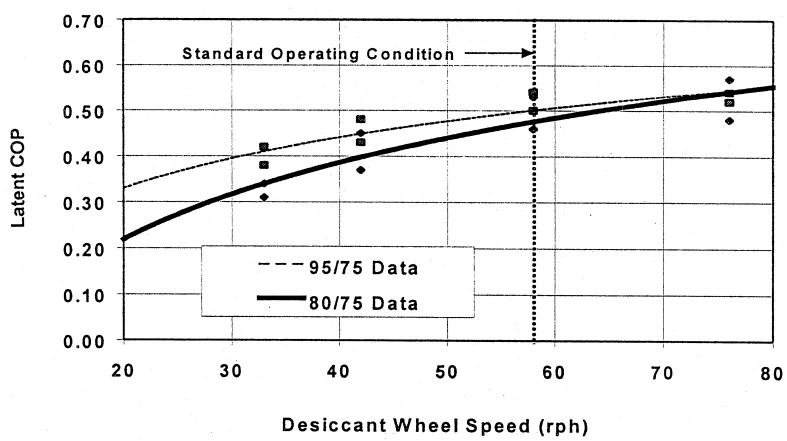

Figure 3 *Desiccant wheel speed vs. latent COP.*

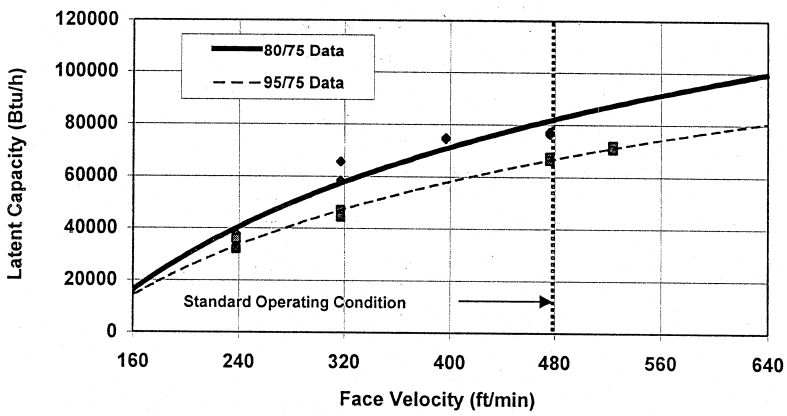

Figure 4 *Face velocity vs. latent capacity.*

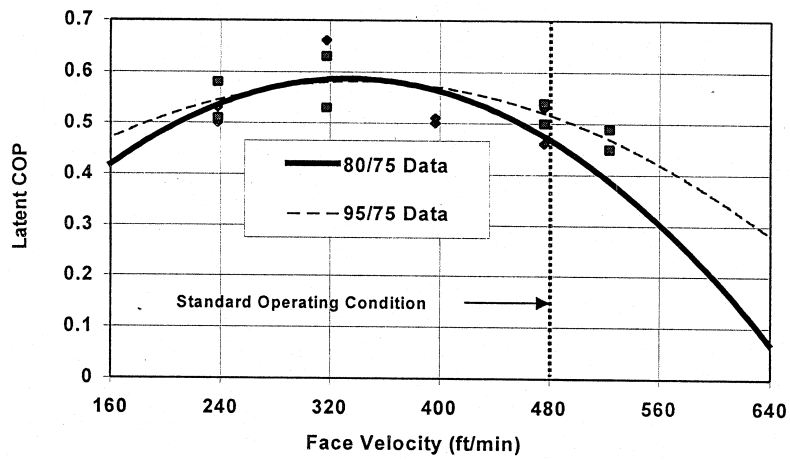

Figure 5 *Face velocity vs. latent COP.*

is higher than that of the lower temperature, a reversal of the latent capacity results. This results from less heat input required to reach the specified regeneration temperature when the inlet air is at the higher ambient temperature. The latent COPs converge for the 95°F (35°C) and 80°F (26.7°C) inlet air temperature curves as the result of the greater increase in capacity at the lower inlet air temperature.

Face Velocity

Each active component in the desiccant system must be designed and operated with a targeted face velocity in order to achieve optimal unit performance. Experimental results from variations in face velocity help define the sensitivity of unit performance to this important operational design variable and establish a face velocity 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 these data was operated such that the desiccant regeneration temperature was maintained at a fixed temperature with the aid of added boiler capacity and a larger regeneration heat exchanger. In a production unit, increased face velocities with a limited boiler capacity and conventionally sized regeneration heat exchanger will eventually result in lower regeneration temperatures that will have an additional, secondary effect on the unit's capacity and efficiency. In addition, process and regeneration face velocities were equal for all the tests.

Figures 4 and 5 summarize the results of variations in process and regeneration face velocities on the latent capacity and COP of the test unit. Figure 4 shows that moisture removal capacity increases with face velocity, 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 face velocity, 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 4 to flatten out. Limitations in the capabilities of the test unit prevent reaching this face velocity. It should be mentioned that the extent of dehumidification, commonly referred to as grain

depression, decreases as the face velocity increases. Therefore, the higher latent capacity results from a larger amount of air being dried less completely.

The curves in Figure 5 indicate that the efficiency peaks at approximately 333.3 ft/min (31.0 m/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 face velocities below 333.3 ft/min (31.0 m/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 face velocity approaches 333.3 ft/min (31.0 m/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 333.3 ft/min (31.0 m/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 face velocities on both the regeneration and process side. A previous study indicates that reducing the face velocity for the regeneration air relative to the process air can produce even higher latent COPs (Jalalzadeh-Azar et al. 2000). However, decreasing the regeneration face velocity would also reduce the post-cooling advantage offered by the heat exchanger wheel, resulting in a higher exiting dry-bulb temperature.

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 airstream. 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 load 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 as a function of regeneration temperature for both entering dry-bulb temperatures are shown in Figures 6 and 7, respectively. As in previously described parametric studies, the other system operating conditions were held to the baseline values shown in Table 1. Figure 6 shows steadily increasing capacities for both conditions as regeneration temperatures are increased. Data plotted in Figure 6 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 face velocity tests.

It is interesting to note the results for latent COP 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

Some compromises in component selection are driven by an attempt to provide an affordable product with versatility so that it is applicable in several modes of operation. The unit tested was originally equipped with a direct evaporative cooler at the entrance to the regeneration side. However, previous test results indicated that the evaporative cooler has marginal utility when the unit is operated in the ventilation mode (Vineyard et al. 2000). Therefore, the evaporative cooler was not used in this particular study.

For both dry-bulb inlet temperatures, the latent capacity significantly increased as desiccant wheel speed, face veloc-

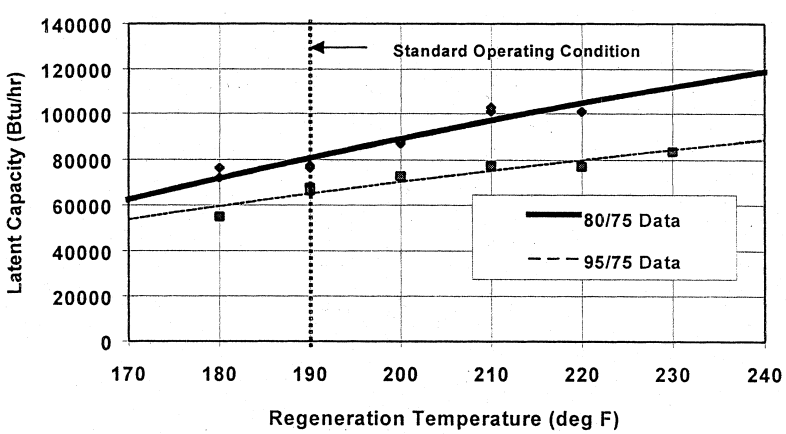

Figure 6 *Regeneration temperature vs. latent capacity.*

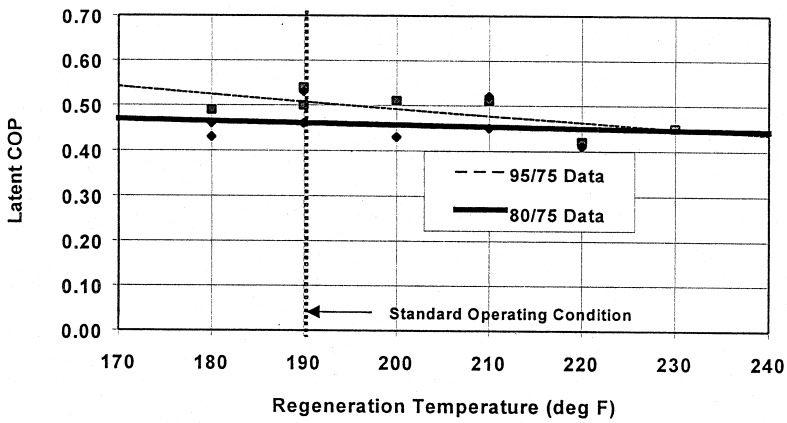

Figure 7 *Regeneration temperature vs. latent COP.*

ity, 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 higher latent load conditions.

The results for latent COP were much different from 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 face velocity variations reached a maxima at approximately 333.3 ft/min (31.0 m/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 higher latent load condition with little or no decrease in efficiency.

The results from this study can be used to modify the desiccant system design to suit the particular situation. For instance, if energy costs are low, then latent COP can be sacrificed by raising the face velocity 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 the face velocity results in an increased latent capacity. However, the air delivered to the space is more humid (higher grains of water per pound of dry air at the process exit). This can reduce the comfort levels that the desiccant system is intended to improve.

FUTURE PLANS

The experimental test results have shown the performance of the desiccant system at different operating parameters. The next phase of this project will entail comparing the experimental results against algorithms to check the validity of predictive models. In addition, a proposed standard is under consideration by an ASHRAE technical committee that will be used to evaluate the performance of the tested unit at standard operating conditions. Comparisons against the proposed ASHRAE standard will improve the interpretation of the experimental results by verifying their accuracy.

All testing to date has been performed on a single manufacturer's unit. Recently, an additional unit has been obtained from another manufacturer that will enable testing desiccant system performance using a different design. The results for the two machines will be compared to determine the relative merits of each.

The next major phase of the work will be to couple the desiccant system with a microturbine generator and utilize the waste heat to regenerate the desiccant material. The main purpose will be to determine if the temperature and flow (capacity) of the waste heat is sufficient to regenerate the desiccant material. By utilizing the waste heat from the microturbine, the overall system efficiency of the microturbine is increased, which improves the potential for microturbine generator technologies.

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REFERENCES

ARI. 1998. Standard 940-98, Standard for desiccant dehumidification components. Arlington, Va.: Air-Conditioning and Refrigeration Institute.

ASHRAE. 1990. *ANSI/ASHRAE Standard 62-1989, Ventilation for acceptable indoor air quality.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1997. *1997 ASHRAE Handbook—Fundamentals.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1998. *ANSI/ASHRAE Standard 139-1998, Method of testing for rating desiccant dehumidifiers utilizing heat for the regeneration process.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

CDH Energy Corporation. 1995. Laboratory testing the Engelhard/ICC desiccant unit at ETL. Brooklyn Union Gas.

Harriman III, L.G., D. Plager, and D. Kosar. 1997. Dehumidification and cooling loads from ventilation air. *ASHRAE Journal* 39(11): 37-45.

Jalalzadeh-Azar, A.A., W.G. Steele, and B.K. Hodge. 1996. Design of a test facility for gas-fired desiccant-based air-conditioning systems. *Proceedings of the 1996 Joint Power Generation Conference,* 2: 149-161.

Jalalzadeh-Azar, A.A., W.G. Steele, and B.K. Hodge. 2000. Performance characteristics of a commercially available gas-fired desiccant system. *ASHRAE Transactions* 106 (1): 95-104.

Slayzak, S.J., A.A Pesaran, and C.E. Hancock. 1996. Experimental evaluation of commercial desiccant dehumidifier wheels. National Renewable Energy Laboratory.

Van den Bulck, E., J.W. Mitchell, and S.A. Klein. 1986. The use of dehumidifiers in desiccant cooling and dehumidification systems. *Journal of Heat Transfer* 108 (August): 684-692.

Vineyard, E.A., J.R. Sand, and D.J. Durfee. 2000. Parametric analysis of variables that affect the performance of a desiccant dehumidification system. *ASHRAE Transactions* 106 (1): 87-94.