PARAMETRIC ANALYSIS OF VARIABLES THAT AFFECT THE PERFORMANCE OF A DESICCANT DEHUMIDIFICATION SYSTEM

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ABSTRACT

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 wet bulb temperature. 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 angle, 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 (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.

INTRODUCTION

In an effort to accelerate the widespread use of desiccant dehumidification systems in commercial building applications, an industry/government Cooperative Research and Development Agreement (CRADA) was established to evaluate and test design concepts for desiccant units using advanced materials and components. The objectives of the CRADA were to: 1) significantly improve indoor air quality in buildings; 2) decrease the use of halogenated refrigerants for vapor-compression equipment; 3) reduce electrical energy consumption resulting in corresponding reductions in carbon dioxide emissions; and 4) introduce new products and markets for U. S. industry through expanded utilization of desiccant dehumidification systems. In addition, data from the CRADA will be used to support rating/certification standards under development for desiccant systems.

The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 62-1989 recommends that indoor levels of relative humidity be maintained between 30 and 60% and the ventilation rates be increased to improve indoor air quality (ASHRAE 1990). In addition, ASHRAE has introduced design dew point temperatures for 239 locations in the United States that are being used a specifiers to size and select building heating, ventilation and air-conditioning (HVAC) equipment

(ASHRAE 1997). 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. This usually requires the airstream to be reheated since overcooling occurs; clearly an unacceptable alternative in light of the today's 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 of the desiccant dehumidification system, reduce energy costs.

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

BACKGROUND

Desiccant dehumidification was used in the mid-1970s primarily for prevention of corrosion during storage or in industrial applications where products being manufactured were moisture sensitive. Energy concerns in the late 1970's led researchers to 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 1980's 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 shapes of desiccant material allowed for different amounts of moisture removal. One particular desiccant shape 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, such as national laboratories and academic institutions, are presently analyzing commercial desiccant systems to improve their performance. The federal government, along with the gas industry, is funding this research to help understand how desiccant systems operate in the field at high relative humidity conditions and to determine the effects of different design parameters on their performance.

The desiccant dehumidification system process, shown in Figure 1, 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 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 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 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 rotating thermal wheel, it picks up heat from the process air stream prior to entry into the regeneration heat exchanger (6). Air passing through the regeneration heat exchanger 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).

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 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 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 varied from 180 to 230EF (82.2 to 110EC), 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 2. 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 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 in a psychometric computer model 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 +/- 0.5EF (0.3EC) 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 +/- 0.5EF (0.3EC) 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:

$$\text{Latent Capacity} = m_{AIR} \times (f_{\mathrm{IN}} - f_{\mathrm{OUT}}) \times h_{\mathrm{WATER}} \times 60 \qquad (1)$$

where m_{air} represents the mass flow rate of air, f represents the absolute humidity ratio, and h is the heat of vaporization for water. The 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 3) which calls for the process and regeneration air inlet conditions to be controlled at 95EF (35EC) dry bulb temperature and 75EF (23.9EC) 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. Future tests will be conducted at the second condition (Table 3) which is representative of conditions where most units will operate during the year. The test unit was operated in a steady-state mode (no boiler cycling) in order 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 190EF (87.8EC). 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 Anext 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 4, 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 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 2 -7. 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 2 and 3 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 1. 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 1. 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 are balanced (equal) for all the tests.

Figures 4 and 5 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 4 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 190EF (87.8EC), which is probably too low for the modified test system with 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 4 to flatten out. Limitations in the capabilities of the test unit prevent reaching this flow rate. If higher air-flow rates were possible, the heavily loaded wheel should show an increase in capacity at some point.

The curves in Figure 5 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 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 wheels as regeneration temperatures are increased. Data plotted in Figure 6 indicates that the latent capacity is slightly higher for the heavily loaded desiccant wheel at regeneration temperatures below 200EF (93.3EC). 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 200EF (93.3EC). 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 - 220EF (93.3 - 104.4EC) appear to give the best combination of increased capacity at near optimal efficiency.

CONCLUSIONS

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 **Aventilation 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 Alatent air conditioning@, it is inappropriate to compare it to other, more conventional approaches based on total cooling B 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 Btu=s or tons 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.

FUTURE PLANS

The initial tests were performed for a desiccant dehumidification system operating at the extreme design condition for most field installations, which is only experienced a small fraction of the time. In order to better evaluate the performance of desiccant dehumidification systems for comparison against vapor compression air-conditioning systems, future plans include investigating performance at higher relative humidity conditions, such as the second condition in ARI standard 940-98 (Table 3). Condition 2, which is more representative of the conditions that the unit will operate for the majority of the year, calls for rating performance with both the process and regeneration air inlet conditions controlled at 80EF (26.7EC) dry bulb temperature and 75EF (23.9EC) wet bulb temperature. This particular condition has a much higher latent load, which is where desiccant dehumidification systems should show a larger advantage when compared to conventional techniques of overcooling/reheating air to control the latent load.

In addition to testing at higher relative humidity conditions, future plans call for investigating ways to increase the regeneration temperature in order to improve the latent capacity of the desiccant dehumidification system. One of the main design improvements that will be investigated is a direct-fired regeneration heater. This type of heater has several advantages compared to the conventional water-heater/heat exchanger combination that is presently used in some systems. The main advantage is that the regeneration temperature is not limited to approximately 200EF (93.3EC). This upper limit results from the fact that the water temperature must be kept below the boiling point to avoid the expense and restrictions associated with pressure vessel codes. A direct-fired heater can reach much higher temperatures and thus improve the latent capacity of the system. Other advantages include: 1) eliminating the heat exchanger losses; 2) reducing the fan power losses associated with the pressure drop across the heat exchanger; and 3) eliminating the secondary heat transfer fluid pump power.

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TABLE 1

Baseline and Parametric System Operation Values

Operational Parameters	**Baseline Values**	**Parametric Variations**
Process/Regeneration Dry Bulb Temperature	95EF (35EC)	----
Process/Regeneration Wet Bulb Temperature	75EF (23.9EC)	----
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	190EF (87.8EC)	180 - 230EF (82.2 - 110EC)

TABLE 2

Desiccant Test Instrumentation

MEASUREMENT	**SENSOR**	**PRECISION/ACCURACY**
Temperature	Averaging RTD	=/- 0.24% at 70EF (21.1EC) Range = -50 to 275E F (-45.6 to 135EC)
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: +/- 0.2EC (-80 to 95EC)
Power	Watt Transducer (2)	+/- 0.5% of full scale Range = 0 to 40000 watts 0 to 500 watts

TABLE 3

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	95EF (35EC)	75EF (23.9EC)	95EF (35EC)	75EF (23.9EC)
2	80EF (26.7EC)	75EF (23.9EC)	80EF (26.7EC)	75EF (23.9EC)
3	80EF (26.7EC)	67EF (19.4EC)	95EF (35EC)	75EF (23.9EC)
4	45EF (7.2EC)	45EF (7.2EC)	80EF (26.7EC)	75EF (23.9EC)

* The tolerance for all temperatures during the test is +/- 0.5EF (+/- 0.3EC)

TABLE 4
Thermal Wheel Carryover Effects

Wheel Speed (rpm)	Carryover (grains/lb$_{DRY\ AIR}$) Evaporative Pad OFF	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)

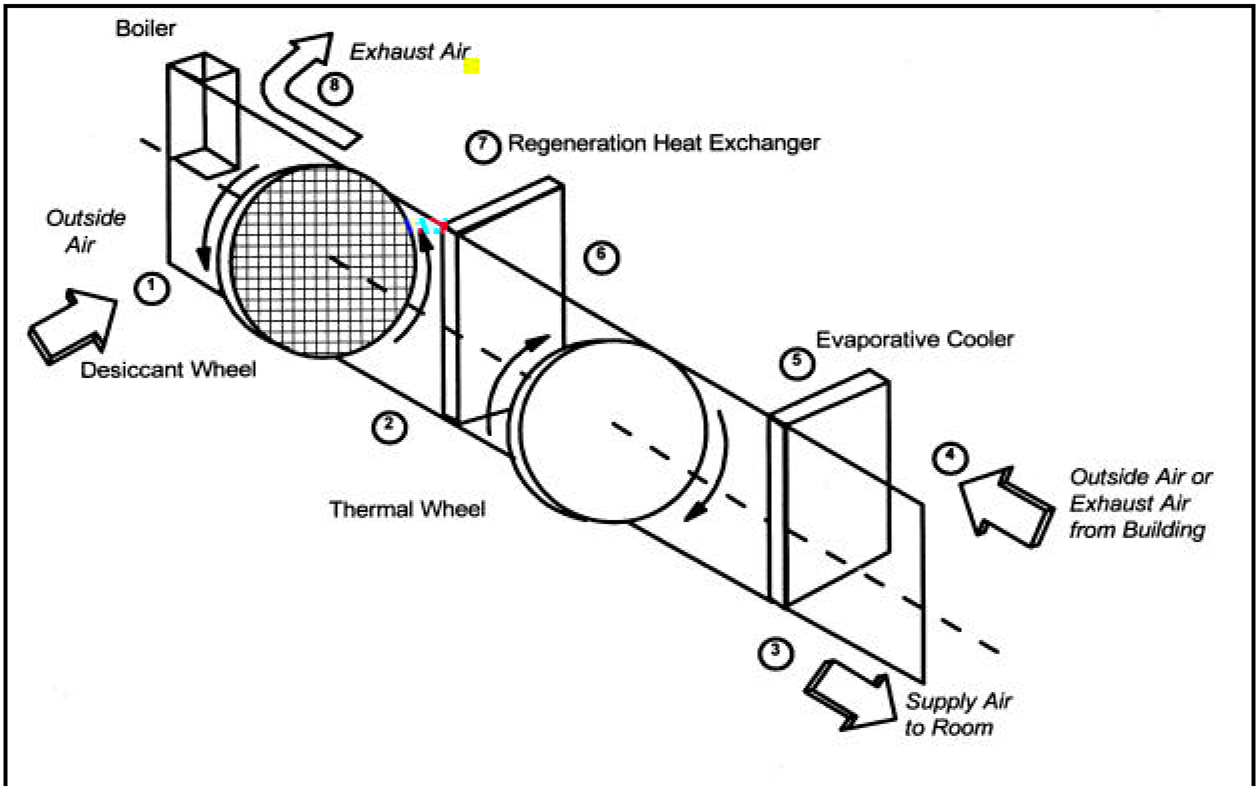

Figure 1 *Desiccant dehumidification system schematic.*

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

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

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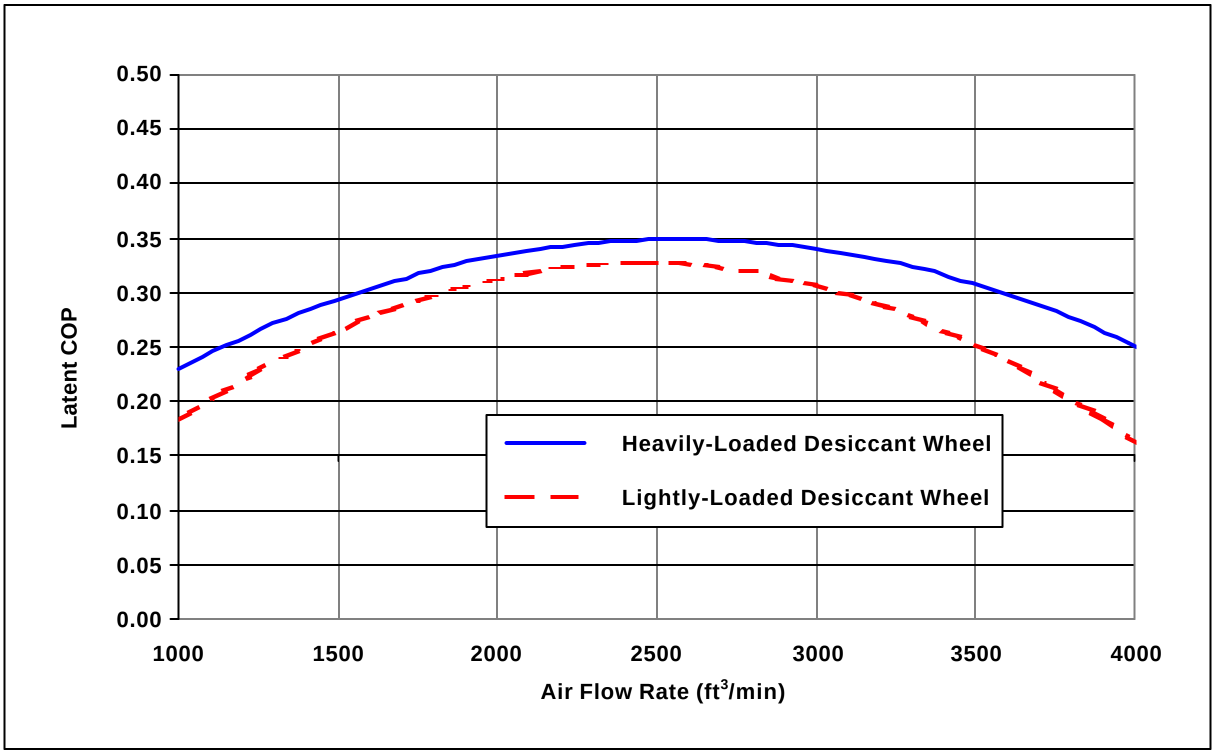

Figure 5 *Air flow rate vs. latent COP.*

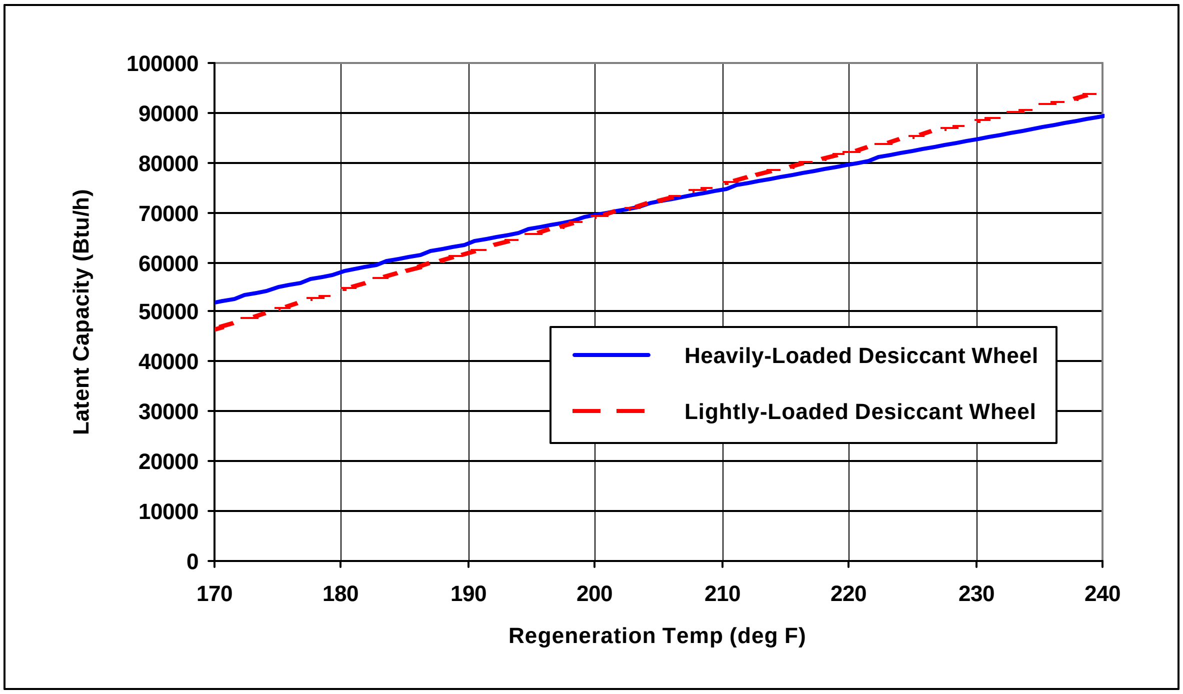

Figure 6 *Regeneration temperature vs. latent capacity.*

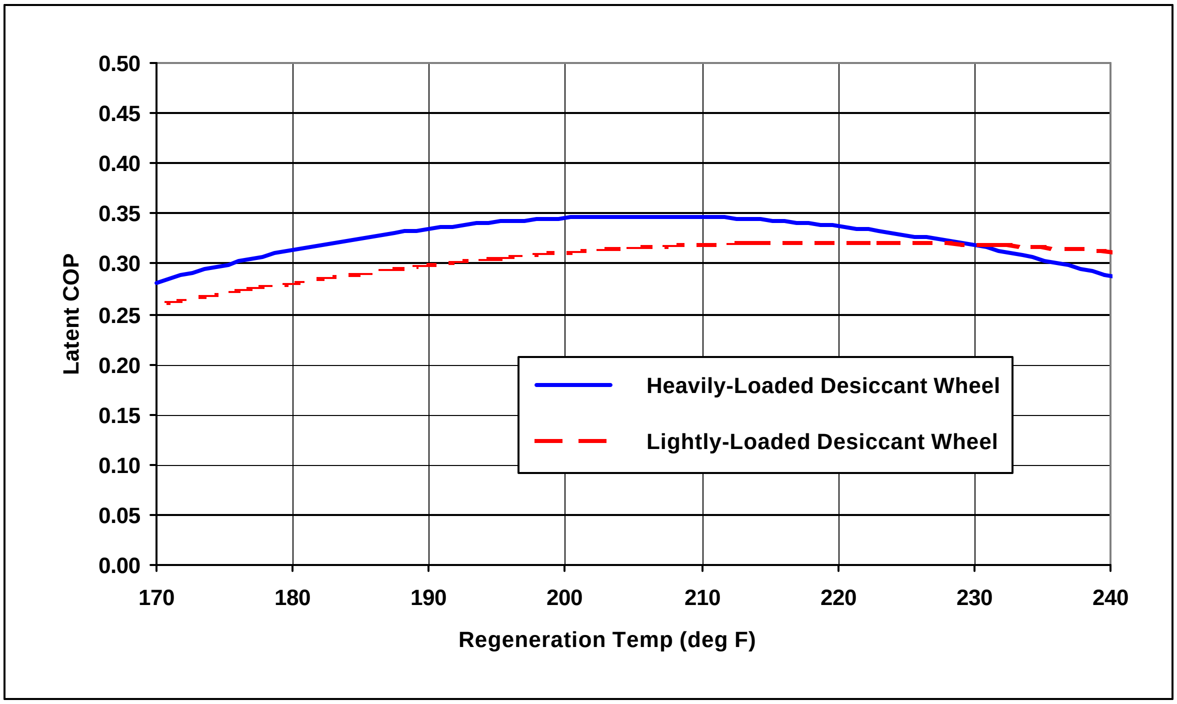

Figure 7 *Regeneration temperature vs. latent COP.*