NREL/TP-471-21167 · UC Category: 1600 · DE96013096

Experimental Evaluation of Commercial Desiccant Dehumidifier Wheels

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Prepared for the
Ab-Sorption Heat Pump Conference '96
Quebec, Canada

September 17-20, 1996

National Renewable Energy Laboratory
1617 Cole Boulevard
Golden, Colorado 80401-3393
A national laboratory of the U.S. Department of Energy
Managed by Midwest Research Institute
for the U.S. Department of Energy
under Contract No. DE-AC36-83CH10093

Prepared under Task No. BE513001

May 1996

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EXPERIMENTAL EVALUATION OF COMMERCIAL DESICCANT DEHUMIDIFIER WHEELS

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ABSTRACT

The National Renewable Energy Laboratory is currently characterizing the state-of-the-art in desiccant dehumidifiers, the key component of desiccant cooling systems. The data are being obtained in our HVAC Equipment Test Facility in accordance with the proposed ASHRAE test standard. The experimental data will provide industry and end users with independent performance evaluation and the United States Department of Energy and NREL with the information necessary to assess advances in the energy savings potential of the technology. This paper proposes several figures of merit for evaluating performance. The results of these tests indicate that dehumidification capacity performance parameters can be correlated to process inlet air relative humidity.

INTRODUCTION

In 1990 about 4.1 EJ (3.9 quads) of primary energy were used to air condition buildings. This energy end use is on the rise and is expected to increase as the population shifts to the warmer southern states [1]. This presents the air conditioning industry with several challenges. Among these are demands for increased energy efficiency and improved indoor air quality, growing concern for improved comfort and environmental control, increased ventilation requirements, phase-out of chlorofluorocarbons (CFCs), and rising peak demand charges. New approaches to air conditioning are being evaluated to resolve these economic, environmental, and regulatory issues. Desiccant cooling and dehumidification, a technology known for some time, is providing important advantages in solving many of these problems. As a result, the use of desiccant cooling and dehumidification systems for building comfort conditioning has increased steadily during the past several years. Recent advances in sorptive materials, in conjunction with dehumidifier design innovations, are making the technology increasingly attractive.

Desiccant Cooling System Operation

The dehumidifier is the heart of a desiccant cooling system. It efficiently removes the moisture (latent load) from the process air; the temperature (sensible load) of the dried air is then reduced to the desired comfort conditions by sensible coolers (i.e., rotary heat exchangers, direct and indirect evaporative coolers, cooling coils). The latent and sensible loads are handled more efficiently than in vapor compression cooling equipment because the components are optimized to independently remove these separate loads. The desiccant in the dehumidifier is regenerated (reactivated) when heat is applied to release the moisture, which is exhausted outdoors. The heat for regeneration can be provided from a number of energy sources such as natural gas, waste heat, solar, and off-peak electricity.

The desiccant can be either solid or liquid. This work focuses on solid desiccant dehumidifiers, in which the process air to be dried is passed through a porous, desiccant-laden matrix. Water vapor is adsorbed into the desiccant, driven by the vapor pressure differential between the process air and the desiccant surface. When the desiccant is nearly saturated, hot air is passed through the bed to release the moisture. The desiccant matrix typically takes the form of a rotor (wheel) so that it may be conveniently rotated between the process and regeneration airstreams.

History of Desiccant Dehumidifiers

Through the mid-1970s, desiccant dehumidifiers were primarily used for dehumidification in specialty industrial applications such as the manufacture of moisture-sensitive products (pharmaceuticals, electronic components, etc.) and the prevention of corrosion or other moisture damage during storage. In the late 1970s, public concern for energy issues led investigators to focus new attention on desiccant dehumidification for commercial and residential air conditioning applications. Most of the desiccant dehumidifiers at the time (e.g., Bry Air) used packed beds of silica gel or other desiccant particles. Such packed beds, however, induced high pressure drops which required too much fan power to be considered for air conditioning applications [2].

For many years, Cargocaire Engineering had been using Honeycomb® rotors formed from porous paper or fiberglass sinusoidal flutes impregnated with lithium chloride or molecular sieves. American Solar King developed and introduced its own lithium chloride corrugated wheels in the early 1980s [3]. Around this time, lithium chloride wheels were first being used for dehumidification in supermarkets. These fluted wheels had an advantage over packed beds in that they exhibited relatively low pressure drops while still exposing the process air stream to large desiccant surface areas. This rotor geometry was better suited to air conditioning applications.

In the 1980s, research focused on developing better materials and wheels with laminar flow geometries to minimize pressure drop. One perceived problem for lithium chloride wheels was their potential for weeping at high relative humidities. Silica gel wheels did not have this problem, and became the material of choice for a number of years. Bharathan et. al. [4] fabricated and tested silica gel coated parallel passage wheels. Schultz [5] compared their data with numerical/theoretical models. Siebu Giken and several other far eastern companies developed and marketed a new class of silica gel dehumidifiers made with fiber-reinforced paper that forms sinusoidal air passages. Although silica gel was a more reliable material in the field, it was not optimized for air conditioning applications. Optimized desiccants were needed.

In the early 1990s, Munters Cargocaire added an enhanced performance titanium-treated silica gel wheel to its product line. Other manufacturers have pursued a concept proposed in 1986 by Collier [6], that the desirable desiccant for air conditioning applications should have a "Type 1M" isotherm shape. At moderate relative humidity, such a desiccant could hold a greater percentage of its maximum moisture capacity than could silica gel. This work was funded by the Gas Research Institute (GRI) which has supported desiccant technology development for air conditioning and identification of new applications in supermarkets, restaurants, and hotels/motels since the early 1980s. LaRoche Chemical, Inc., supported by GRI, spent several years developing rotary dehumidifiers with Type 1M desiccants in a sinusoidal flow passage geometry, and has recently formed LaRoche Air Systems to bring them to market. ICC Technologies and Engelhard Corporation jointly developed and marketed a rotary dehumidifier that consists of titanium silicate material (considered Type 1M) with hexagonal air passages. As the improved technologies make greater gains in the commercial market, more desiccant manufacturers are introducing new dehumidifiers.

Advanced Desiccant Technology Program

In 1995, the U.S. Department of Energy established a program to assist industry in accelerating the integration of desiccant cooling technologies into broad building comfort- conditioning markets where their full energy savings and potential to enhance indoor air quality can be realized. The National Renewable Energy Laboratory (NREL) and Oak Ridge National Laboratory (ORNL) are working in concert with desiccant system manufacturers and major heating, ventilation, and air conditioning (HVAC) equipment manufacturers to reach this goal. ORNL is managing the subcontracts that currently pair desiccant cooling manufacturers with HVAC manufacturers to develop, market, and implement the next generation of this technology. NREL is conducting baseline performance testing and developing figures of merit to concisely summarize this data.

NREL is currently establishing a database at manufacturer-specified operating parameters over a range of inlet air conditions that end users can use to make informed decisions. Because desiccant cooling and dehumidification system operation is substantially different than that of traditional equipment, the public must be provided with an unbiased evaluation of performance and benefits if it is to accept this promising technology. Broad performance maps developed through the NREL testing program will also allow designers to quickly evaluate HVAC system flexibilities provided by recent desiccant material and wheel design improvements.

To this end, NREL has obtained several desiccant dehumidifier wheels from a number of major desiccant equipment manufacturers and suppliers. These wheels are being tested in accordance with the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) proposed national standard [7]. Several figures of merit can be employed. Some describe dehumidifying capacity while others pertain to regeneration energy use. Because thorough treatment of either requires extensive discussion, we will only examine capacity figures here. We have attempted to present these figures in equitable and clear fashion for the reader to decide which are critical to their particular application.

EXPERIMENTAL APPARATUS

NREL's HVAC Equipment Test Facility was built to test a range of HVAC equipment including dehumidifiers, heat exchangers, heat pipes, and essentially any device that requires two independently monitored and controlled airstreams at design flowrates, temperatures, and humidities. Airstreams exiting the

Test Unit	Wheel Diam. cm (in)	Wheel Depth cm (in)	Wheel Vol. m^3 (ft^3)	Process Face Area cm^2 (ft^2)	Regen. Face Area cm^2 (ft^2)
New Technology	125 (49)	14.6 (5.7)	0.178 (6.28)	5930 (6.38)	5930 (6.38)
Conventional Design	46 (18)	19.7 (7.7)	0.033 (1.16)	1071 (1.15)	298 (0.32)

Table 1 - Wheel Dimensions

device under test are also monitored to provide a complete evaluation of the unit's performance and to allow calculation of moisture mass balance, an indicator of steady state operation. Heating is provided by electric resistance heaters with a maximum total output of 42 kW, and humidification is achieved by steam injection from a 50 kW dedicated boiler. The current apparatus is an upgraded version of equipment originally designed to examine experimental rotary desiccant subsystems that operate on airflows of around 17 m^3/min (600 SCFM) and regeneration temperatures as high as 90°C [8]. The study of current gas-fired dehumidifiers required the upgrade to allow for regeneration temperatures between 80 and 140°C and some airflows in excess of 56.6 m^3/min (2000 SCFM). The HVAC Equipment Test Facility is part of NREL's unique capability and, to the best of our knowledge, is the only public laboratory in the United States capable of full-scale commercial dehumidification equipment testing. A description of the modifications required to attain these conditions follows.

To physically accommodate the widely varied dimensions (Table 1) of test wheels from several manufacturers, a very flexible interface is required to connect them to the flow conditioning loops. Each wheel arrives from the manufacturer pre-installed in a housing (cassette), which contains the appropriate circumferential and wheel face seals as well as the drive motor and belt. We decided on a modular plenum/flex-duct approach, which offers the benefits of short cassette exchange times (1-4 hr) and preservation of seal integrity between regeneration and process flows during exchange. Custom plenums are fabricated from sheet metal, screwed to the cassette, and sealed with silicone. The plenums are then connected to the flow conditioning loops by flexible ducting capable of withstanding the extreme temperatures and preventing moisture transport into or out of the system. The entire desiccant test unit is insulated with 5 cm (2″) thick fiberglass, including the ducts up to the nozzles in both flow directions (Figure 1). Baffles are installed in the inlet plenums to prevent maldistribution of airflow at the wheel face. Also, because the air leaving the wheel is spatially nonhomogeneous, mixing vanes are installed in both outlets prior to the measurement stations. Both the baffles and mixing vanes provide the additional benefit of shielding the temperature sensors from radiative heat exchange with the wheel. The power input to the wheel drive motor is measured by a Hall effect watt transducer with an accuracy of ±5%.

Figure 1 summarizes the air condition measurements at the test section. Each measurement station consists of a cross of 6 mm (1/4") tubing that samples air from several points along two of the duct's diameters for spatially averaged humidity readings. Each cross also supports several type-T thermocouples for drybulb temperature measurement with absolute accuracy of ±0.2°C. Air samples are continuously pumped from the measurement station at the rate of 0.7 l/min (1.5 CFH) through a National Institute of Standards and Technology-traceable, calibrated model D-2 General Eastern chilled-mirror hygrometer with a dewpoint accuracy in the range of interest of ±0.15°C.

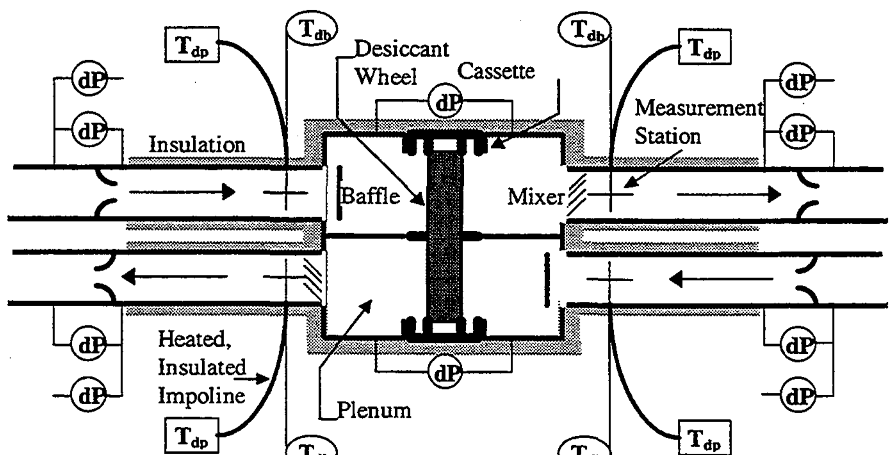

Figure 1 - Test Section Instrumentation Detail

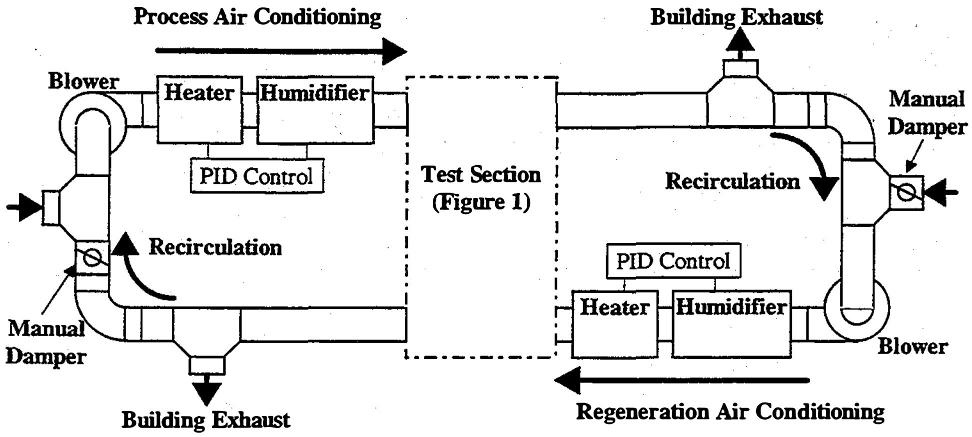

Figure 2 - Recirculation Flow Loop Configuration Used in the HVAC Equipment Test Facility

Hydrophobic Impoline sampling tubes that lead from the measurement stations to the hygrometers prevent moisture exchange with the tube walls. These tubes are wrapped with variable output heat tape and insulation to ensure that condensation cannot occur through heat exchange with laboratory ambient temperature prior to the humidity sensor. When dewpoint temperature nears ambient temperature, the sensor body itself is heated and its temperature self-regulated.

New nozzles were constructed and installed to American Society of Mechanical Engineers (ASME) specifications upstream and downstream of the test section in both flow loops. Nozzle throat diameters range from 10 to 20 cm (4 to 8″) to measure the full range of design flowrates. All pressure transducers are capacitance-type and calibrated by NIST-traceable procedures to accuracies of ±0.5% before being incorporated into the HVAC Equipment Test Facility. Differential pressure drop across the wheel is measured on both the process and regeneration sides to assess the fan power required to attain the measured performance. Differential pressures are measured between laboratory ambient and the nozzle inlets and combined with a measured laboratory ambient absolute pressure for use in calculating pressure-dependent fluid properties in the duct. Flowrates are subsequently calculated by standard ASME procedure with an absolute accuracy of ±3%.

Substantial increases in the heating/humidifying capacity of the facility have been effected with minimal capital investment through the use of a recirculation concept diagrammed in Figure 2. Recirculation directs hot processed air into the regeneration blower, providing lift of up to 70°C above ambient. This value could be substantially increased by exchanging the fan blades for high-temperature models if necessary. At the same time, hot, wet regeneration air exiting the wheel is fed to the process blower, which can then make use of the free humidification. This allows moisture, which is generated in limited supply by a boiler, to be reused, substantially raising the system's humidifying capacity.

The recirculation ducting design permits rapid reconfiguration to meet varied experimental requirements. Although not shown in the diagram, exiting process and regeneration flows can also be fed into their own inlet streams; alternatively, either exit flow can be directed back to both blower inlets to provide still more conditioning flexibility. Increases in air heating/humidifying capacities were accomplished without bringing new electrical power to the laboratory. Two 7.5-hp blowers capable of maintaining a 2.2 kPa (9″w.g.) head at 56.6 m^3/min (2000 SCFM) were the only major capital additions, and were required to achieve the fourfold increase in flowrates.

EXPERIMENTAL METHOD

Operation of the new loop configuration is as follows. Operational parameters, including process and regeneration flowrates, regeneration temperature, and wheel speed are set according to manufacturer's specification. These values are summarized in Table 2. The regeneration air humidity ratio is set to match that of the process air. Wheel rotational speed is measured with a stopwatch to within 0.2 revolutions per hour (rph). Because strict maintenance of a constant wheel speed is not critical to performance, most designs turn the wheel by belt drive, which allows slip that contributes to a variation typically around 0.5 rph. All other parameters are scanned every 10 s, and their averages taken and recorded every minute through a PC-controlled Hewlett Packard 3497A datalogger.

Test Unit	Proc. Air Flowrate m^3/s (SCFM)	Regen. Air Flowrate m^3/s (SCFM)	Regen. Temp. C (F)	Wheel Rotation Speed rph
New Technology	0.897 (1900)	0.897 (1900)	88 (190)	18
Conventional Design	0.236 (500)	0.079 (167)	140 (285)	19

Table 2 - Manufacturer-Specified Parameters

To attain the desired inlet air conditions, base levels of recirculation are first set by manual dampers to provide temperature or humidity lift at the blower inlets. Fine control over humidity and temperature is then supplied by Proportional-Integral-Derivative (PID) controllers at the steam injector humidifiers and resistance duct heaters in both airstreams. Inlet conditions are measured every 10 s. One-minute averages (the average of six of these measurements) are maintained by this method to within ±0.3°C (Figure 3) and ±0.5 g_v/kg_{da} (Figure 4) of the drybulb temperature and humidity ratio setpoints, respectively, during the test period (40-100 min in these examples). The one-hour averaged values for drybulb temperature and humidity ratio are typically within ±0.1°C and ±0.2 g_v/kg_{da} of the setpoint. Each test consists of approximately 60 min of steady-state data. Figure 5 and Figure 6 show inlet and outlet conditions for temperature and humidity, respectively. Steady-state behavior in all parameters is observed by the time the measured values of moisture entering and exiting the test section agreed to within ±5%, which typically occurred within the time it took to reach new humidity setpoints (20-30 min). Figure 7 shows the development of moisture mass balance for these data. Figure 8 shows that the calculated performance parameter described in the following paragraph is largely unaffected by such small variations in the inlet conditions.

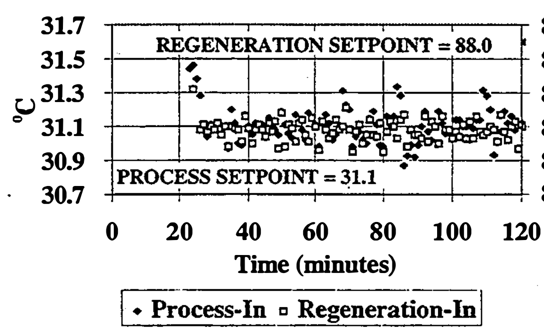

Figure 3 - Typical Inlet Air Temperature Data

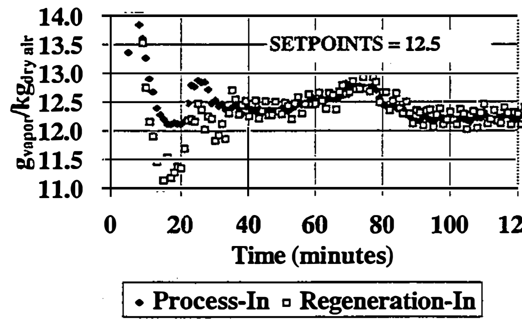

Figure 4 - Typical Inlet Air Humidity Ratio Data

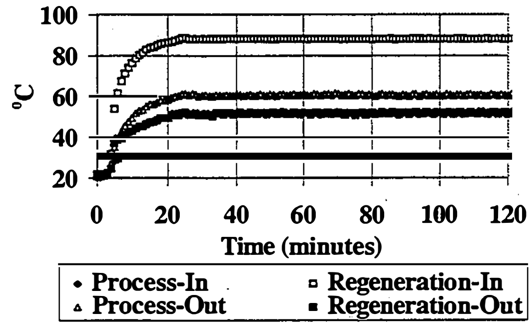

Figure 5 - Typical Air Temperature Data

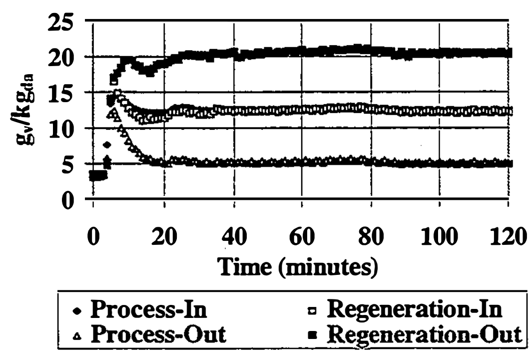

Figure 6 - Typical Air Humidity Ratio Data

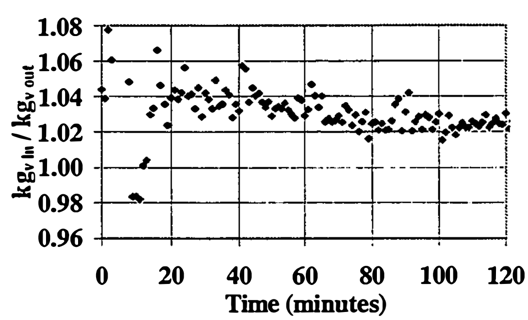

Figure 7 - Typical Moisture Mass Balance

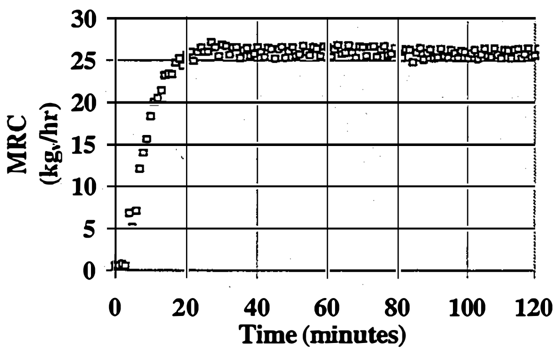

Figure 8 - Typical Moisture Removal Capacity Data Example

The primary performance indicator calculated from these data is the Moisture Removal Capacity (MRC), as described in the ASHRAE proposed national standard method of test [7]. MRC is typically measured in mass of moisture removed per hour, but can be normalized by several parameters, including process air volumetric flowrate, theoretical fan power required to supply the design flowrates across the wheel, and wheel volume. Each normalization highlights a different aspect of performance. MRC is calculated by the following equation.

$$MRC = \dot{m}_{PI}\left[w_{PI} - w_{PO}\right]$$

Dry air mass flowrate, although not currently included in the ASHRAE Standard, is technically the correct term to use here and improves the accuracy of the calculation by up to a 2%. Our method of uncertainty analysis [9] for this calculation yields a nominal accuracy of ±5% based on the accuracies of the individual measurements at 35°C and 40% relative humidity.

RESULTS AND DISCUSSION

Figure 9 shows gross MRC for both wheels over a range of relative humidity conditions. As shown in Table 1 and Table 2, the new technology wheel is physically the larger of the two and operates on nearly four times the process flowrate. As such, it is expected to remove the most moisture and clearly does so with an MRC of four to five times that of the conventional design. We have used process air volumetric flowrate to normalize the MRC for a more meaningful comparison. Looking at MRC on a per-volume-flowrate basis yields Figure 10. On this basis, the new technology exhibits a 15-40% higher capacity than the conventional design.

Because MRC is the product of process air mass flowrate and absolute humidity depression, normalizing by air volume flowrate essentially gives the humidity drop across the wheel. By applying the appropriate constants to MRC/Q we can plot a second axis on this graph showing the corresponding absolute humidity depression produced by each unit. Because the air density decreases by about 2% from 30 to 70% rh, however, this is not an exact calculation. Figure 11 shows the resulting process outlet air temperatures. Process inlet air temperature is 35 °C in all cases.

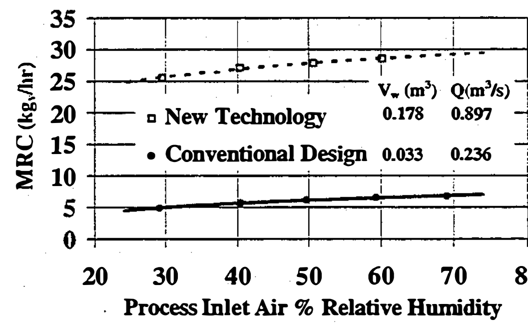

Figure 9 - Gross Moisture Removal Capacity (note differences in wheel size and flowrate)

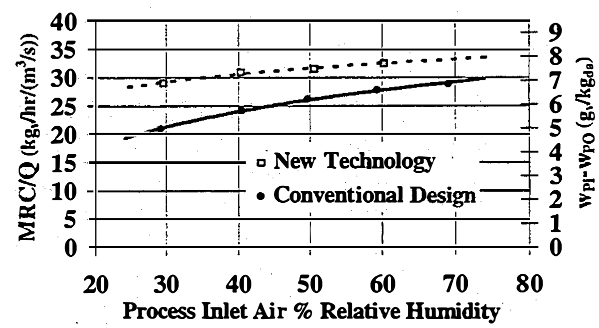

Figure 10 - Moisture Removal Capacity Normalized by Process Air Volume and Mass Flowrates

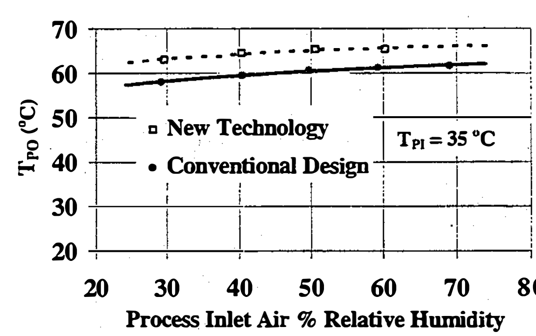

Figure 11- Process Outlet Air Temperature

The notion of correlating dehumidification performance to process inlet air relative humidity is potentially a very useful one, as it implies that a relatively limited data set can be used to characterize capacity over a broad envelope of operating conditions. If valid, this concept substantially reduces the burden placed on experimental methods by time constraints. This method can be useful only if two conditions are met. First, data must follow the relative humidity correlation irrespective of temperature and absolute humidity levels, within the domain of reasonably anticipated ambient conditions. Second, if the correlations are to be used to predict performance at various flowrates, the effect of this variable must also be assessed.

Preliminary experiments indicate that MRC and MRC/Q collapse well in the range of 20-35 °C process inlet air temperature. Inlet absolute humidity ranged from 5.3 to 12.4 g/kg at 20 °C, and 12.6 to 30.0 g/kg at 35 °C. Preliminary data also show a minimal variation of MRC/Q with process flowrate. These data were taken at 28 °C and 68% relative humidity. Gross capacity (MRC) decreases by 44%, but normalized performance (MRC/Q) increases by only 7% when the process flowrate is cut in half. This phenomenon could be explained by heat/mass transfer theory in that the Number of Transfer Units (NTU) must increase with decreasing volumetric flowrate. The effect is very small, however, nearly within experimental uncertainty over a large flow turndown. Further investigation of process air temperature and flowrate is necessary to fully quantify their effects on performance parameters.

Still another performance comparison can be made by noting the disparity in process-to-regeneration flowrate ratios. The conventional technology operates with a 3:1 ratio, while the new design uses proportionally more regeneration airflow at a 1:1 ratio, potentially incurring disproportionate additional pressure drops and fan power requirements. Further analysis indicates otherwise.

One measure of a wheel's fundamental fluid dynamic characteristics is pressure drop normalized by wheel depth and face velocity. This criterion is not used here, however, as hydraulic power (summation of process and regeneration pressure drop-volume flowrate products) is a much more compelling point in practice. This factor represents the theoretical fan power required to supply the design flowrates across the wheel. Values for MRC normalized by theoretical fan power are given in Figure 12. The new technology does not suffer from excessive pressure drop, and in fact provides 40-60% more dehumidification capacity in this category as well.

Because the unit's physical size is an issue in the field, both due to first cost and space constraints, it is also useful to examine performance on a per-unit-volume-of-desiccant-rotor basis. Figure 13 shows MRC normalized by wheel volume. In this light, the conventional wheel provides 0-25% more capacity. This can be partially explained by the difference in regeneration temperatures.

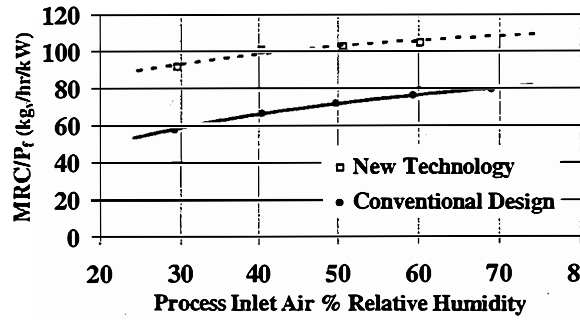

Figure 12 - Moisture Removal Capacity Normalized by Theoretical Fan Power

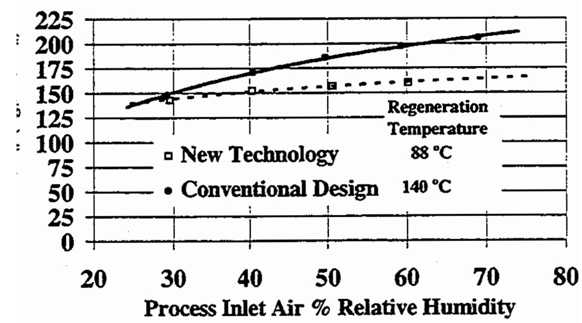

Figure 13 - Moisture Removal Capacity Normalized by Wheel Volume

The previous figure shows gross capacity per unit wheel volume. As we have noted, normalizing gross capacity by volume flowrate (MRC/Q) gives a measure of absolute humidity depression. This parameter can also be normalized per-unit-wheel-volume. Figure 14 shows the conventional design produces four to nearly five times more humidity depression per unit of matrix volume. Again, disparate regeneration temperatures could explain the result of this normalization.

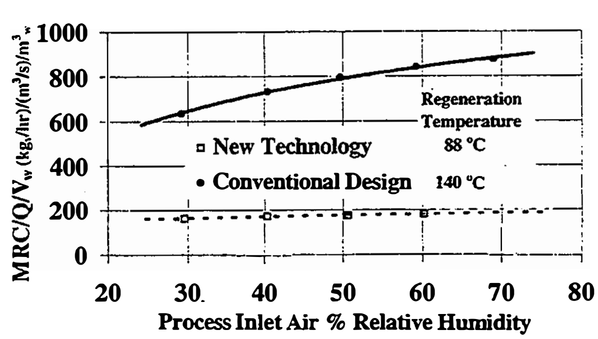

Figure 14 - Moisture Removal Capacity Normalized by Process Air Volume Flowrate and Wheel Volume

CONCLUSIONS

As part of DOE's Advanced Desiccant Technology Program, commercial desiccant dehumidifier wheels are being tested at NREL's HVAC Equipment Test Facility in accordance with the proposed ASHRAE test standard. The units are supplied with air conditioned to various temperature and humidity combinations at manufacturer-specified flowrates and regeneration temperatures. Several performance figures of merit are proposed for public discussion.

Performance can be determined on the basis of gross Moisture Removal Capacity (MRC) measured in kg/hr, capacity normalized by flowrate measured in m^3/s (MRC/Q), capacity normalized by theoretical fan power (MRC/P_f), and MRC per unit of desiccant wheel volume (MRC/V_w). MRC/Q is essentially a measure of absolute humidity ratio drop in the process air and can be converted to units of kg_v/kg_{da}. Experience gained during the execution of these experiments is providing practical insights for evaluating ASHRAE's currently proposed method of test.

Performance parameters are shown to correlate with the relative humidity of process inlet air. Initial tests indicate that the parameters can be correlated in this way over a representative range of geographic temperature and absolute humidity combinations. These correlations may then be applied with confidence as long as process inlet air relative humidity is known and units are operated at or near manufacturer-specified mass flowrates.

ACKNOWLEDGMENTS

Funding for NREL's Advanced Desiccant Technology Program was provided by the DOE's Office of Buildings Technology. The authors thank Ron Fiskum, DOE Program Manager, and Ren Anderson, NREL Technology Manager, for their support. We would also like to thank Doug Powell and Greg Barker for their contributions in the laboratory.

NOMENCLATURE

dP	differential pressure	Pa
MRC	Moisture Removal Capacity	kg_{vapor}/hr
$\dot{m}$	mass flowrate	$kg_{dry\ air}/hr$
Q	volume flowrate	m^3/min
P_f	theoretical fan power	kW
rh	relative humidity	%
V_w	wheel volume	m^3
w	absolute humidity ratio	$kg_{vapor}/kg_{dry\ air}$ or, $g_{vapor}/kg_{dry\ air}$

Subscripts

da	dry air
db	dry bulb
dp	dew point
PI	Process Inlet
PO	Process Outlet
RI	Regeneration Inlet
RO	Regeneration Outlet
v	water vapor
w	wheel matrix
wa	wet air

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