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Technical Report

Desiccant Dehumidification Wheel Test Guide

S.J. Slayzak and J.P. Ryan

National Renewable Energy Laboratory

1617 Cole Boulevard
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Introduction

Desiccant cooling systems are energy efficient and environmentally benign. According to one estimate, desiccant dehumidification could reduce total residential electricity demand by 25% or more[1] in humid regions, providing a drier, cleaner, more comfortable indoor environment with a lower energy bill. Desiccant systems allow more fresh air into buildings, thus improving indoor air quality without using more energy. Desiccant systems also displace chlorofluorocarbon-based cooling equipment, the emissions from which contribute to the depletion of the Earth's ozone layer.

When fresh outdoor air is brought into a building, it often carries a high humidity load relative to the building's internal latent load. Conventional vapor-compression cooling systems are not suited to efficiently treat large humidity loads. To sufficiently dry the air in many applications, vapor-compression systems must be operated at low temperatures, which reduces their efficiency and results in inefficient reheating of the dry, cold air to achieve some degree of comfort. Additionally, matters are made worse by common use of oversized compressors controlled by dry-bulb set points. This leads to short-cycling, which can reintroduce condensate from a wet cooling coil back into the supply air .

Currently, desiccant cooling and dehumidification systems are being used successfully in industrial and various commercial markets and provide clear advantages in many applications throughout the United States. Desiccant cooling systems are used to improve the indoor air quality of all types of buildings by efficiently controlling moisture in large quantities of fresh, ventilation air. In these systems, a desiccant removes moisture from the air via a process called sorption, which releases heat and increases the air temperature. A combination of heat exchange with ambient air and evaporative or conventional cooling coils then cools the dry air. Temperature and humidity loads are very effectively and efficiently met by separating them in this way. The desiccant is then dried out (regenerated) to complete the cycle using thermal energy supplied by natural gas, waste heat, or the sun. Commercially available desiccants include silica gel, activated alumina, natural and synthetic zeolites, titanium silicate, lithium chloride, and synthetic polymers. An excellent summary of desiccant technology and applications can be found in *The Desiccant Dehumidification Handbook*, produced by the Munters Corporation of Amesbury, Massachusetts.

The desiccant wheel is at the heart of these systems, providing large surface areas for desiccant-to-air contact at pressure drops suitable for HVAC application. Two national standards have recently been developed for testing and rating. They are:

- American Society of Heating, Refrigerating and Air-conditioning Engineers MOT Standard 139—"Method of Testing for Rating Desiccant Dehumidifiers Utilizing Heat for the Regeneration Process."

[1] Houghton, D.J., R.C. Bishop, A.B. Lovins, and B.L. Stickney, with J.J. Newcomb and B.J. Davids (August 1992). *State of the Art Technology Atlas: Space Cooling and Air Handling.* Boulder, Colorado: E-Source, Inc.

- Air-conditioning and Refrigeration Institute Rating Standard 940—"Desiccant Dehumidification Components."

ARI is also in the process of developing its Certification Program Operational Manual implementing these two standards. This Desiccant Dehumidification Wheel Test Guide is intended to facilitate their use by certification labs and manufacturers. It is a product of more than 20 years of experience gained at the National Renewable Energy Laboratory's (NREL) desiccant research facilities. The Test Guide details practical experimental experience with rotary mass exchangers in relation to the standards, and is aimed at developing this testing expertise in industry quickly and cost-effectively.

A Desiccant Dehumidifier Wheel Test Method and Rating Workshop took place February 24–26, 1999, at NREL's Advanced HVAC Test Facility where the Test Guide was presented to industry. The workshop was co-sponsored by ARI, the Gas Research Institute (GRI), and the U.S. Department of Energy's Office of Building Equipment Technology (DOE/OBT). The workshop supported the co-sponsors' goal of accelerating desiccant technology's transition to widespread use. As a result of the workshop, several manufacturers and certification labs across the country have made improvements in rotor test capability. Typical areas where extra attention has been required include airflow measurement, humidity measurement, and rotor-face pressure differentials.

This Test Guide describes performance figures of merit that are useful in evaluating rotary dehumidification equipment and practical advice on how to successfully measure the physical parameters needed for calculating these figures. This Guide also calculates representative limits of uncertainty for these figures, giving experimentalists a reasonable sense of the maximum accuracy they can expect from good data in this field. This is necessary to prevent test results from being applied in ways that are not justified by the experimental method. Finally, we offer safeguards for testing to avoid damage to equipment and researchers.

Definitions

Definitions follow industry standards outlined in the ASHRAE Terminology of Heating, Air-conditioning, and Refrigeration. New definitions currently under consideration by ASHRAE Technical Committee 3.5, Sorption and Desiccant Technology, are described in Appendix 1a. Other terminology used in this document is included in Appendix 1b. Nomenclature for equations is in Appendix 2.

Performance Figures of Merit

In the Standards

Standard 139 defines two primary figures of merit for comparing desiccant wheel performance. They are Moisture Removal Capacity (MRC), referred to here as "performance," and Regeneration Specific Heat Input (RSHI), referred to here as "energy efficiency." MRC is presented as mass of moisture removed per hour, (lbs/hr or kg/hr), and RSHI as hourly regeneration energy supplied to the device, normalized by MRC, (kBtu/lb or kJ/kg).

$$MRC = \rho_{std} \cdot 60Q \cdot \frac{1}{7000} \Delta GPP \tag{1}$$

$$RSHI = \dot{E}_{regen} \big/ MRC \tag{2}$$

where:

MRC	=	moisture removal capacity, lb/hr
ρ_{std}	=	standard density of air, 0.075 lb/ft^3
Q	=	process air volume flow rate, (ft^3/min)
ΔGPP	=	absolute humidity depression of the process, grains/lb
RSHI	=	regeneration specific heat input, kBtu/lb
$\dot{E}_{regen}$	=	thermal energy input rate, kBtu/hr.

The Standard 940 rating is concerned with MRC only. Standard 139 requires the acquisition of more than 30 data points per test, allowing the calculation of several other relevant figures of merit that NREL has researched. Standard 139 also describes one "figure of merit" that rates the test itself rather than the device being tested. That figure is Moisture Mass Balance, defined as:

$$Moisture\,Mass\,Balance = MRC / MRR\ , \tag{3}$$

where MRR, Moisture Removal Regeneration, is analogous to MRC, but is calculated using regeneration flow rate and grain pickup across the wheel. It confirms that the measured adsorption on the process side matches the measured desorption during regeneration, and it must fall in the range of 0.95–1.05 for a test to be considered valid. This DOES NOT imply that the MRC is known to within five percent; the acceptable range is empirical, based on decades of collective industry experience. It is a tough standard to satisfy because of the inherent difficulty in psychrometric measurement, but a balance outside this range indicates a condition in the system that must be corrected. It is also important to calculate the balance as defined—do not sum inlet moisture fluxes and compare to outlet fluxes ($\dot{m}_{in} \big/ \dot{m}_{out}$) as this results in a ratio of sums rather than differences. This is a much easier ratio to balance and definitely allows testing under conditions that could seriously misrepresent wheel performance.

For Wheel Designers

Some of the fundamental physical parameters describing rotary heat/mass exchangers, sometimes classified as regenerators, are summarized in Table 1. Residence time and the basic mass transfer parameters are critical. Residence time combines the effects of face velocity, open area, and wheel depth. Overall mass transfer is governed by driving potential, airside transfer coefficient, diffusion within the desiccant, and surface area. Thermal and material sciences are used together to optimize these parameters.

The driving potential is the difference in partial pressure of water vapor between the air and the surface of the desiccant. Water vapor pressures in terrestrial dehumidification applications are on the order of two to five kilopascals (kPa) (0.6-1.5 in Hg) at the wheel inlet. Vapor pressures of a few hundred pascals exist locally at the wheel outlet. Vapor pressure at the desiccant surface varies with desiccant type and temperature and is on the order of hundreds of pascals.

Airside transfer coefficient is governed by fluid dynamic phenomena and is typically correlated for both heat and mass transfer to Reynolds number, Prandtl number, and geometry, including number of transfer units (NTU). NTU relates rotor surface area exposed to the thermal loads embodied in the airstreams.[2] Prandtl number is a function of air thermophysical properties. Reynolds number is a ratio of momentum to viscous forces. Air velocity enters the correlations in the momentum term. The correlations change to reflect the flow regime present in the flutes. Flute velocity determines the regime, which may be roughly categorized as laminar or turbulent. In laminar flow, viscous forces dominate, so that nearly all air motion is in the direction of the bulk flow, along the axis of the flute. In turbulent flow, momentum is strong enough to produce substantial eddies within the bulk flow that continuously mix the air as it passes through the flute. This mixing generally means turbulent flow produces higher heat/mass transfer, but in doing so, also generates higher-pressure drops. The pressure drops incurred by turbulent airflow put an unacceptable load on the face and circumferential seals and drastically increase seal wear and fan power requirements. Laminar flow keeps pressure drops within HVAC application ranges and has the added benefit of keeping the internal surfaces of the matrix relatively clean because airflow moving parallel to the flute walls tends not to deposit dirt there.

NTU is a figure of merit commonly applied to heat exchangers that can also be applied to rotary mass exchangers. It is typically defined for the thermal component as the ratio of convective heat transfer at a given matrix-to-air temperature potential to the thermal capacity of the air over that potential:

$$NTU_j = \left(\frac{h\,A\Delta T}{\dot{m}_{air} c_p \Delta T} \right)_j = \left(\frac{h\,A}{\dot{m}_{air} c_p} \right)_j \tag{4}$$

where:

---

[2] Number of transfer units, NTU, is a function of convective transfer coefficient, making the correlations recursive.

NTU	=	number of transfer units
T	=	temperature
j	=	hot or cold side of the wheel
m	=	mass flowrate of air
h	=	convective heat transfer coefficient
c_p	=	specific heat of air
A	=	convective transfer surface area.

This calculation must be performed on the hot and cold sides of the wheel (j) separately. The resulting values can be combined with the use of the parameter C*:

$$\frac{1}{NTU_{thermal,total}} = \frac{1}{NTU_{cold}} + \frac{C^*}{NTU_{hot}} \tag{5}$$

where C* is the ratio of minimum to maximum air heat capacity rates:

$$C^* = \frac{(\dot{m}_{air} c_p)_{\min}}{(\dot{m}_{air} c_p)_{\max}}. \tag{6}$$

Heat exchange effectiveness for a direct counterflow heat exchanger is then calculated with total NTU:

$$\varepsilon_{cf} = \frac{NTU_{thermal,total}}{NTU_{thermal,total} + 1} \tag{7}$$

where:

ε_{cf}	=	heat exchange effectiveness.

Heat exchange effectiveness (and thereby outlet temperatures) for a rotary exchanger is then correlated using a parameter tailored to rotary devices that represents the thermal capacitance of the matrix:[3]

$$\varepsilon = \varepsilon_{cf}\left(1 - \frac{1}{9(C_r)^{1.93}}\right) = \frac{(\dot{m}c_p)_P}{(\dot{m}c_p)_{\min}} \frac{T_{PO} - T_{PI}}{T_{RI} - T_{PI}} = \frac{(\dot{m}c_p)_R}{(\dot{m}c_p)_{\min}} \frac{T_{RI} - T_{RO}}{T_{RI} - T_{PI}}$$

$$C_r = \frac{(Mc_p\phi)_{matrix}}{(\dot{m}c_p)_{\min}} \tag{8}$$

[3] These correlations are valid for values of C_r over 0.4 (high wheel speed; temperature does not vary with rotational angle but with distance through the wheel only). This is the case for enthalpy exchangers; dehumidifiers might have a heat capacitance one-tenth this value. The concept applied here to thermal potentials is often also applied to enthalpy.

where:

M = the mass of the matrix
ϕ = its rotational frequency.

NTU for mass transfer is similarly defined:

$$NTU_{mass,j} = \frac{h_{m,j}\ A_j}{\dot{m}_j} \tag{9}$$

where:

h_m = the mass transfer analog to thermal convection coefficient h.

Mass transfer parameters are modeled analytically by heat transfer analogy or computed numerically.

Diffusion and surface area are closely related in wheel dynamics. Given the restrictions of residence time, the limitations of the former require a lot of the latter to achieve acceptable grain depression. Air is in contact with the desiccant only for a few hundredths of a second, making mass transfer for a given flute primarily a surface phenomenon. When performance depends on a single pass, surface area is critical in inherently slower processes like mass diffusion in solids. Diffusion comes into play as the desiccant/matrix slowly rotates within the same airflow; mass diffusion within the desiccant must keep the surface as dry as possible (on the adsorption side) until it can be regenerated and vice-versa during desorption.[4]

Maximizing surface area means packing a lot of matrix into as small an area as possible, which leads to flutes with small cross sections. This is convenient because laminar flow is best achieved in small flow channels. This also means matrix walls should be as thin as possible to maximize open area and keep flute velocities as low and residence time as long as possible. This too is convenient because thin walls are less likely to waste unexposed desiccant by relying on slower solid-side diffusion to utilize drying potential. Surface area as a function of matrix design is complimented by the effect chemistry can produce with desiccant pore structure. Silica gels typically have on the order of 100 million square feet of surface area within their pores for each cubic foot of material. Activated carbon has several times that volumetric surface area but has lower water vapor uptake because its pore void space is too small to hold much water.

Residence time is the result of a number of important parameters. We propose the formulation of a fundamental performance figure of merit "grain depression per unit of residence time."

[4] In slowly rotating dehumidifier wheels, solid-side diffusion can be a bottleneck to convective mass transfer, indicated by another fundamental figure of merit, Lewis number (Le) = NTU/NTU_{mass}, when it takes values greater than unity.

$$\Delta GPP_{rt} = \frac{7000}{3600\rho_{std}} \frac{MRC}{Q} \frac{V_{fl}}{d} = \Delta GPP \frac{V_{fl}}{60d} \tag{10}$$

where:

ΔGPP_{rt}	=	absolute humidity depression per second of residence time, grains/lb_{air}/s
MRC	=	moisture removal capacity, lb/hr
V_{fl}	=	flute velocity, ft/min
Q	=	process air volume flowrate, ft^3/min
d	=	wheel depth, ft.

This simultaneously normalizes tests for differing face velocities, open areas, and wheel depths and would be of interest to a rotor designer trying to maximize mass transfer per unit of desiccant contact area.

RSHI is an indicator of energy consumed by the regeneration heater. This type of figure is entirely appropriate for a standard where its primary purpose is calculating energy consumption for dehumidifiers. RSHI can be used for this purpose at the rated face velocity only and does not include the effect of a heat exchanger that can be employed at the process air outlet to recover heat of adsorption and preheat regeneration air. This configuration is commonly found in ventilation air conditioning applications. To include the effect of heat recovery, we use the term $RSHI_{HX}$.

$$RSHI_{HX} = RSHI\left[1 - \frac{\varepsilon_{HX}(T_{PO} - T_{PI})}{T_{RI} - T_{PI}}\right] \tag{11}$$

where:

ε_{HX} = the heat exchanger effectiveness.

This formulation assumes the PI and the heater receive air from the same source, and the heat exchanger is operated with balanced airflows, as is the case with many ventilation air pre-conditioners. This figure is particularly useful in comparing the efficiency of wheels with different face splits. For example, 50/50-split wheels often have very high RSHI compared to 75/25 wheels.[5] However, due to their lower regeneration temperature, 50/50 wheels are among the most efficient in terms of $RSHI_{HX}$.

Regeneration specific heat drop (RSHD) is an indicator of the energy consumed by the wheel.

$$RSHD = \frac{\dot{E}_{drop}}{MRC} = \frac{\dot{m}_{RO} c_p (T_{RI} - T_{RO})}{MRC} \tag{12}$$

[5] RSHI for 50/50 wheels is often higher than 75/25 wheels by 25%-50% or more.

RSHD focuses on the energy performance of the matrix itself by focusing on sensible energy drop in the regeneration air as it passes through the wheel rather than the energy supplied to the regeneration air. It is very nearly independent of face velocity for many wheel configurations, although there are exceptions. RSHD is also much less sensitive to mass-flow ratio than RSHI, again for many wheels but not all, and trends in the opposite direction as RSHI in some instances. Unlike high RSHI, high RSHD does not necessarily indicate reduced efficiency. High RSHD may indicate poor grain depression, as might RSHI, or it may show that the wheel is able to utilize lower temperature air for regeneration, or that the matrix is picking up a lot of heat. RSHI does not register these and other phenomena on it own. RSHD is a distinct parameter that adds to the understanding of a wheel's energy consumption characteristics.

Heat dump-back is another feature of dehumidifier wheels that becomes important when process outlet temperature is a design requirement. Some processes benefit from the sensible energy evolved from the desiccation process; for others, this represents a load that must be removed. In quantifying heat dump-back, we calculate adsorption heat ratio:

$$AHR = \frac{T_{PO,adiabatic} - T_{PI}}{T_{PO} - T_{PI}} \tag{13}$$

where:

AHR	=	adsorption heat ratio
T_{PO}	=	temperature achieved upon reaching measured grain depression.

$T_{PO,adiabatic}$ is the temperature achieved upon reaching the measured grain depression with no change in enthalpy—essentially evaporative cooling in reverse. If AHR = 1.0, the process is adiabatic. Fractional AHR indicates the degree of heat dump-back.

Table 1. Fundamental Wheel Parameters

Adsorption heat ratio (AHR)	The ratio of sensible heat gain due to adsorption to the actual sensible heat gain.	$AHR = \frac{T_{PO,adiabatic} - T_{PI}}{T_{PO} - T_{PI}}$
Convective transfer coefficient	Fundamental ratio relating heat or mass flux to driving potential.	h or h_m
Effectiveness	Ratio of temperature or enthalpy change accomplished to the potential between the inlets of a heat/mass exchanger.	$\varepsilon = \frac{\dot{m}c_p}{(\dot{m}c_p)_{\max}} \frac{T_{PI} - T_{PO}}{T_{RI} - T_{PI}}$

Table 1 continued. Fundamental Wheel Parameters

Energy drop	Sensible energy given up by the regeneration air as is passes through the wheel.	$\dot{E}_{drop} = \dot{m}_{RO} c_p \left(T_{RI} - T_{RO}\right)$
Face area	Wheel area perpendicular to the process airflow.	$A_f = \frac{\pi D^2}{4} \frac{\alpha}{100}$
Face velocity	Nominal process air velocity as it uniformly approaches the wheel.	$V_f = Q_{actual} / A_f$
Flute velocity	Actual air velocity inside the wheel channels.	$V_{fl} = Q_{actual} / (A_f \, \Omega)$
Lewis number	Ratio of heat to mass transfer convective coefficients.	$Le = NTU / NTU_{mass}$
Number of (mass) transfer units	Ratio of mass exchanger capacity relative to the load.	$NTU_{mass} = \frac{h_m \, A}{\dot{m}_{air}}$
Number of (heat) transfer units	Ratio of heat exchanger capacity relative to the load.	$NTU = \frac{h \, A}{\dot{m}_{air} c_p}$
Open area	Fraction of the wheel face area not occupied by the wheel matrix.	Ω
Residence time	Length of time air takes to pass through the wheel.	$t = d / V_{fl}$
Specific heat	Heat capacity in units of energy normalized by mass and temperature potential.	c_p
Surface area	Area within the flutes upon which convective transfer coefficients are based.	A
Wheel depth	Thickness of the wheel matrix in the direction of airflow.	d
Wheel diameter	Maximum wheel dimension perpendicular to the airflow.	D
Wheel split	Wheel face area percentage allocation for process/regeneration airflows (e.g. 75/25 or 50/50).	α/β

For Application Engineers

We have found the most useful figure of merit to be MRC normalized by volume flow rate (MRC/Q). This figure of merit is analogous to grain depression (ΔGPP). Applying a few constants converts lbs/hr/cfm to grains/lb:

$$\Delta GPP\ (grains/lb) = \frac{7000}{60\rho_{std}} \cdot MRC/Q = 1{,}555\ MRC/Q \tag{14}$$

It has the benefit of allowing comparison of wheels of various diameters, however, it still depends strongly on face velocity, and this parameter must be the same for direct performance comparison. This appears to be an acceptable compromise between rigor and practicality.

The dehumidification rate, MRC, defined in the standards in lbs/hr, can also be expressed as a cooling rate (Btu/h or tons).

$$MRC_{Btuh} \approx 0.7 \cdot Q \cdot \Delta GPP \tag{15}$$

This is an approximation, because a grain's enthalpy value is dependent on its location on the psychrometric chart. The approximation is accurate to within 5% for cases of interest. MRC_{Btuh} can then be combined with energy input rate to calculate a latent coefficient of performance.

$$COP_{latent} = MRC_{Btuh} \big/ (\dot{E}_{regen} + \dot{E}_{para}) \tag{16}$$

where:

COP_{latent}	=	coefficient of performance for latent cooling
MRC_{Btuh}	=	cooling rate equivalent to moisture removal capacity, (kBtu/hr)
$\dot{E}_{regen}$	=	thermal energy input, (kBtu/hr)
$\dot{E}_{para}$	=	parasitic energy input for fans, wheel drive, etc, (kBtu/hr)

Methods for measuring regeneration energy input are detailed in Standard 139. Parasitic energy inputs include the drive motor used to rotate the wheel and fan power required to overcome the pressure drops through the process and regeneration sides of the wheel. Fan power (in watts) can be calculated by the following equation:

$$P_f = (\dot{m}_p \cdot \Delta p_p + \dot{m}_r \cdot \Delta p_r) \big/ (\eta_{fan} \cdot \eta_{motor}) \tag{17}$$

Where pressure drop is in pascals, the mass flow rate is in kilograms per second, and one watt equals 3.41 Btu/h. This, of course, only considers pressure drop through the wheel itself, and not the balance of system.

To calculate actual primary energy consumption, natural gas combustion processes should account for combustion efficiency and a 91% distribution efficiency. Twenty-eight percent generation/distribution efficiency should be applied to determine the primary energy impact of electric-powered heaters or parasitic devices.

We recommend MRC/P_f at a given face velocity be used to quantify the tradeoff between mass transfer and pressure drop within the wheel. This figure has the advantage of accounting for the fact that some wheels use proportionately more or less regeneration air than others. The mass-flow ratio typically ranges between 0.25 and 1.0. It also accounts for the fact that flute geometry has an important effect on mass-transfer-to-pressure-drop ratio (see Figure 1).

This figure's value as a system-energy-use indicator is limited by two facts. One is that fan power is typically a considerable, but not large, consumer of primary energy (<20%) in a system. The other is that typically only about 20% of the system fan-power requirement can be attributed to pressure drops across the desiccant wheel and heat exchanger combined. The bulk of the pressure drop comes from turbulent airflow through the cabinet and ducting. These figures assume a 1.0 mass flow ratio exists at some point in the system. This is the case for designs that employ a heat exchanger (i.e. most cooling applications where supply temperature must be minimized), so even for systems that exhaust a portion of the regeneration-side airflow prior to the regeneration heater; the wheels consume a fraction of the total fan power. Therefore, wheel pressure drop is more important a factor to seal performance than overall energy consumption.

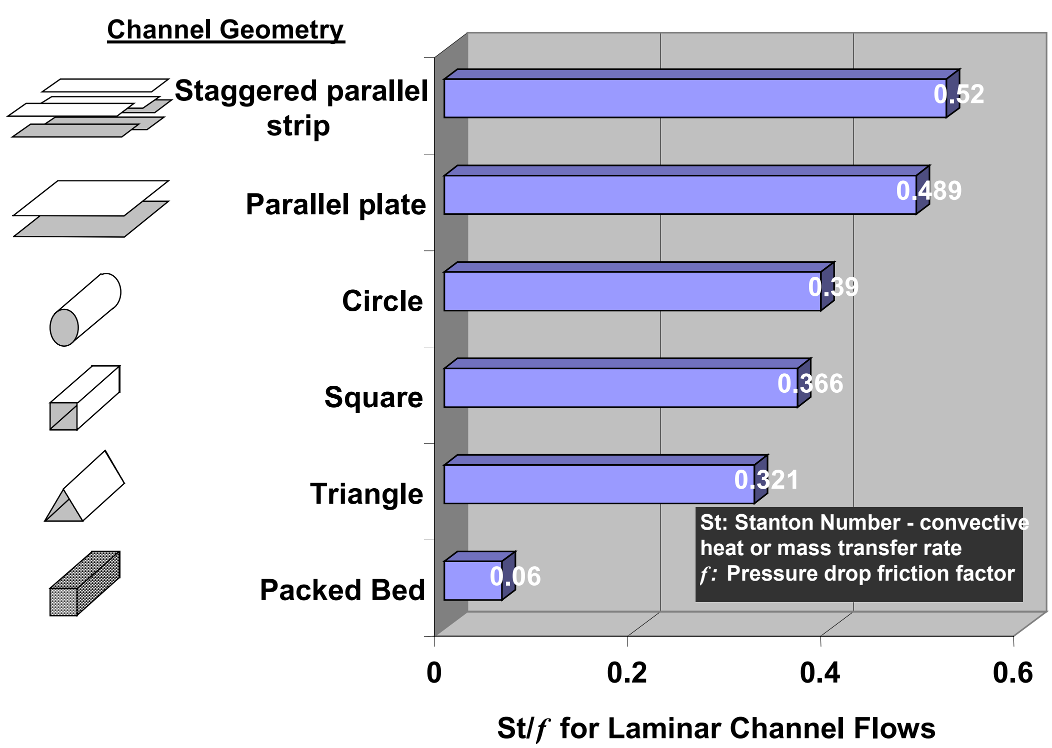

Figure 1. Analysis of mass-transfer-to-pressure-drop ratio (St/*f*) for various flute geometries.

Differential Pressure Measurement

Standard 139 calls for the measurement of pressure differential according to ASHRAE Standard 41.3–1989 across the regeneration and process sides of the wheel. It is equally important to measure the PI-RO and RI-PO face pressure differentials[6]. Maintaining reasonable face differentials is critical to successful testing. Subjecting the face seals to differentials larger than 2" w.c. will often lead to poor performance assessment of commercial products. Face differentials may have to be maintained even lower when testing prototypes. All four differentials need not be continuously monitored; so two separate sensors can do the job. Once reasonable face differentials have been established, these sensors can easily be switched over to monitor wheel pressure drops.

Leaks from inlets to outlets affect actual face velocities and contaminate outlet flows. In the field, fans are often arranged in blow/draw configuration to preserve grain depression in the supply air. Supply air is blown through the wheel, and regeneration air is drawn through. This prevents any regeneration air from forcing its way into the process side of the cassette, which can seriously degrade performance. In the laboratory, it helps to utilize four fans—one on each inlet and outlet[7]. In this way, face pressures can be varied to either minimize face differentials or simulate field conditions to test seal integrity. If four fans are not used, minimize face differentials by minimizing pressure drops on the wheel inlets/outlets opposite the fans.

Moisture Mass Balance and MRC

Leakage across face seals is a common condition that prevents moisture mass balance. The seals on commercial units typically will allow balance when face differentials are kept below 2" w.c. A balance of less than 1.0 usually indicates leakage from RI to PO, and degradation in MRC. The bone-dry PO air is very susceptible to small leaks of wet regeneration air. If the test system does not employ four fans, it may be necessary to induce a pressure drop on the PO ductwork to stop the leak[8].

Circumferential seals typically do not contribute to poor moisture mass balance on commercial wheels. If a cassette is sealed fairly airtight, any circumferential leak would have to bypass the wheel, passing through two circumferential seals. This effective "double-sealing" forces the path of least resistance to be through the wheel[9]. If the

[6] Section 6.15.7 of Standard 139 calls for the measurement of RI-PO differential, but it is not included on the sample data sheet or the system diagram. We recommend recording this value.

[7] The only concern about negative duct pressures in the lab is that leaks into the system can easily contaminate outlet airflows prior to measurement.

[8] To achieve the most accurate measurement of a wheel's performance, the pressure differential across the RI-PO face seal should be held at zero. This will minimize leakage between these two airstreams and the RI and PO flow rates will be measurements of the air actually passing through the wheel. In this case, the process outlet airflow should be used in the calculation of MRC (Eq. 1). This will force the moisture mass balance to be greater than 1; however, a well-sealed wheel will still produce a moisture mass balance within 5% of 1.

[9] This is true for typical, low-pressure-drop commercial wheels. A deep industrial wheel may have sufficient airflow resistance to force some air to bypass.

cassette is open to lab pressure, it is easier for air to escape through circumferential seals. This is particularly true if the wheel is not supported by an axle and is shipped on its face. The rotor is fairly heavy, and will tend to compress the circumferential seal it rests on. When the cassette is placed upright, a gap is formed if the seal cannot spring back sufficiently. Leaks such as these will lower the actual air mass flow through the wheel *after* it has been measured[10]. The resulting air velocity through the wheel will be lower than expected, enhancing grain depression across the wheel slightly[11]. If circumferential leaks leave the cassette, they will lead to a high bias in calculation of MRC as defined in Standard 139.

Cyclic Pressure Flux

Monitoring pressure differentials serves another important purpose. Wheel matrices are generally not perfectly uniform, in either open area or desiccant loading, and excess desiccant or compressed flutes will tend to restrict the air passages. This means airflow resistance varies with circumferential location. If the wheel has sufficient authority in the airflow circuit, its rotation will cycle the flow rates in synch with its frequency. It also means that performance can vary the same way. This is most noticeable in the regeneration airflow of 75/25 split wheels, where a non-uniformity in the matrix can occupy the greatest percentage of flow area. The amplitude of the cycle is not typically large enough to be detrimental to performance measurement, but fluctuating pressure can severely tax some duct-based psychrometric control schemes.

For example, steam injection was used to control humidity in the original design of our Advanced HVAC Test Facility. At low regeneration flow rates typical for small 75/25 wheels, humidity control was extremely difficult to maintain because injection rate depended both on injector valve position and duct-boiler pressure differential. It was difficult to modulate the valve adequately to compensate for both boiler pressure fluctuations and the cyclic variations in airflow and duct pressure caused by the rotating wheel. Our current humidifiers, evaporative saturators, are airside-limited devices and therefore provide very even humidification under such conditions.

This control issue should not affect the psychrometric chamber-based conditioning approach typically employed in the HVAC certification industry. Keeping wheel-non-uniformity in mind, however, can be a useful troubleshooting tool. When faced with unexpected results, measurements that do not follow the cyclic pattern can immediately be identified as suspect. One of the first steps in troubleshooting an experiment should be to check the frequency of a phenomenon to see if it coincides with wheel rotation.

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[10] Standard 139 section 9.2 calls for calculations based on inlet flow rates. Inlet air mass flow rates should be checked against the outlets. Mass flow rate agreement within 3% is an indication that circumferential leakage is not a problem. It is also useful to periodically "short" the inlet and outlet ducts as a check against each other.

[11] The relationship between face velocity and grain depression is not one-to-one—a 10% reduction in face velocity would not produce a 10% rise in grain depression. However, it could be an unacceptable few percent.

Flow Measurement

Standard 139 calls for the measurement of airflow rates according to ASHRAE Standard 41.2-1987 (RA 92).

Purge Sections/Carryover

As the matrix rotates out of the regeneration airflow, it carries with it both regeneration air trapped in the flutes and heat, contained in the air and in the matrix itself. This amounts to a small, constant "rotation leak" or carryover from RI to PO, which is acceptable in most instances. Purging purposely misaligns one of the seals on the RI/PO face of the wheel to eliminate this leak by forcing a "purge leak" from PI to RI. Figure 2 diagrams the purge concept. Purge sections are not addressed by either standard, but are commonly used in industrial applications when very low PO dew points are required. Purges can also be necessary in applications that demand minimal carryover of regeneration air into the supply air.

One of the reasons purge was not included in the test standards is that it would be very difficult to monitor in the lab. The purge section is extremely compact. Sampling is not likely to provide useful results. The entire purge flow would have to be extracted, measured, and reintroduced to the RI flow. This would require substantial modification to the cassette and seals and would certainly affect performance. Moisture mass balance could not be calculated without monitoring the purge flow in this way.

In low dew-point applications, the purge is designed to pre-cool the matrix before it begins to condition supply air. This is necessary because hot desiccant does not adsorb very well; without a purge, the first several degrees of rotation do very little dehumidification, allowing untreated air into the process outlet. This is in addition to carryover from regeneration air trapped in the flutes by wheel rotation. Purge is very effective at eliminating these performance inhibitors.

In the case of carryover, the purge prevents regeneration air trapped in the flutes from carrying contaminants into the supply air. This could be a concern if the unit is direct-fired and if there are combustion products in the regeneration air, or if the regeneration air comes from an indoor or outdoor source that may have high levels of volatile organic compounds (VOCs) or other pollutants. For example, if building exhaust air is used for regeneration, and the interior is emitting high levels of VOCs (e.g. new construction), RI to PO carryover reduces the effectiveness of the ventilation air for maintaining indoor air quality. Generally speaking, seal leakage and rotation carryover combined are not large enough to be a concern.

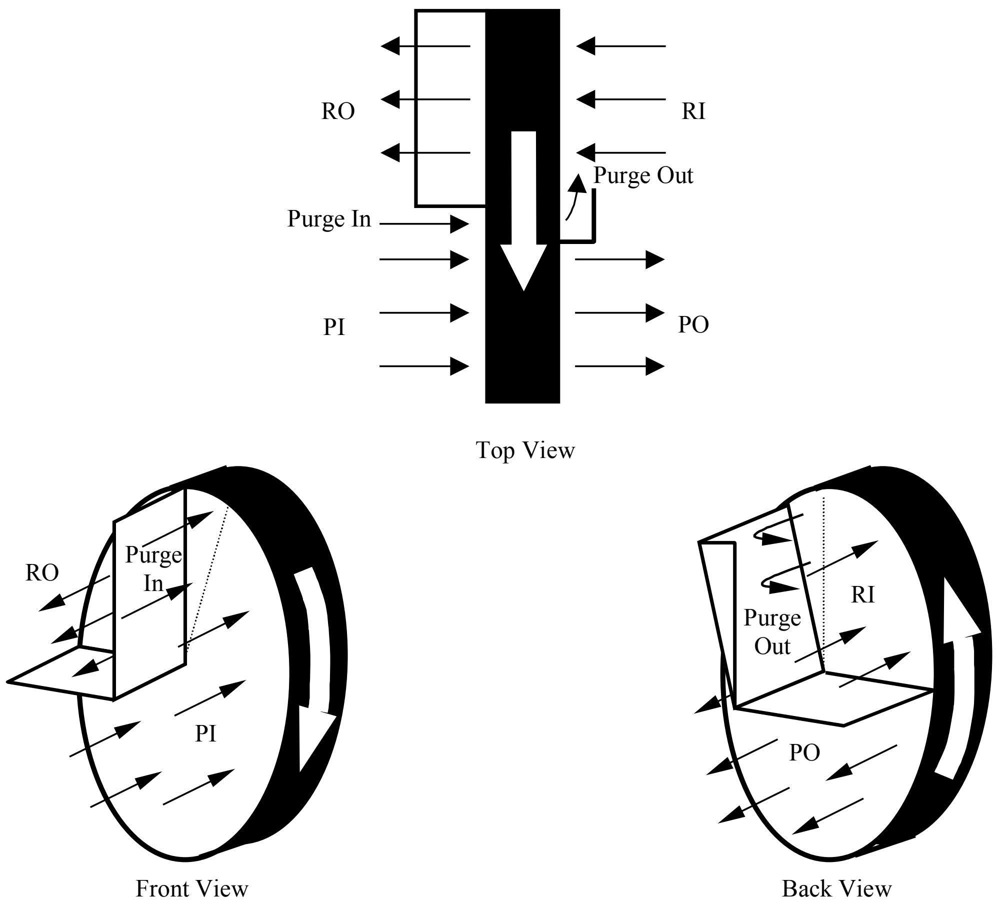

Figure 2. Purge section forces a PI to RI "leak" to prevent RI to PO carryover contamination inherent in wheel rotation.

Another potential concern is co-sorption. Co-sorption is the potential for desiccants to adsorb other chemicals with the water vapor. If the desiccant were able to pick up considerable amounts of undesirable chemicals from an exhaust flow and dump them back into the supply air, this would create a much more powerful carryover effect than wheel rotation could produce, and essentially concentrate the pollutants in the building. This is very unlikely for several reasons. First of all, in actively regenerated systems, the pollutants would have to be picked up by the desiccant at elevated temperatures and released at low temperature, the opposite sense in which sorbents work. In passive systems, this reasoning does not apply because the regeneration air is not heated. There are two lines of reasoning for these enthalpy exchange systems. One is size exclusion. Pollutant molecules larger than the desiccant pores are physically excluded from adsorption, making carryover impossible. Three angstroms is sometimes cited as a practical pore size in which water vapor fits, but many pollutants cannot. The other is that co-sorption does not happen in the presence of water vapor. Sorption on the molecular level is a very electrically influenced phenomenon. Water vapor is a highly polar molecule; that is, it has strongly positive and negative ends. Analyses predict that desiccants will always adsorb the most polar molecules first. Experience shows this to be true. Even ammonia, which is moderately polar, is not picked up in appreciable quantities when water vapor is present. Carryover does not currently appear to be an issue for rotary desiccant equipment, but it should be kept in mind for each new application.

Air Mass Balance

Outlet and inlet nozzles should be checked against each other as a quality-of-test figure of merit in parallel with moisture mass balance. This air mass balance can take a couple of forms and is useful when troubleshooting. In one form, inlet flows should be summed and compared to the sum of the outlet flows. This balance will remain between 1.00 and 1.02 when there are no substantial leaks out of the system. In another form, the mass flows of inlet/outlet pairs should be compared to each other. They should match within the 3% experimental uncertainty called for in the standard. Each level of air mass balance can be a clue to narrowing down a problem with the device under test or the test rig itself.

Airflow Uniformity/Blowthrough

Another important consideration in testing rotary equipment related to airflow is uniformity. The desiccant wheel typically has a relatively high pressure drop (~1"w.c.). This is convenient for testing because it helps even out the airflow distribution upstream of the wheel. Improper ducting, however, can overcome this feature and present a very non-uniform air distribution that will degrade performance. Introducing inlet air too close to the wheel or at an odd angle through too small a duct can cause this "blowthrough." It starves some portions of the wheel, and raises flute velocities in others for a net negative effect on performance. Transitions and/or flow conditioning baffles are in order to ensure reasonable uniformity within several percent.

Introducing air at an odd angle is of particular concern in testing desiccant wheels. Flexible ducting is often required to connect the test rig to the wide range of available equipment sizes. This required flexibility leaves the possibility that ductwork ends up at non-ideal angles that can contribute to blowthrough. Introducing air in this way can also severely affect standard pressure taps. Conventional design relies on parallel flow along the duct axis. Impinging flow on the pressure tap will naturally ruin the measurement. Take note of the pressure drops across the wheel from test to test and compare inlet/outlet flows to guard against this. Move flexible ducts around during steady state testing to be sure their positions don't affect results.

Dry-Bulb Temperature Measurement

Standard 139 calls for the measurement of temperatures according to ASHRAE Standard 41.1-1986 (RA 91). As an additional reference, the general insights on thermocouple use provided by Moffat (1962)[12] are especially valuable.

Mixing/Sampling

Rotary heat/mass transfer devices produce very spatially non-uniform air temperature distributions (see Figure 3)[13]. Mixing prior to measuring or sampling is critical to accurate testing. Standard sampling trees are very effective in helping obtain representative averages, but cannot be relied on. Sufficient mixing can be achieved by flow conditioning or development length. Standard baffles, screens, or mixing vanes accelerate thermal mixing, as shown in Figure 4, and can help shield sensors or sampling trees from radiative heat exchange with the rotor. A drawback of these devices is that the pressure drop they add is not adjustable, and may adversely affect face pressure differentials at times. Infrared imaging shows that ten hydraulic diameters of flow development provide excellent uniformity (see Figure 5).

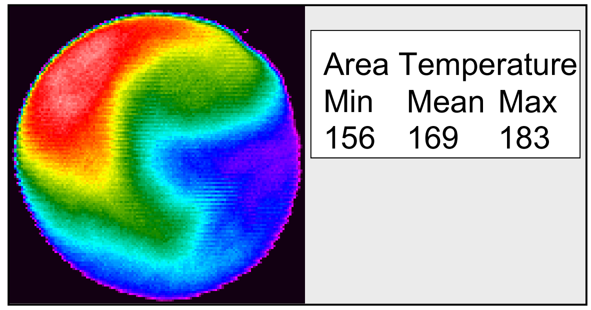

Figure 3. IR image of regeneration outlet air with no mixer.

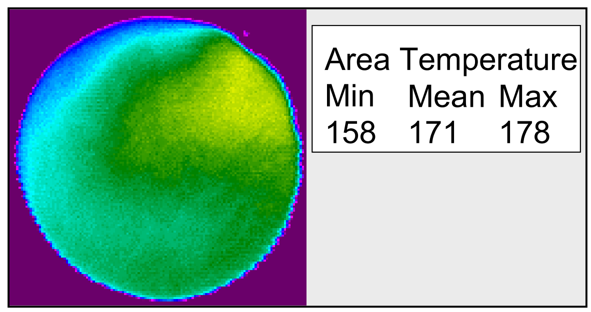

Figure 4. IR image of regeneration outlet air with mixer.

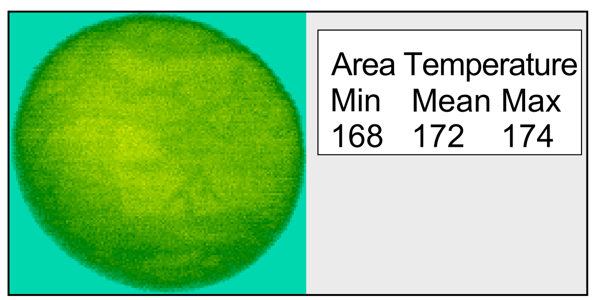

Figure 5. IR image of regeneration outlet air 10 duct diameters from plenum.

The drawback of using development length alone is the potential to lose heat before the outlet flows can be measured. Average process and regeneration outlet temperatures of active desiccant dehumidifier wheels range from 110°F to 180°F—not particularly high relative to room temperature. Simple heat transfer calculations should be applied to determine

[12] Moffat, R.J., "The Gradient Approach to Thermocouple Circuitry," Proceedings of 4th Symposium on Temperature, Its Measurement and Control in Science and Industry, v.3, Reinhold, New York, 1962.
[13] These infrared images illustrate the effect of different mixing techniques on temperature uniformity after air has exited the wheel into a plenum and then entered a 12-inch round duct. A screen, secured over the end of the duct, served as a target for the infrared camera. The screen itself imposes a pressure drop that tends to mix the flow, so actual spatial variation in the duct is probably slightly greater than that depicted here.

the level of duct insulation required to eliminate this concern. Apply a safety factor in the calculations to account for hot spots in the unmixed flow. Both mixing and development length approaches should be applied for an optimized solution.

Near-Rotor Measurements

Near-rotor measurements are unlikely to provide reliable average outlet temperatures; however, there are times when measuring air properties close to the rotor face is useful. For example, near-rotor measurements at the wheel inlet face are necessary to assess inlet air temperature uniformity. Radiative heat exchange between the wheel face and sensors is the critical concern that needs to be addressed. Radiation shields that allow ample aspiration for the sensor are required. Figure 6 shows a solution for inlet airflow that has been successfully deployed in the lab. Shielded thermocouple grids show that inlet temperature uniformity within 1.3°F is possible even for 40-inch diameter wheels. Without shielding, the same temperature distribution would appear to vary over 15°F. This solution will not work for outlet flows. One possible configuration for near-rotor outlet air measurements is described in Figure 7.

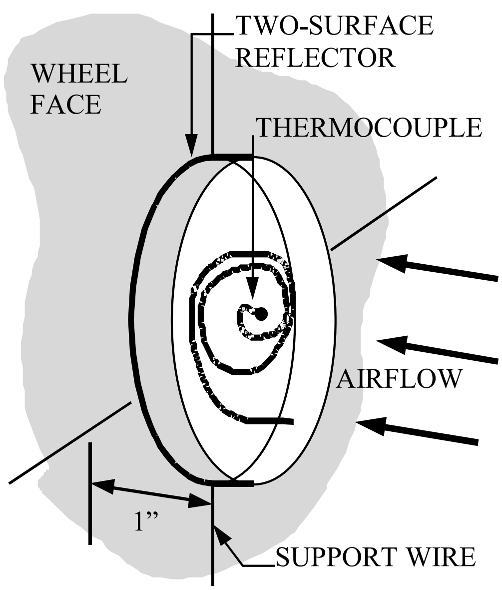

Figure 6. Radiation shield for near-rotor inlet air temperature

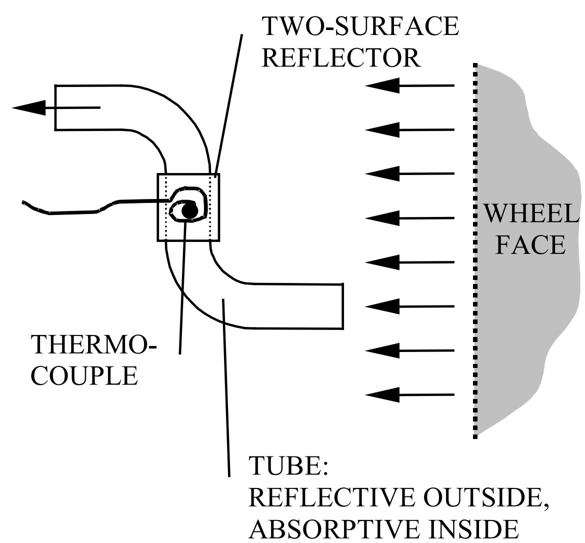

Figure 7. Radiation shield for near-rotor inlet or outlet air temperature

The following rules-of-thumb apply to tube-type radiation shield design:

- Shield length (L) should be at least eight times its internal diameter
- Air gap between shields should be larger than 20L/(RePr)
- All surfaces should have low emissivity (not white), except the innermost surface, which is black and the outermost surface, which is white
- Type-T thermocouple wire length should be at least 50 wire diameters
- If possible, aspirate the shielding to show that the reading doesn’t change.

Humidity Measurement

Standard 139 calls for the measurement of air wet-bulb temperature according to ASHRAE Standard 41.1-1986 and measurement of air dew-point temperature according to ASHRAE Standard 41.6-1994. Humidity measurement is a critical parameter in the testing of desiccant dehumidification rotors. Although the absolute humidity ratio is what is used to calculate figures of merit, typical humidity sensors measure an air property other than humidity ratio: wet-bulb temperature, dew-point temperature, or relative humidity. Wet-bulb and relative humidity methods require the additional measurement of total pressure and dry-bulb temperature to calculate humidity ratio. The dew-point method determines humidity ratio with the additional knowledge of total pressure only. Recent advances in relative humidity sensors have increased their accuracy dramatically (±3% rh → ±1% rh). Due to the unique nature of a desiccant dehumidification rotor, the moist air properties of the air streams leaving one of these devices will be quite different from other HVAC equipment. The process outlet air will typically be single-digit relative humidity while the regeneration outlet will be hotter and more humid than naturally occurring, terrestrial environments. Because of this, careful selection of appropriate humidity sensors is required.

Mixing/Sampling

The infrared images in the previous section (Dry-Bulb Temperature Measurement) also apply to humidity measurements. The air leaving an actively regenerated desiccant rotor is very non-uniform in humidity. It is postulated by Reynolds Analogy that once the air is thermally uniform, moisture uniformity is also achieved. As such, it is recommended that a combination of mixers and a minimum of five duct lengths be used to achieve well-mixed air. A sampling tree should then be used to sample air from the cross section of the duct.

Wet-Bulb Method

The most common humidity measurement method in the HVAC laboratory is the aspirated psychrometer. This device is simple and inexpensive yet can be used to make relatively accurate humidity measurements by the trained user.

The following practices apply to making accurate humidity measurements using the wet-bulb method[14]:

- Use a sampling tree at a point of well-mixed air.
- Avoid a dirty or contaminated wick (wicks should not be handled without gloves and should be changed on a regular basis).
- The water in the reservoir is distilled and within 3°C of the wet-bulb temperature of the air.

[14] Taken from "Psychrometrics - Theory and Practice" and "ASTM Standard E 337: Standard Test Method for Measuring Humidity with a Psychrometer (the Measurement of Wet- and Dry-Bulb Temperatures)."

- An instrument-quality wick is used.
- The wick is of an appropriate diameter to assure a snug fit around the temperature probe and extends at least 1” above and below the tip of the probe.
- Air flow across the sensors is approximately 1000 fpm.
- The effects of thermal radiation and stem conduction are considered.
- Avoid wet-bulb depressions greater than 15°C and relative humidity < 10%.
- Calibrate the unit annually or as recommended by the manufacturer.

Three measurements are required to calculate the humidity ratio using the wet-bulb method: dry-bulb temperature, wet-bulb temperature, and ambient pressure. The following equation out of the ASHRAE Handbook of Fundamentals (1997)[15] is recommended for calculating humidity ratio using the wet-bulb approach:

$$w = \frac{\left(K_1 - K_2 t^*\right) w_s^* - \left(t - t^*\right)}{K_1 + K_3 t - K_4 t^*} \tag{18}$$

where K_1 - K_4 are found implicitly in the Handbook of Fundamentals and w_s^* is the humidity ratio at the thermodynamic wet-bulb temperature, which is approximated by using the wet-bulb temperature in its place. So,

$$w_s^* = \frac{0.622 \cdot p_{vs}^*}{p - p_{vs}^*} \tag{19}$$

where p_{vs}^* is the saturation pressure of water evaluated at the wet-bulb temperature:

$$p_{vs}(T) = \exp\begin{pmatrix} \dfrac{C_8}{T} + C_9 + C_{10}T + C_{11}T^2 \ldots \\ \ldots + C_{12}T^3 + C_{13}\ln(T) \end{pmatrix} \tag{20}$$

where the coefficients C_8 - C_{13} are found in the ASHRAE Handbook of Fundamentals.

Dew-Point Method

A chilled mirror hygrometer[16] is used to accurately measure the dew point of an airstream. Although this is a sophisticated and relatively expensive instrument, the high accuracy and increased reliability have made its use in the laboratory and some field applications more common. The primary advantage to a dew-point hygrometer is its ability to measure low relative humidity air while maintaining a high degree of accuracy. Like the aspirated psychrometer, a chilled mirror hygrometer suffers from contamination. The surface of the mirror must be cleaned periodically to remove contaminants. Unlike the other humidity measurement sensors, the chilled mirror hygrometer uses a control

[15] ASHRAE Handbook: Fundamentals, Chapter 6: Psychrometrics, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, GA, 1997.

[16] This device operates by having a sample of air drawn over a small mirror that is chilled by a thermoelectric heat pump. Once condensation is optically sensed on the surface of the mirror, the temperature of the mirror is maintained and measured with a platinum resistance thermometer. This process is continuously monitored to maintain a constant mass of water on the surface of the mirror.

loop to maintain accurate measurements. At times the instrument will "get lost" and search for its equilibrium point. Depending on the nature of the event, the hygrometer may not be able to get back in control on its own and will have to be reset manually. Some units allow this to be done remotely. A very small air sample is needed (1–5 ft^3/hr) for the modern chilled mirror hygrometer. The sample lines should be kept as short as possible, and they must be heated to prevent condensation from forming in them. The elevated dry-bulb temperature of the air sample does not effect the humidity measurement so long as the thermoelectric heat pump can provide sufficient temperature depression of the chilled mirror. A two-stage cooler will provide 65°C of sensor temperature depression. Some dew-point sensors do not have the cooling capacity to measure very low dew points, and the cooling rate will affect response times. Check manufacturer's specifications to match sensors to the task.

The following practices apply to making accurate humidity measurements using the dew point method:

- Use a sampling tree at a location of well-mixed air.
- Periodically clean the chilled mirror surface as recommended by the manufacturer—more frequently is not necessarily better and may be detrimental.
- Periodically zero the instrument to account for trace amounts of contaminants.
- Locate the sensor close to the sampling tree to minimize the length of sampling tube.
- Heat the sampling tube to prevent condensation from occurring.
- Make sure the thermoelectric heat pump has sufficient capacity for the air stream being measured.
- Calibrate the unit annually or as recommended by the manufacturer.

Two measurements are required to calculate the humidity ratio using the dew point method: dew point temperature and duct static pressure. The following equation out of the ASHRAE Handbook of Fundamentals is recommended for calculating humidity ratio using the dew-point method:

$$w = \frac{0.622 \cdot p_{vs}}{p - p_{vs}} \tag{21}$$

where p_{vs} is the saturation pressure evaluated at the dew-point temperature (Eq. 20).

Relative Humidity Method

In the past, relative humidity sensors have been used to monitor the moisture level of the air in a building. An accuracy of ±3% relative humidity was sufficient for this monitoring, but was insufficient for measuring the performance of HVAC equipment. However, recent advances have increased the best available accuracy of these sensors to ±1% relative humidity. This enables their use in monitoring the performance of HVAC equipment without incurring high uncertainties, while providing low maintenance and reliable performance.

Typically, these sensors use a material whose capacitance varies with the relative humidity of the airstream in which they are exposed. The humidity sensor is usually coupled with a temperature sensor within a filtered cavity. The output from this temperature sensor should be used for all humidity calculations. The velocity of the air passing over the sensors should be monitored and kept within the manufacturers recommended range. This will prevent slow response times and decrease the possibility of conduction and radiation errors. It is not uncommon to insert a relative humidity sensor directly in the duct. If this is done, the thermal and moisture uniformity at that location is paramount. The flow uniformity should also be verified to assure oneself that sufficient flow over the sensor is provided.

The following practices apply to making accurate humidity measurements using the relative humidity method:

- If a sampling method is used, use a sampling tree at a location of well-mixed air.
- If the sensor is inserted in a duct, do so at a location of very well mixed air.
- Monitor the airflow across the sensor.
- Use the temperature output from the temperature sensor provided with the unit.
- Maintain strict control of duct air temperatures within the sensor's safety range to avoid damaging the sensing element.
- Calibrate the unit annually or as recommended by the manufacturer.

Three measurements are required to calculate humidity ratio using the relative humidity method: dry-bulb temperature, relative humidity, and ambient pressure. The following equations from the ASHRAE Handbook of Fundamentals are recommended for calculating the humidity ratio using the relative humidity method:

$$w = \frac{0.622 \cdot p_v}{p - p_v} \tag{22}$$

where

$$p_v = \Phi \cdot p_{vs} \tag{23}$$

where p_{vs} is Eq. 20 evaluated at the dry-bulb temperature, and Φ is the decimal representation of relative humidity.

Total Combined Uncertainty

Standard 139 calls for specific limits on uncertainty for instrumentation, but does not discuss total combined uncertainty for its primary figures of merit, MRC and RSHI. Its requirement that moisture mass balance fall within 5% of 1.0 ***must not be taken as the accuracy of these calculated results***. There are several ways to mathematically propagate random and bias uncertainties into a total combined uncertainty for a given figure of merit.

Calculation of what is commonly called *true* uncertainty involves a "what if" exercise to determine a worst-case scenario in the calculations. It assumes all measurements are in error to the maximum extent possible, and each in a sense that skews the calculated results in the same direction. For example, if experimental technique is perfect, measured dry-bulb temperature is 0.3°C *high*, and measured wet-bulb temperature is 0.3°C *low*, calculated absolute humidity will be low by the maximum amount possible using these instruments. With this approach, and standard instrumentation, it is easy to realize MRC uncertainties in excess of 25%. Thankfully, in the absence of extremely biased errors, it is statistically very unlikely that this condition will exist. It is much more likely that random errors will partially compensate for each other. This is the approach detailed in Kline and McClintock (1953)[17], and the one recommended and used here.

Uncertainty in a test result has two components: random uncertainty and systematic (bias) uncertainty. Uncertainty analysis should help determine which instruments will play a significant role in the magnitude of the uncertainty and which will not. This information should then be used to focus more resources in those instruments playing a major role.

Sources of systematic uncertainties that will be an issue in testing an actively regenerated desiccant rotor have been discussed in the previous sections of this test guide and are summarized here. They include (but are not limited to):

Pressure/Flow

- Maldistribution of air supplied to the rotor (blowthrough)
- Air leaks between air measurement stations
- Use of instrumentation outside of published range
- Use of instrumentation out of calibration
- Not allowing appropriate development lengths upstream or downstream of nozzles
- Poor nozzle construction
- Poor pressure tap construction/location.

Temperature/Humidity

- Sampling of a non-uniform air stream
- Conduction and/or radiation affecting dry-bulb and/or wet-bulb measurements
- Use of instrumentation outside of published range
- Use of instrumentation out of calibration
- Allowing condensation to form in sampling tubes
- Insufficiently insulated ducts or sampling tubes
- Contaminated wicks for wet-bulb measurements
- Contaminated mirror for dew-point sensors
- Insufficient air flow across a sensor
- Requiring a dew point sensor or wet-bulb sensor to develop a temperature depression greater than their capability.

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[17] Kline, S.J., and F.A. McClintock, "Describing Uncertainties in Single-Sample Experiments," Mechanical Engineering, Vol.75, No.1, pp. 3-8, 1953

Systematic errors, if not sufficiently addressed, can overwhelm random errors. With so many different possibilities, quantifying the effect of systematic errors on a test result is difficult, and varies from lab to lab and test to test. Good testing procedures will minimize their effect, but not eliminate it. As researchers and test engineers, it is important that we maintain an awareness of their existence and work to minimize their effect.

Instrument readings contain both random and bias errors. The following section illustrates the propagation of instrument uncertainty into test results assuming that the manufacturers' stated instrument accuracies are entirely random and that other non-instrument systematic errors are negligible. Under some conditions, the effect of including bias components would be that total uncertainty would be slightly more than that presented here, and, under other conditions, it would be slightly less. One could argue that sensors calibrated to each other could substantially reduce uncertainty in a differential measurement (like grain depression across a wheel). But non-instrument systematic uncertainties cannot be totally eliminated, and so our approximation gives a sense of what is reasonable and achievable based on our experience. The intent of this discussion is to show that even under ideal testing conditions, all of the humidity instruments examined here have distinct limitations. All laboratories conducting desiccant wheel testing should complete detailed uncertainty analyses including the effects of their specific instruments' bias errors and quantify their rigs' systematic biases. References for conducting detailed uncertainty analyses include Coleman and Stuck (1999)[18] and Dieck (1992)[19].

Instrument Uncertainty Propagated into Humidity Ratio

To calculate a figure of merit, the humidity measurements must be converted into a humidity ratio. In the previous section, three methods of humidity measurement were discussed: wet-bulb method, dew-point method, and relative humidity method. Multiple measurements are required to calculate the humidity ratio for each of these methods. This section will illustrate how the uncertainty in each of these individual measurements propagates into the calculation of the humidity ratio.

The root-sum-square method of uncertainty calculation is applied here to the individual equations used in calculating the humidity ratio for each individual approach. If the instrument uncertainties are independent, it is statistically likely that the errors will partially counteract each other most of the time such that the square root of the sum of the squares of the individual uncertainties is a more representative gauge of the overall random uncertainty. If w is a function of three independent variables (x,y,z), the random uncertainty in w (δw) is:

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[18] Coleman, H.W., and W.G. Stuck, *Experimentation and Uncertainty Analysis for Engineers*, Wiley, New York, 2nd edition, 1999.

[19] Dieck, R.H., *Measurement Uncertainty Methods and Applications*, Instrument Society of America, North Carolina, 1992.

$$\delta w = \left[\left(\frac{\partial w}{\partial x} \delta x \right)^2 + \left(\frac{\partial w}{\partial y} \delta y \right)^2 + \left(\frac{\partial w}{\partial z} \delta z \right)^2 \right]^{1/2} \tag{24}$$

where $\frac{\partial w}{\partial x}$ is the partial derivative of w with respect to x and δx is the uncertainty in x, and so on. The partial derivatives can be interpreted as sensitivity coefficients of the humidity ratio. The magnitude of each sensitivity coefficient enables one to determine which measurements play a significant role in the uncertainty in w. Slayzak and Ryan (1998)[20] give a thorough description of this uncertainty analysis applied to the three humidity measurement methods described above. This uncertainty analysis is now applied to the four ARI rating conditions given in Table 2.

Table 2. ARI Rating Conditions

ARI Condition Number	Process Inlet Condition (T_{DB}/T_{WB}/T_{DP}/RH)	Regeneration Heater Inlet Condition (T_{DB}/T_{WB}/T_{DP}/RH)
1	95°F/75°F/67°F/40%	95°F /75°F/67°F/40%
2	80°F /75°F/73°F/80%	80°F /75°F/73°F/80%
3	80°F /67°F/60°F/51%	95°F /75°F/67°F/40%
4	45°F /45°F/45°F/100%	80°F /75°F/73°F/80%

For this analysis, it will be assumed that the humidity measurement of the regeneration air occurs before it enters the regeneration heater. This is the only option for both the wet-bulb method and the relative humidity method because of temperature limits. The analysis is applied to the two levels of instrument accuracies given in Table 3.

Table 3. Instrument Accuracies Used in Uncertainty Analysis

	Temperatures (T_{DB}, T_{WB}, T_{DP})	Relative Humidity[21]	Mass Flow Rate	Pressure
Standard 139 Accuracies	±0.3°C	±3% rh	±3%	±0.13 kPa
High Accuracy	±0.15°C	±1% rh (0 - 90% rh) ±2% rh (90 - 100% rh)	±1%	±0.13 kPa

A series of figures follow giving the results of this uncertainty analysis. The results are reported as a percent uncertainty, which is found by dividing the uncertainty in the calculated value by the calculated value:

[20] Slayzak, S.J., and J.P. Ryan, "Instrument Uncertainty Effect on Calculation of Absolute Humidity Using Dew-Point, Wet-Bulb, and Relative Humidity Sensors." Solar '98: ASME International Solar Energy Conference Proceedings, 1998

[21] ASHRAE Standard 139 does not consider the use of relative humidity sensors for humidity measurements; however, representative models are included here for completeness.

$$\% \text{ uncertainty} = \frac{\delta w}{w} \tag{25}$$

Figure 8 gives the uncertainty in the calculation of the process inlet humidity ratio for each of the four ARI process inlet rating conditions using the Standard 139 accuracies given in Table 3. It can be seen that the uncertainty is least for the dew-point method. The uncertainty is greatest for the relative humidity method for the first three points. The high relative humidity of the fourth point enables the relative humidity method to incur less uncertainty than the wet-bulb method. Figure 9 gives the uncertainty results for the regeneration heater inlet conditions. Looking at Table 2, points (1,3) and (2,4) are identical pairs. Again, the dew-point method incurs the least uncertainty and the relative humidity method the greatest.

The outlet air conditions from an actively regenerated desiccant rotor will vary depending on the inlet conditions, airflow rates, and the rotor's performance. The first assumption made here is that the rotor removes half the moisture from the process inlet air (e.g. ARI condition #1 has an inlet humidity ratio of 98.6 grains/lb; therefore, the process outlet humidity ratio is assumed to be 49.3 grains/lb)[22]. Due to the heat of adsorption and dump-back from the regeneration air stream, the process outlet air will be hot and dry: not ideal for the wet-bulb and relative humidity methods. Three possible process outlet conditions will be illustrated in this analysis: isothermal, adiabatic, and adsorption heat ratio (AHR) of 0.7. AHR is the ratio of sensible heat gain in the process air due to adsorption to the actual sensible heat gain:

$$AHR = \frac{T_{PO,adiabatic} - T_{PI}}{T_{PO} - T_{PI}} \tag{26}$$

[22] Experience shows this assumption gives a reasonable figure for the outlet conditions.

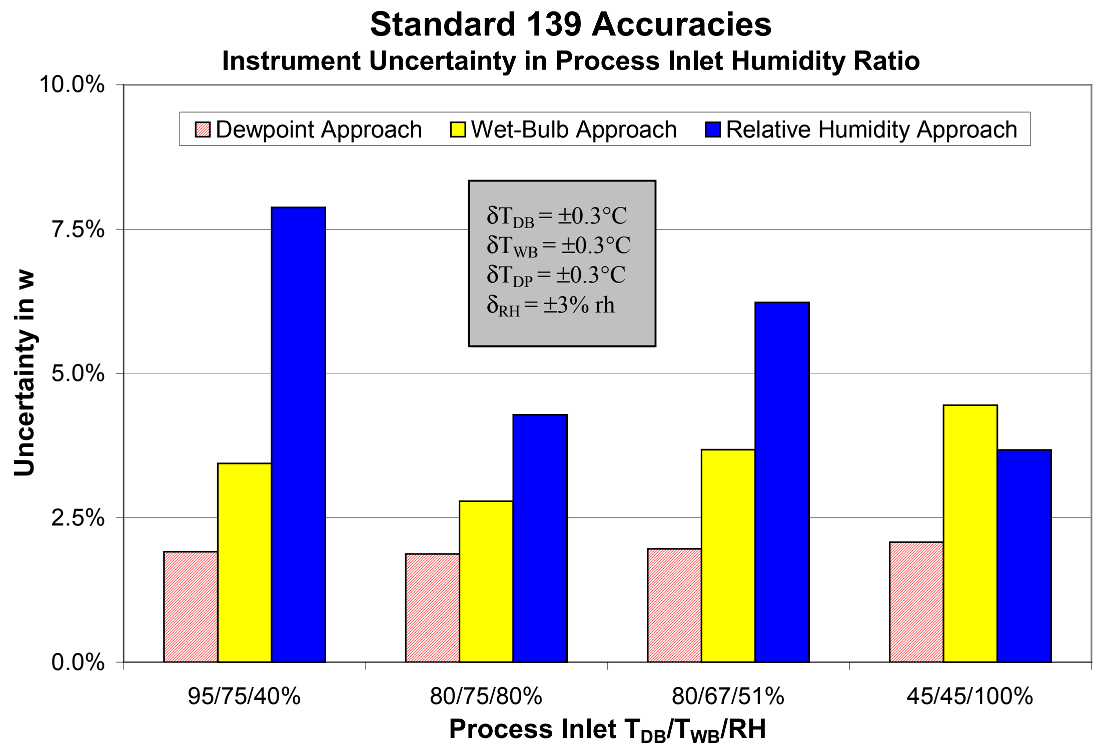

Figure 8. Instrument uncertainty in calculation of process inlet humidity ratio (Standard 139 accuracies).

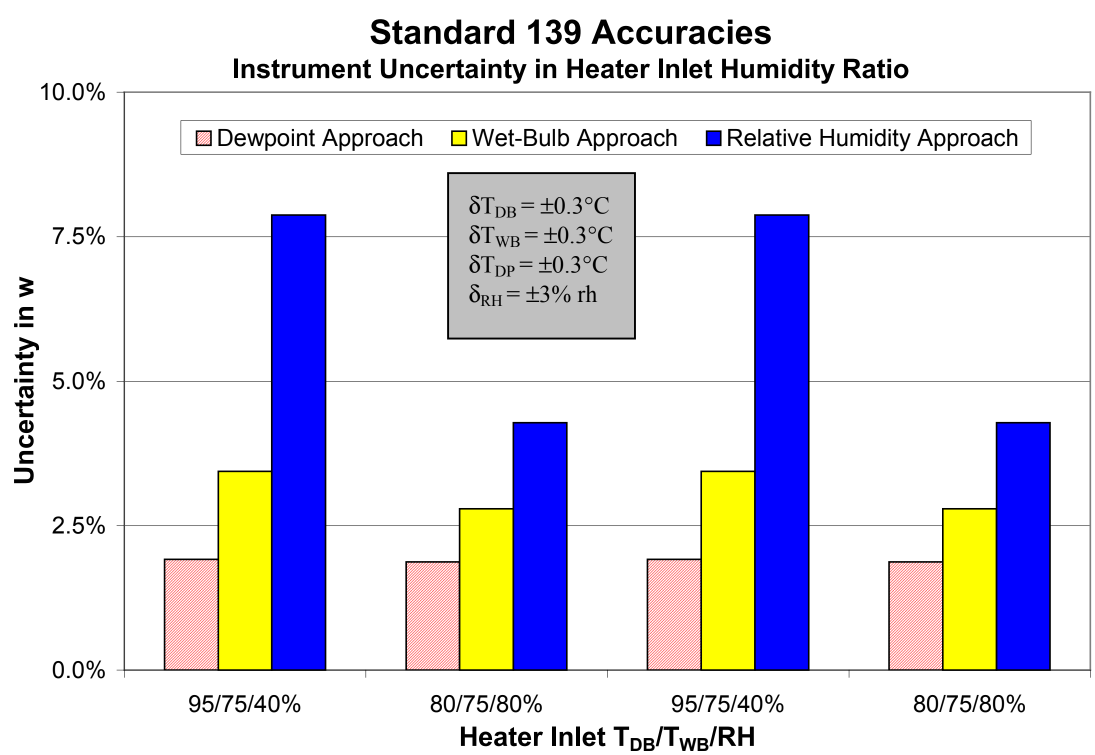

Figure 9. Instrument uncertainty in calculation of heater inlet humidity ratio (Standard 139 Accuracies).

These three outlet conditions are illustrated in Figure 10. Clearly, the isothermal process will have the highest relative humidity, and it is the only suitable process for the wet-bulb and relative humidity methods due to the low relative humidities of the other processes.

Figure 11 gives the instrument uncertainty in calculating the process outlet humidity ratio assuming an isothermal process takes place. This process assumes that the sampled air is cooled prior to measurement to provide conditions that are more favorable for the sensor. Figure 12 gives the instrument uncertainty in calculating the process outlet humidity ratio assuming an adiabatic process (AHR = 1.0) takes place. This process assumes no dump-back occurs and will result in the highest relative humidity that can be achieved given the inlet conditions and the moisture removal. A more representative outlet condition is with AHR = 0.7 (Fig. 13) resulting in a lower relative humidity at the process outlet due to dump-back.

For this analysis, the regeneration outlet conditions are determined by setting the moisture mass balance to 1.0 and solving for $w_{RO;}$ and setting the change in enthalpy on the process side equal to the change in enthalpy on the regeneration side and solving for T_{RO}. This analysis will assume AHR = 0.7 to calculate the regeneration outlet conditions. Figure 14 gives the instrument uncertainty in calculating the regeneration outlet humidity ratio. Because of the low relative humidities for conditions 1, 3 and 4, the relative humidity method incurs large uncertainties. It should also be noted that the wet-bulb temperature depression for these three conditions exceeds the maximum recommended depression for accurate wet-bulb measurements.

A similar analysis can be performed to determine the effect of instrument uncertainty with the use of high accuracy instruments. This analysis is not reported here.

Instrument Uncertainty Propagated into MRC

The moisture removal capacity is the figure of merit reported under the ARI Certification Program. MRC is defined as:

$$MRC = \dot{m}\left(w_{PI} - w_{PO}\right) \tag{27}$$

The root sum square method of uncertainty applied to Eq.27 gives:

$$\delta MRC = \left[\left(\frac{\partial MRC}{\partial \dot{m}}\delta\dot{m}\right)^2 + \left(\frac{\partial MRC}{\partial \Delta w}\delta\Delta w\right)^2\right]^{1/2} \tag{28}$$

Using a similar approach, the instrument uncertainties in [w_{PI} - w_{PO}], or Δw, are illustrated in Figure 15 for the Standard 139 accuracies. Because the isothermal process, described in the previous section, is the only possibility for accurate measurements with the wet-bulb and relative humidity methods, this is the process evaluated here. This analysis assumes a heat exchanger is being used to remove the sensible heat gain across the rotor prior to the process outlet humidity measurement.

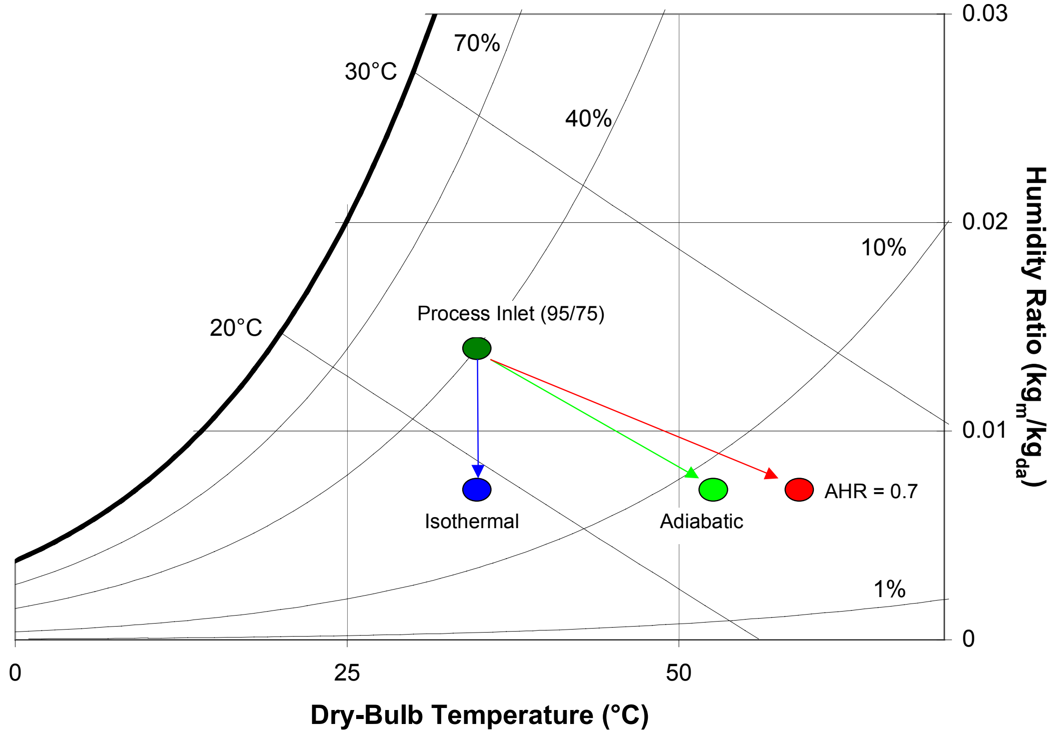

Figure 10. Illustration of the three process air-outlet conditions used in the uncertainty analysis.

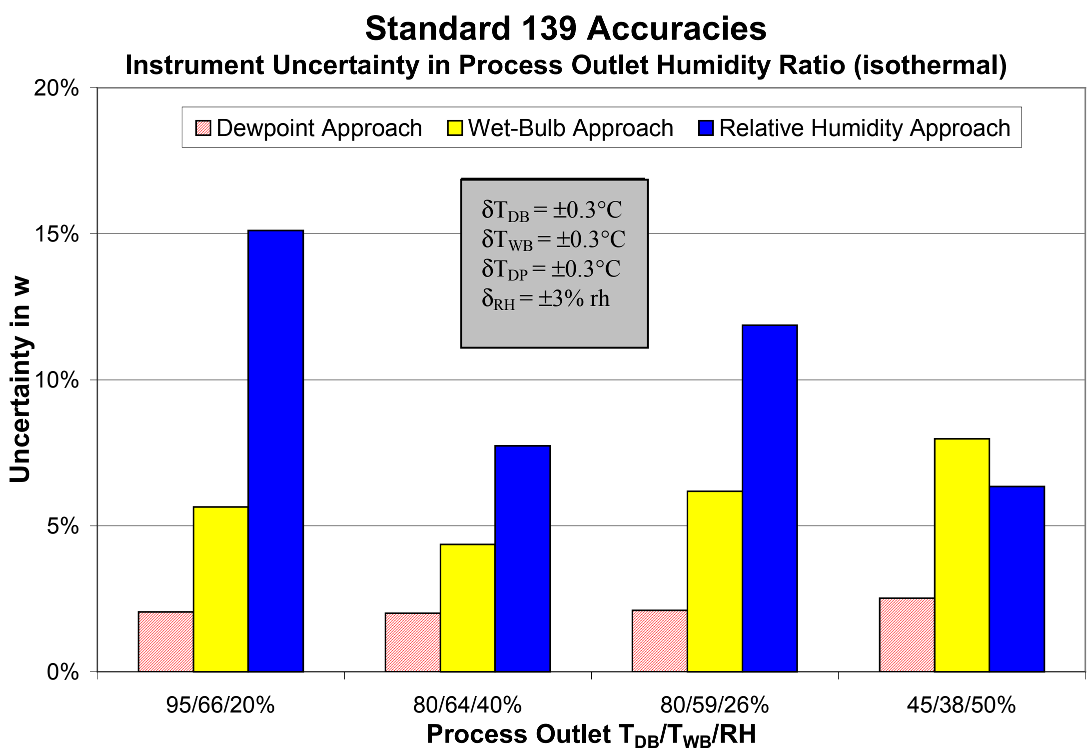

Figure 11. Instrument uncertainty calculation of process outlet humidity ratio assuming an isothermal process (Standard 139 Accuracies).

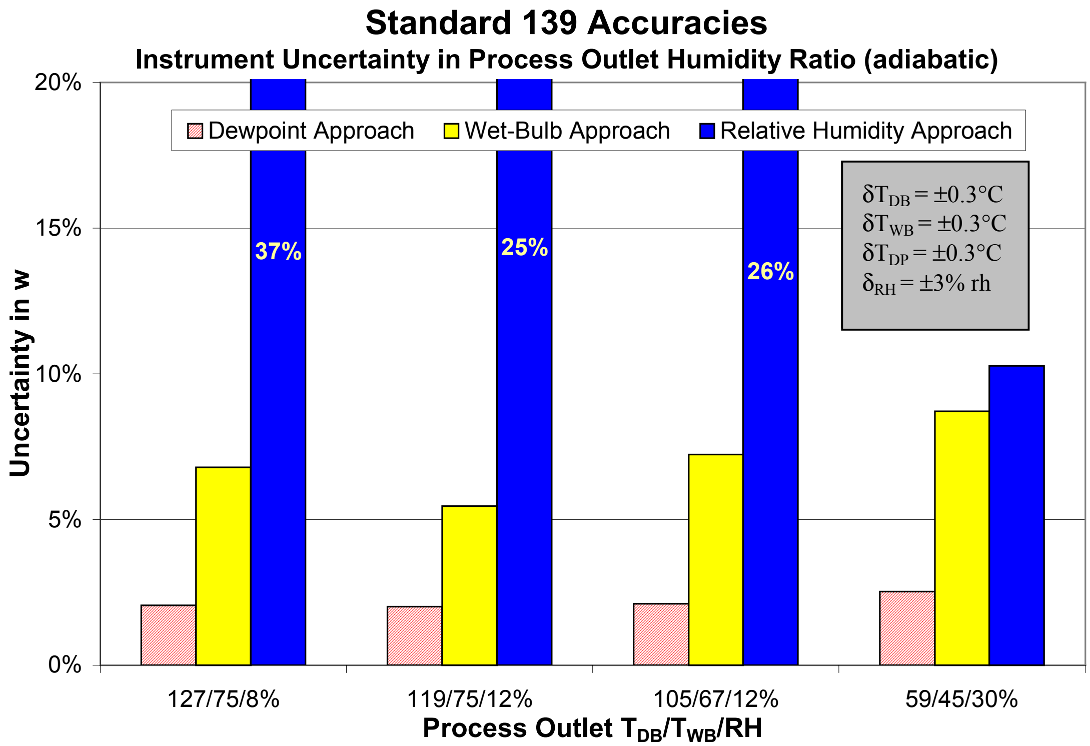

Figure 12. Instrument uncertainty in calculation of process outlet humidity ratio assuming an adiabatic process (Standard 139 Accuracies).

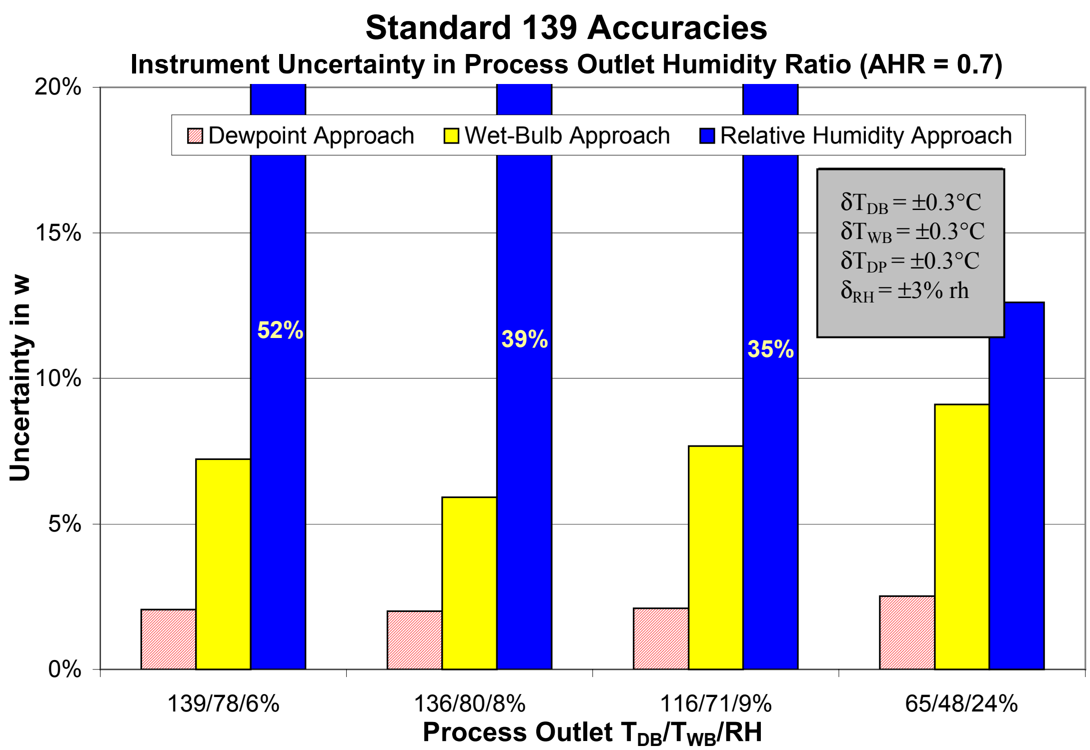

Figure 13. Instrument uncertainty in calculation of process outlet humidity ratio assuming AHR = 0.7 (Standard 139 Accuracies).

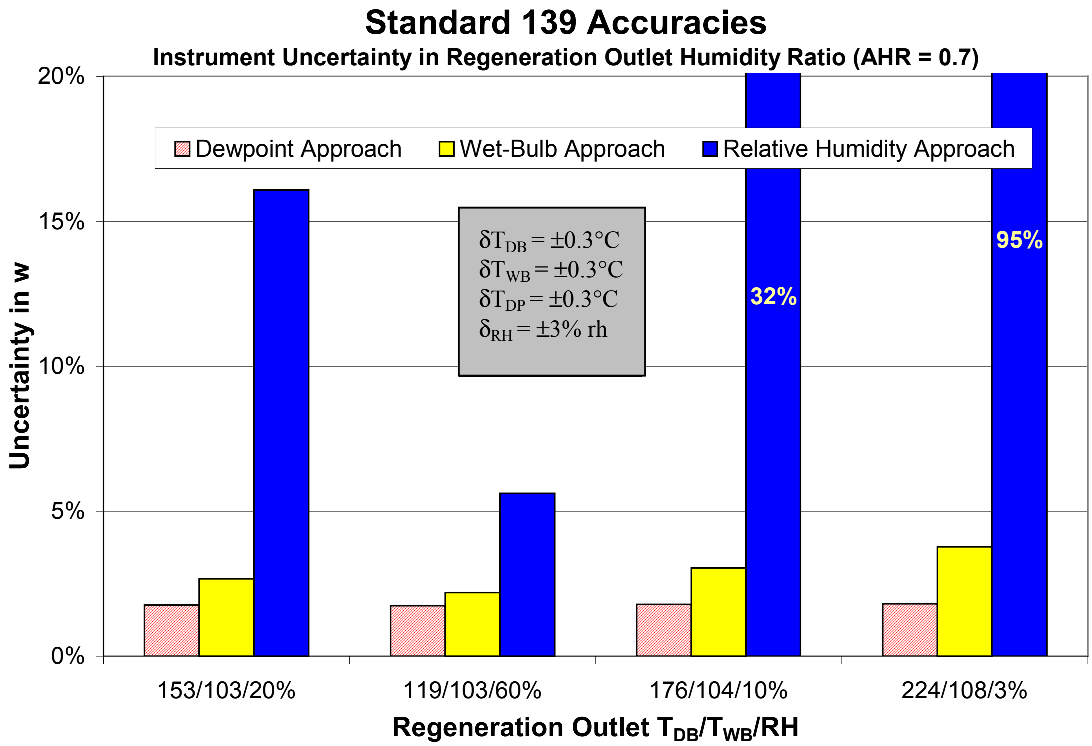

Figure 14. Instrument uncertainty in calculation of regeneration outlet humidity ratio assuming AHR = 0.7 (Standard 139 Accuracies).

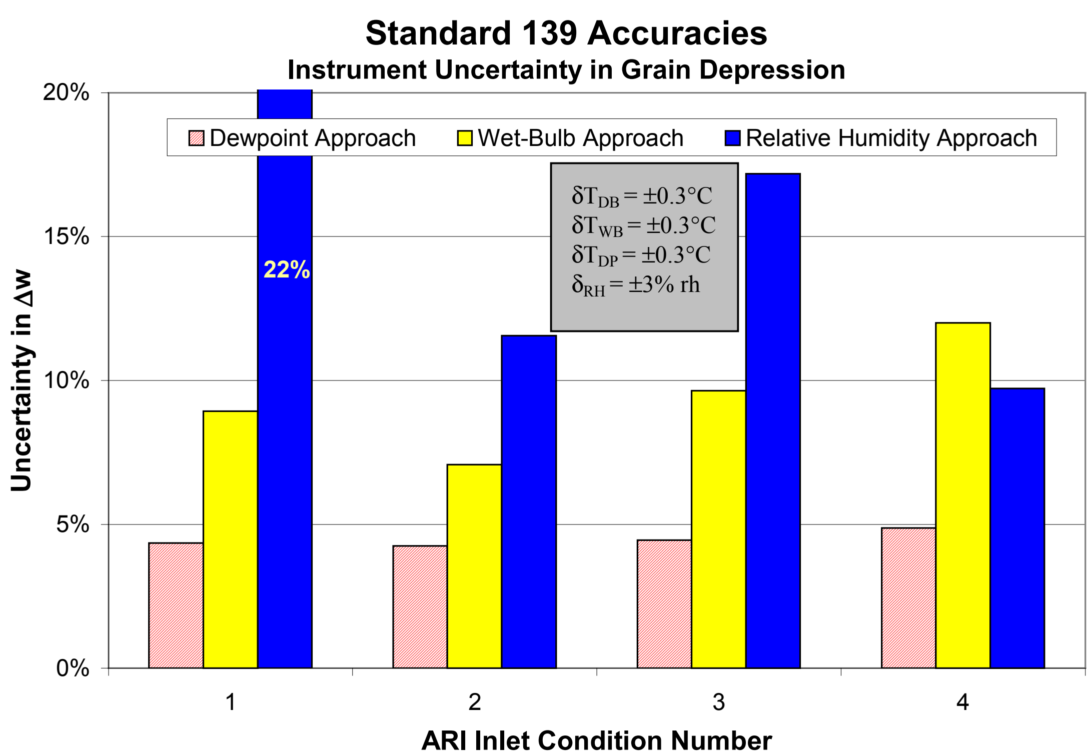

Figure 15. Instrument uncertainty in calculation of delta w on the process side assuming an isothermal process (Standard 139 Accuracies).

A typical uncertainty in the mass flow rate ($\delta\dot{m}$) with the use of ASME nozzles is ±3% of the flow rate. A carefully constructed flow station, combined with highly accurate instruments, can reduce this uncertainty to as low as ±1%. As shown in Table 3, the Standard 139 accuracies produce an uncertainty of ±3%, and the high accuracies produce ±1%. Using these uncertainties in $\dot{m}$ and the uncertainties given in Figure 15, Eq. 28 is used to determine the uncertainty in MRC. Figure 16 gives the instrument uncertainty in MRC using instruments meeting the Standard 139 accuracies. Because Figure 16 shows the "isothermal case," which is the best case for wet-bulb and relative humidity sensors, it highlights the need to post-cool the PO air stream to achieve reasonably accurate results using standard instrumentation. Trying to directly measure non-cooled PO conditions with these sensors will lead to much higher uncertainties than those presented in Figure 16.

Figure 17 gives the instrument uncertainty in MRC using "High Accuracy" instruments. The instrument uncertainty for both the wet-bulb and relative humidity methods has come down considerably using these instruments, again assuming post-cooling to bring the PO temperature back down to its inlet condition. The improvement for the dew-point method is also significant because of the improved uncertainty in mass flow rate.

Instrument Uncertainty Propagated into RSHI

A similar approach is used in calculating the instrument uncertainty in RSHI. Applying the root sum square method (Eq. 24) to RSHI (Eq. 2), results in the following equation:

$$\delta RSHI = \left(\left(\frac{\partial RSHI}{\partial \dot{E}_{\mathrm{Re}\,gen}} \delta \dot{E}_{\mathrm{Re}\,gen} \right)^2 \left(\frac{\partial RSHI}{\partial MRC} \delta MRC \right)^2 \right)^{1/2} \qquad (29)$$

where δMRC is given in Figures 16 and 17. The accuracy in measuring regeneration power will depend on the source energy (gas-fired or electric) and the instrumentation used to measure it. This analysis assumes an accuracy of ±3% for measuring the regeneration power. Figures 18 and 19 give the instrument uncertainty in calculating RSHI for the Standard 139 accuracies and the "High Accuracy" instruments, respectively.

Instrument Uncertainty Propagated into Moisture Mass Balance

Standard 139 requires the moisture mass balance to stay within 5% of 1.0. This requirement is achievable and should be followed, but does not imply that the accuracy of the moisture mass balance is 5%. In fact, experience and analysis show that getting a 5% uncertainty in moisture mass balance is quite difficult. Using the same procedures as above for calculating the root sum square uncertainty, Figures 20 and 21 illustrate the instrument uncertainty in calculating the moisture mass balance for the Standard 139 accuracies and the "High Accuracy" instruments, respectively. This analysis assumes the regeneration outlet air has been cooled, prior to its humidity measurement, to raise its relative humidity to 60%. This is necessary to keep the uncertainty using the relative humidity approach within reasonable limits. And although this cooling has little effect on the instrument uncertainty using the wet-bulb approach, it does reduce the wet-bulb depression to a reasonable magnitude (<10^{o}C). If the regeneration outlet air is not cooled prior to its humidity measurement, the uncertainty encountered in calculating the regeneration outlet humidity ratio using the relative humidity method (Figure 14), can induce a very large instrument uncertainty in moisture mass balance (>100%). Therefore, like the process outlet air stream, if the wet-bulb or relative humidity method is used to measure the moisture content of the regeneration outlet air, the air should first be cooled, raising the relative humidity above 50%, to achieve reasonable accuracy. Figure 21 shows that the instrument uncertainty in the moisture mass balance, using the best commercially available dew-point hygrometers coupled with other high accuracy instruments, is typically 3-4%.

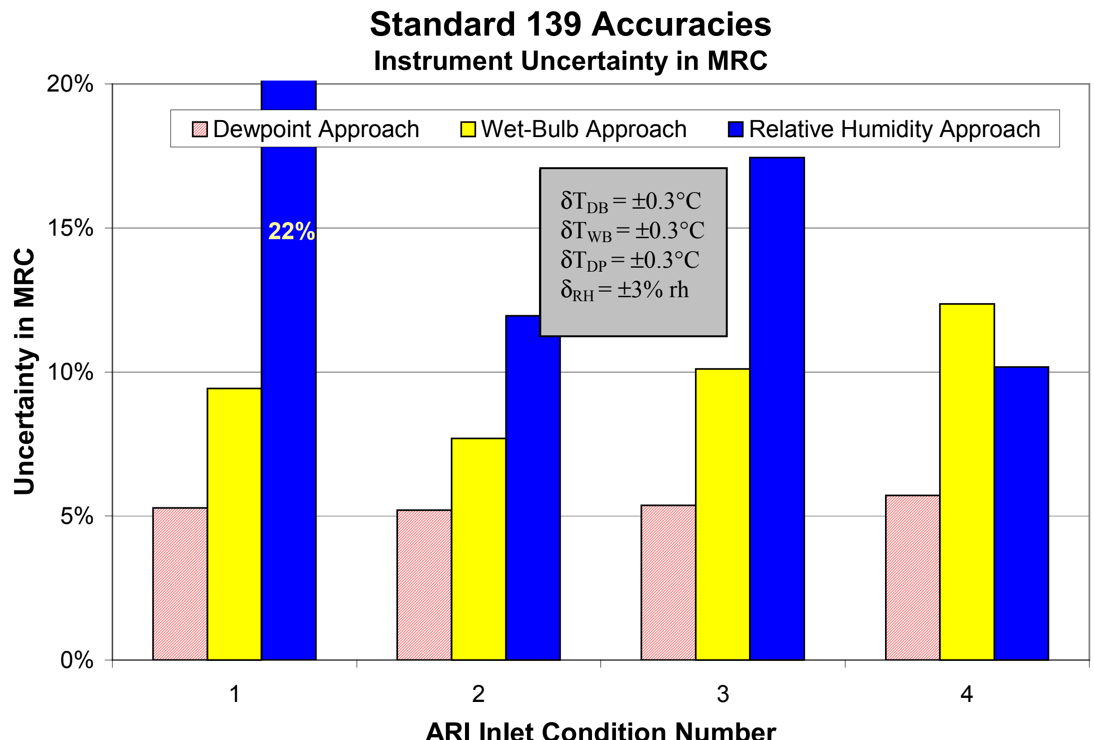

Figure 16. Instrument uncertainty in calculation of MRC assuming an isothermal process (Standard 139 Accuracies).

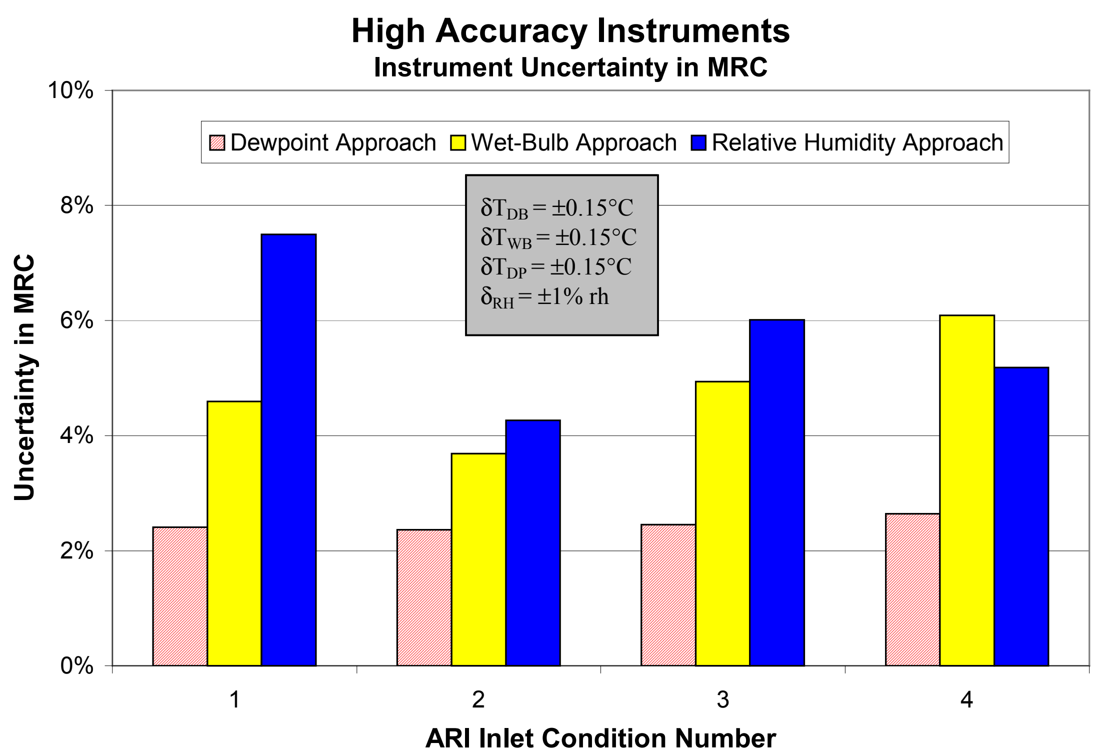

Figure 17. Instrument uncertainty in calculation of MRC assuming an isothermal process ("High Accuracy").

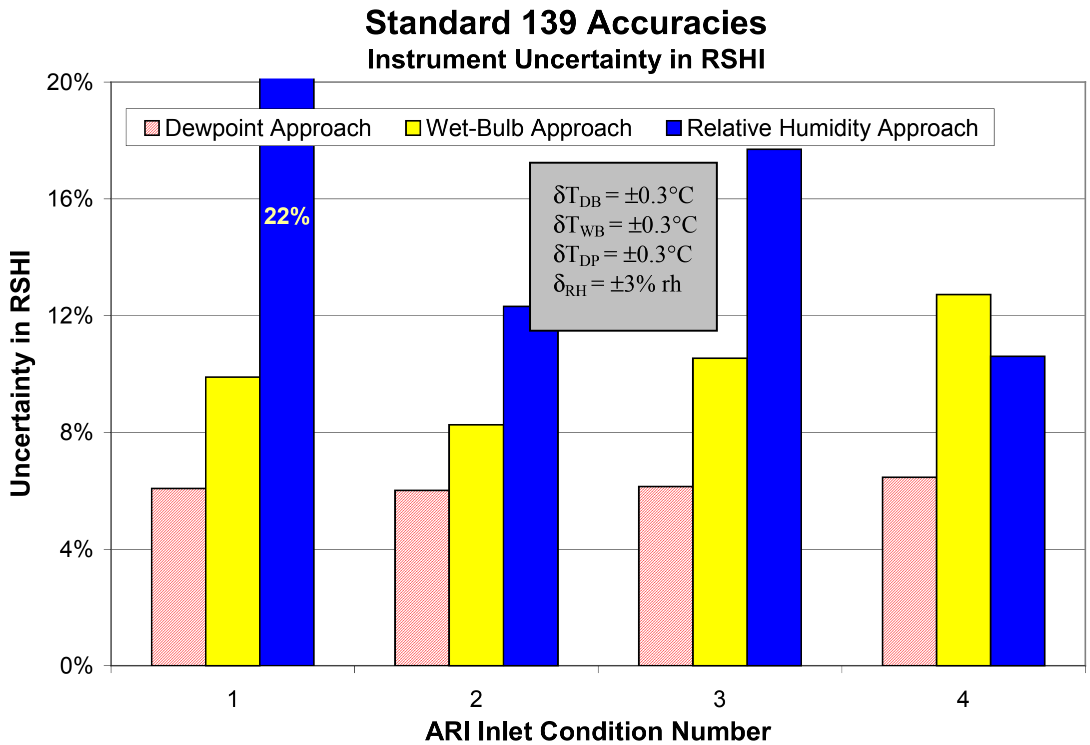

Figure 18. Instrument uncertainty in calculation of RSHI (Standard 139 Accuracies).

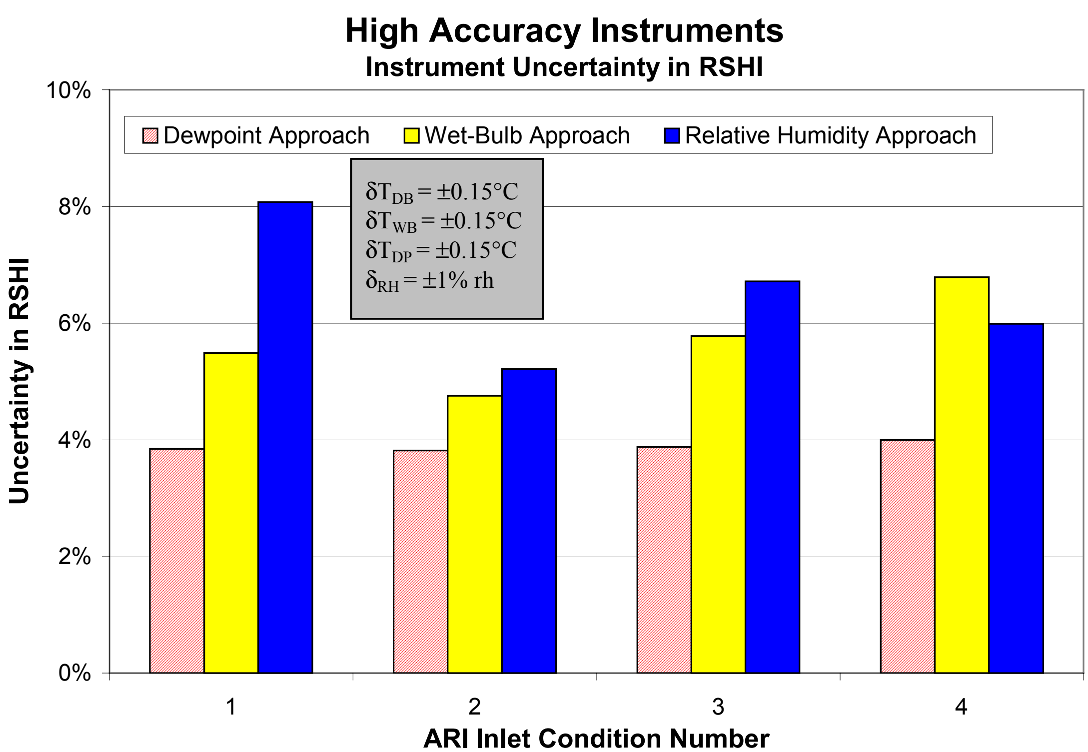

Figure 19. Instrument uncertainty in calculation of RSHI ("High Accuracy").

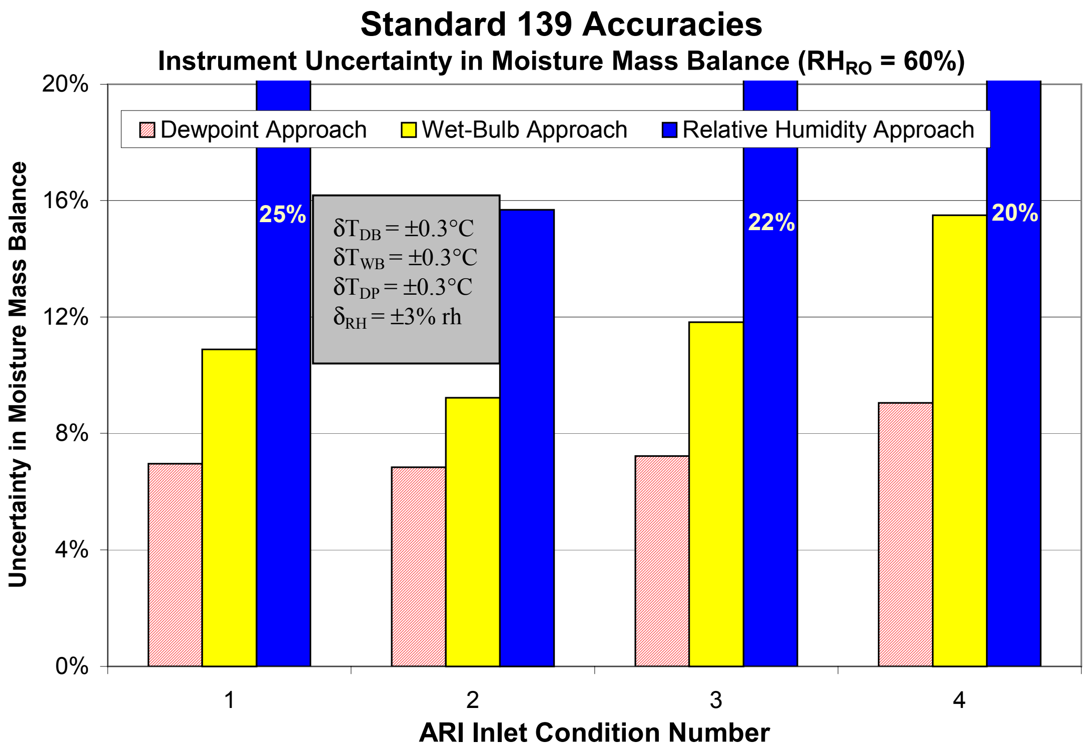

Figure 20. Instrument uncertainty in calculation of moisture mass balance (Standard 139 Accuracies).

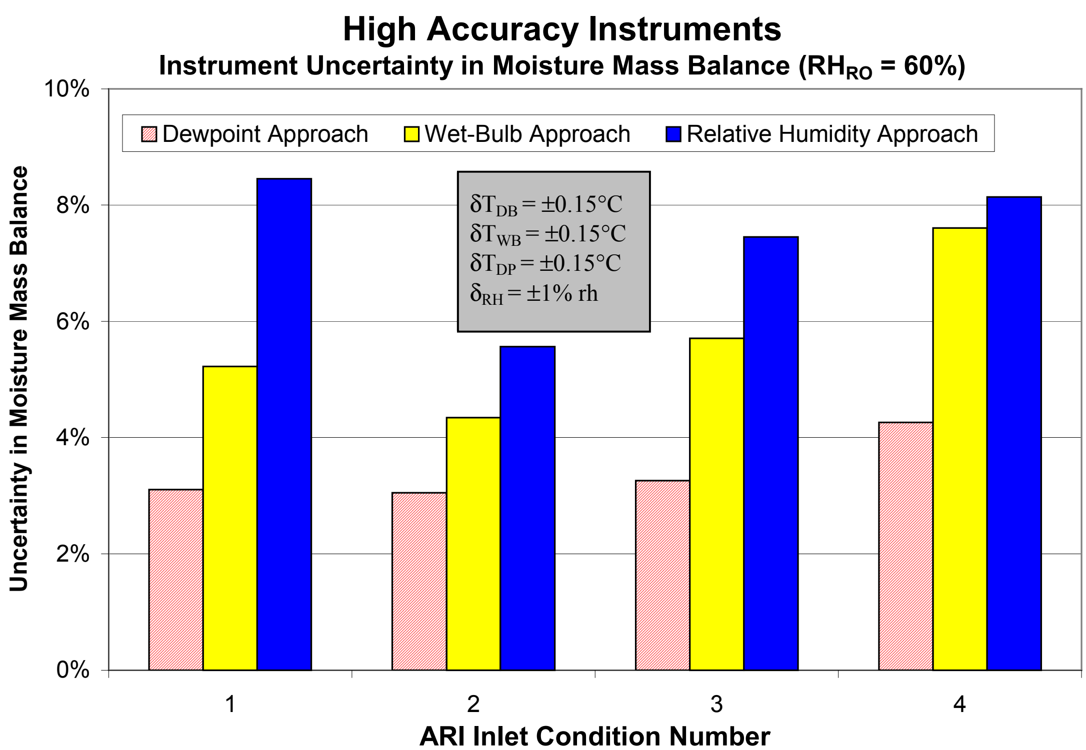

Figure 21. Instrument uncertainty in calculation of moisture mass balance ("High Accuracy").

Safeguards

High Temperature Air

Actively regenerated desiccant dehumidification rotors can be regenerated with air at temperatures up to 400°F. Ducts and cabinets carrying this air should be well insulated to prevent contact with personnel. The temperature of the regeneration air should be continuously monitored by a high temperature limit controller separate from the computer to prevent the possibility of excessively high temperature air from entering the test article and causing damage to the rotor or seals. Where possible, make measurements (moisture, flow, etc.) prior to heating, so sampling tubes and instruments are not required to handle high temperature.

Manual Control of Fans

Typically, fans can be controlled by either the data acquisition system or manually. Manual control is recommended due to the damage that can be caused by a fan receiving a faulty control signal. The fans do not need continuous adjustments for steady-state testing and, therefore, lend themselves to manual control.

LiCl Wheels

If exposed to high relative humidity air without sufficient regeneration, lithium chloride (LiCl) rotors will deliquesce, a phenomenon whereby the desiccant over-adsorbs moisture to the point where damage occurs. LiCl has a very high capacity for holding moisture. If it is not regenerated, the LiCl will continue to absorb moisture until it becomes such a weak solution that it can drain out of the matrix. The combination of water weight and the soaking effect can also destroy the structural integrity of the matrix. Either will ruin the rotor's ability to dehumidify. The manufacturer should be contacted for advice on the maximum relative humidity to which the rotor should be subjected. In addition, the rotors should be stored in airtight containers or bags to prevent damage during storage.

Initial Running of Rotors

The first time air is passed through a rotor, desiccant dust and fumes from the manufacturing process may cause problems for sensitive personnel. An initiation period of approximately two hours is recommended, during which the outlet air is not sampled and is exhausted outdoors.

APPENDICES

Appendix 1a. Definitions Currently Under Consideration by ASHRAE

Desiccant Cooling:	The use of desiccant dehumidification systems for latent heat (humidity) removal in air conditioning systems in conjunction with sensible heat exchange and/or evaporative cooling equipment to perform cooling. This process may use vapor compression equipment to provide the final stage of cooling. Desiccant cooling equipment is used to treat outdoor air by reducing both the sensible and latent (temperature and humidity) loads during cooling seasons. Some systems also include winter heating and humidification components for all season service.
Desiccant Dehumidification **Static:**	The removal of moisture from humid air by exposing the air to a desiccant without the use of an air-moving device. Once the desiccant achieves moisture equilibrium with the ambient air, the dehumidification process stops and the desiccant must be replaced with regenerated desiccant, e.g., packaged or "bagged" desiccant placed in shipping containers to prevent moisture condensation during transit or storage; also laboratory desiccators.
Active:	The removal of moisture from the air by exposing the humid air to a desiccant using an air-moving device and a concurrent reactivation (regeneration) process such as by applying thermal heat or pressure swings. The reactivation process can be either intermittent, such as "dual power" dehumidifiers, or concurrent and continuous such as for "rotary" dehumidifiers.
Passive Dehumidification:	The removal of moisture from the air by exposing the humid air to a desiccant with the use of an air-moving device and a temperature/humidity sink (i.e., difference in the partial vapor pressure of moisture between the ventilation (outdoor) air and exhaust air streams). For example, enthalpy wheels that are used to reduce the impact of ambient temperature and humidity of ventilation air on cooling and heating systems.

Appendix 1b. Terminology Used in the Desiccant Wheel Test Guide

Absorption	Sorption phenomenon characterized by chemical combination with the sorbent material. Typical for liquid desiccants.
Adsorption	Sorption phenomenon characterized by physical inclusion within a sorbent's internal pore structure. Typical for solid desiccants.
Adsorption Heat Ratio (AHR)	The ratio of sensible heat gain due to adsorption to the actual sensible heat gain. If AHR = 1.0, the process is adiabatic. As AHR drops, the process becomes less adiabatic, indicating increasing heat dump.
Air mass balance:	Either the ratio of inlet/outlet mass flow pairs, or the ratio of the summations of all inlets to outlets
Blowthrough:	A highly non-uniform inlet airflow that overcomes a wheel's natural ability to even out air distribution and degrades performance by starving some sections of the wheel
Breakthrough:	Preventable design condition when the mass exchange wave front reaches the outlet face of the wheel before the wheel rotates out of the current airflow
Cassette:	The wheel housing, including the drive motor and air seals
Carryover:	Transfer of air containing moisture or other contaminants from regeneration to the process side of the wheel, either by seal leakage or wheel rotation
Circumferential seals:	The air seals that connect the rotating wheel to the cassette
Co-sorption:	Desiccants' very low potential for adsorbing contaminant molecules along with water vapor
Desiccant Loading:	Percent of rotor weight made up by desiccant—typically about 80% for active wheels, and about 15% for passive
Desiccant Isotherm:	Traditional graphical display of moisture pick-up by desiccant type—ordinate is relative humidity to which the desiccant is exposed, abscissa is either mass of moisture pick-up per mass of desiccant, or percentage of mass pick-up relative to maximum pick-up
Dump-back:	Transfer of heat from regeneration air to process air either by carryover or within the matrix itself
Face Seals:	Air seals on the face of the wheel that divide the process and regeneration airflows and establish the wheel split
Face Velocity:	Nominal average air velocity approaching the wheel—volume flow rate divided by wheel face area
Flute:	Individual airflow passage through the wheel
Flute Geometry:	Physical shape of a flute including aspect ratio—affects the relationship between air-side mass transfer and pressure drop
Flute Velocity:	Actual average air velocity within the wheel—related to face velocity by matrix open area
Heat of Sorption:	Sensible heat released into the process air and matrix when desiccant sorbs water vapor

Mass Flow Ratio:	Ratio of regeneration air mass flow to process air mass flow
Matrix:	Wheel core consisting of a substrate impregnated or coated with desiccant—provides large surface area required for mass transfer
Moisture mass balance:	Figure of merit that indicates quality of test—the ratio of measured moisture drop in the process air to measured moisture rise in the regeneration air—must be between 0.95 and 1.05
Moisture Removal Capacity (MRC):	Dehumidification rate in pounds of dehumidification per hour—(also performance)
Open Area:	Percent of wheel face area not taken up by the matrix
PI:	Process inlet
PO:	Process outlet
Process:	The airflow or side of the wheel where dehumidification occurs—taken from industrial terminology ("supply" in commercial terms)
Primary energy:	Fossil fuel combusted to produce heat or electricity—(also source energy)
Regeneration:	The airflow or side of the wheel where desorption occurs
Regeneration Specific Heat Input (RSHI):	Energy input to the regeneration heater per pound of dehumidification (also efficiency)
Regeneration Specific Heat Input with pre-heat ($RSHI_{HX}$):	Energy input per pound of dehumidification including the benefit of heat exchange between process outlet air and regeneration air prior to the heater
Regeneration Specific Heat Drop (RSHD):	Energy consumed by the wheel per pound of dehumidification
Residence Time:	Amount of time air spends within the wheel—wheel depth divided by flute velocity
RI:	Regeneration inlet
RO:	Regeneration outlet
Rotor:	Wheel
Source energy:	Fossil fuel combusted to produce heat or electricity, (also primary energy)
Std. 139:	ASHRAE Standard 139 "Method of Testing for Rating Dehumidifiers Utilizing Heat for the Regenerative Process"
Std. 940:	ARI Standard 940 "Desiccant Dehumidification Components", the basis for its upcoming National Certification Program for these products.
Wavefront:	Region within the wheel where mass transfer is occurring—varies axially and circumferentially
Wheel:	Disk consisting of the matrix, rim, and possibly a hub with spokes and axle for support (also rotor)

Wheel Depth:	Thickness in the flow direction
Wheel Diameter:	Largest wheel dimension
Wheel Speed:	Rotation rate—given in rotations per minute for passive and heat-exchange wheels, in revolutions per hour for active wheels.
Wheel Split:	Percent of wheel face area allocated to process vs. regeneration (e.g., 75/25 or 50/50)
1M:	Type 1 Moderate—Theoretically ideal desiccant isotherm shape required for effective/efficient performance at higher regeneration temperatures

Appendix 2. Nomenclature

A	Convective transfer surface area (ft^2)
AHR	Adsorption Heat Ratio
c_p	Specific heat of air (Btu/lb/°F)
C*	Ratio of minimum to maximum air heat capacity rates in an air-to-air heat exchanger (unitless)
C_r	Ratio of heat capacity rate of the matrix to the minimum air heat capacity rate in a rotary air-to-air heat exchanger
COP_{latent}	Coefficient of performance for latent cooling
d	Wheel depth (ft)
$\dot{E}_{regen}$	Thermal energy input rate for active dehumidification (kBtu/h)
$\dot{E}_{para}$	Parasitic (electric) energy input for fans, wheel drive, etc (kBtu/h)
f	Pressure drop friction factor (unitless)
ΔGPP	Absolute humidity depression of the process air (grains/lb_{dryair})
ΔGPP_{rt}	Absolute humidity depression of the process air per second of residence time (grains/lb_{air}/s)
h	Convective heat transfer coefficient (Btu/hr/°F/ft^2)
h_m	Mass transfer analog to thermal convection coefficient h (lb/hr/ft^2)
M	Mass of a rotary exchanger matrix (lb)
$\dot{m}$	Mass flow rate of air (lb/hr)
$\dot{m}_{in}$	Moisture flux into the cassette ($lb_{moisture}$/min)
$\dot{m}_{out}$	Moisture flux out of the cassette ($lb_{moisture}$/min)
MRC	Moisture Removal Capacity ($lb_{moisture}$/hr)
MRC_{Btuh}	Moisture Removal Capacity as a latent cooling rate (kBtu/hr)
MRC/P_f	Moisture Removal Capacity per Watt of fan power needed to move air through both sides of the wheel ($lb_{moisture}$/hr/W)
MRR	Moisture Removal Regeneration ($lb_{moisture}$/hr)
NTU	Number of transfer units for a heat exchanger (unitless)
NTU_{mass}	Mass transfer analog to NTU (unitless)
p	Thermodynamic (static) air pressure (pascals)
P_f	Fan power required to overcome the wheel pressure drops (W)
Δp_p	Pressure drop across the process side of the wheel (pascals)
Δp_r	Pressure drop across the regeneration side of the wheel (pascals)
p_v	Vapor pressure of water (pascals)
p_{vs}	Saturation vapor pressure of water (pascals)
p_{vs}^*	Saturation vapor pressure of water evaluated at the wet-bulb temperature (pascals)
Q	Process air volume flow rate (ft^3/min)
RSHI	Regeneration Specific Heat Input Energy (kBtu/$lb_{moisture}$)

$RSHI_{HX}$	Regeneration Specific Heat Input Energy including the benefit of heat exchange between process outlet air and regeneration air prior to the burner ($kBtu/lb_{moisture}$)
RSHD	Regeneration Specific Heat Drop ($kBtu/lb_{moisture}$)
St	Stanton Number – convective heat or mass transfer rate (unitless)
T	Temperature (F)
t	Dry-bulb temperature (F)
t*	Thermodynamic wet-bulb temperature (approximated as wet-bulb temperature) (F)
$T_{PO,adiabatic}$	The temperature achieved upon reaching a given grain depression with no change in enthalpy—essentially evaporative cooling in reverse
V_{fl}	Flute velocity (ft/min)
w	Absolute Humidity Ratio ($lb_{moisture}/lb_{dryair}$)
Δw	Absolute humidity depression of the process air (w_{PI} - w_{PO}) ($lb_{moisture}/lb_{dryair}$)
w_s^*	Humidity ratio at the thermodynamic wet-bulb temperature ($lb_{moisture}/lb_{dryair}$)
Greek symbols:	
ε_{cf}	Counter-flow heat exchange effectiveness (unitless)
Φ	Decimal representation of relative humidity (unitless)
ϕ	Rotational frequency of a rotary exchanger (rph)
η_{fan}	Efficiency of the fans (unitless)
η_{motor}	Efficiency of the fan motors (unitless)
ρ_{std}	Standard density of air (0.075 lb/ft^3)
Subscripts:	
DB	Dry-bulb
DP	Dew-point Approach for humidity measurement
j	Hot or cold side of the wheel
max	The airflow in a heat exchanger with the larger heat capacity rate
min	The airflow in a heat exchanger with the smaller heat capacity rate
P	Process
PI	Process inlet
PO	Process outlet
R	Regeneration
RH	Relative Humidity Approach for humidity measurement
RI	Regeneration inlet
RO	Regeneration outlet
rt	Residence time of air inside the exchanger matrix
WB	Wet-bulb Approach for humidity measurement

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NREL's Desiccant Dehumidification Wheel Test Guide intends to facilitate execution of the new standards by certification labs and manufacturers. The Test Guide is a product of more than 20 years of experience gained at NREL's desiccant research facilities. It details practical experimental experience with rotary mass exchangers in relation to the standards and aims to develop testing expertise in industry quickly and cost effectively.

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