Developing a Standard Method of Test for Packaged, Solid-Desiccant Based Dehumidification Systems

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Abstract

A draft Method of Test (MOT) has been proposed for packaged, air-to-air, desiccant-based dehumidifier systems that incorporate a thermally-regenerated desiccant material for dehumidification. This MOT is intended to function as the "system" testing and rating compliment to the desiccant "component" (desiccant wheels and/or cassettes) MOT (ASHRAE 1998) and rating standard (ARI 1998) already adopted by industry. This draft standard applies to "packaged systems" that:

- Use desiccants for dehumidification of conditioned air for buildings;
- Use heated air for regeneration of the desiccant material;
- Include fans for moving process and regeneration air;
- May include other system components for filtering, pre-cooling, post-cooling, or heating conditioned air; and
- May include other components for humidification of conditioned air.

The proposed draft applies to four different system operating modes depending on whether outdoor or indoor air is used for process air and regeneration air streams. Only the "ventilation" mode which uses outdoor air for both process and regeneration inlets is evaluated in this paper. Performance of the dehumidification system is presented in terms that would be most familiar and useful to designers of building HVAC systems to facilitate integration of desiccant equipment with more conventional hardware.

Parametric performance results from a modified, commercial desiccant dehumidifier undergoing laboratory testing were used as data input to evaluate the draft standard. Performance results calculated from this experimental input, results from an error-checking/heat-balance verification test built into the standard, and estimated comparisons between desiccant and similarly performing conventional dehumidification equipment are calculated and presented. Some variations in test procedures are suggested to aid in analytical assessment of individual component performance.

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Introduction

Several years ago, ASHRAE published a standard (ASHRAE 1998) that provides test methods for determining the moisture removal capacity of heat-regenerated desiccant components and the energy requirement for regeneration. The method of test (MOT) did not consider ancillary equipment such as fans, coils, fuel-consuming regeneration systems, and pre- and post-conditioning components that may be included to complement a desiccant-based dehumidifier system. The MOT also does not address the impact that the desiccant component has on the sensible temperature of conditioned air. Therefore, the rating information provided by the standard is primarily of value to purchasers of desiccant components to be installed in field-erected systems or to be included in a packaged desiccant-based dehumidifier product by a manufacturer.

There are several packaged desiccant-based dehumidifiers available that include many of the ancillary components mentioned above. An HVAC system designer who considers a packaged product needs to have complete information on the conditioned-side outputs and energy inputs to evaluate and compare options. Therefore, there is a need for a method of test and rating procedure that provides both the latent cooling and the sensible cooling (which could be negative-with the leaving process air warmer than inlet air-for these products) performance of a packaged product.

An effort is underway to fill this need. A preliminary draft standard for a method of test has been prepared and testing is underway to evaluate its provisions. In January 2001, the formation of an ASHRAE Standard Project Committee (SPC 174) was initiated to review this early work as a starting point with the objective of preparing an ASHRAE draft standard for public review and comment. This paper summarizes the status of that effort.

Desiccant-based dehumidifying systems have proven successful in handling latent loads in many, specialized applications. These include supermarkets, indoor ice rinks, hospitals, schools, high tech manufacturing facilities, etc. But, for desiccants to be used on a broader basis for comfort conditioning, it must be demonstrated that they are also suitable to meet the needs of more conventional HVAC applications.

Packaged unitary air conditioners are widely used because they provide the HVAC system designer with a complete, factory-built space conditioning system with certified cooling and/or heating capacity at standard rating conditions. These products have typically provided rated cooling capacities with a Sensible Heat Ratio (SHR) of about 75%; that is, 75% of the cooling delivered is sensible (decreased dry bulb temperature) cooling and the balance is latent cooling (moisture removal). In the past, equipment with SHRs near 75% has provided satisfactory air conditioning for buildings by reasonably matching the SHR of most building cooling loads.

Recent trends toward reducing energy consumption through tighter building envelopes and improving indoor air quality with increased fresh air ventilation rates have decreased the SHR of cooling loads. As a result, mismatches in the SHR of the building cooling load and the SHR of the cooling capacity of unitary air conditioning equipment are becoming more prevalent. Factoring in the latent loads associated with ventilation air dehumidification results in air systems with SHRs less than 0.5 for many applications.

HVAC system designers are employing various approaches to meet this SHR challenge. Historically, the most widely used approach was to cancel some of the sensible cooling by reheating the supply air with electric resistance heaters or with some form of recovered waste heat. While effective in reducing the SHR, this approach is definitely not energy efficient. Reducing the level of evaporator airflow will lower the SHR, but it also lowers the overall efficiency of the air conditioner. Another approach that is often employed is the use of run-around loops that pre-cool the return air and use the recovered heat to reheat the supply air. This requires a relatively small, additional energy use for pumping the heat exchange fluid and/or overcoming the airside pressure drop of the two additional heat exchangers in the supply air stream. A similar approach that accomplishes the same result without added energy use for pumping makes use of a heat pipe that wraps around the evaporator coil. While these approaches can moderate the mismatch between equipment and load SHRs they are somewhat limited by the extent that they can lower the SHR of the HVAC equipment.

The proposed method of test for packaged desiccant-based equipment provides performance information in a form that is most familiar to an HVAC design engineer. That is, cooling performance is reported in terms of total, sensible, and latent cooling. The cooling performance would also be reported in terms of the air flow quantity and the dry-bulb and wet-bulb temperatures of the leaving process air. These temperatures permit locating the leaving air condition of the dehumidifier at a point on a psychrometric chart. On the chart, this point will usually be well to the right of (for example, drier and warmer than) the temperature conditions of the air leaving a typical unitary air conditioner. A unitary air conditioner with a SHR in the mid-seventy percent region will have a leaving dry-bulb temperature in the 55 to 60º F (12.8 to 15.6º C) range, at near saturation conditions. Using the 0-rpm heat recovery wheel data latter in Table 3, the reader can see that a desiccant-based air conditioner without heat recovery could have a process air leaving temperature of 155-160° F (68-71° C), whereas with heat recovery this temperature drops to 95-108° F (35-42° C) range.

To achieve a reduced SHR, the process air from the packaged dehumidifier can be mixed with the air leaving a conventional air conditioner. Mixing air at one condition with some air at another condition is represented on a psychrometric chart by a straight line drawn between the two points representing the two air conditions. The condition of the resulting mixture will fall on this line at a point determined by the relative airflow quantities. For example, if the two air flow rates are equal, the mixture condition will be halfway between the conditions of the two mixture components. If one airflow rate is twice the magnitude of the other, the mixed condition will be two-thirds the way along the line at a point nearer the point representing the higher airflow rate or quantity, see Figure 1.

Mixed air from the dehumidifier and the conventional air conditioner can conveniently constitute the supply air to the building. By matching an appropriate packaged desiccant dehumidifier with a packaged unitary air conditioner, the HVAC system engineer has the ability to meet the peak SHR loads of the building. The SHR of the output of the system can be easily adjusted when latent loads are lower by controlling one or more of the dehumidifier's components, such as reducing the heat input to the desiccant regeneration process or decreasing the rotational speed of the desiccant wheel.

The proposed method of test for the desiccant dehumidifier utilizes procedures that are similar to those which have been used for decades in the testing of unitary air conditioning products. The test facility consists of two adjacent, ambient-controlled rooms to provide both indoor and outdoor temperature/humidity conditions. The test provides a measurement of the enthalpy change in the process air stream and the resulting leaving air dry bulb and dew point or wet bulb temperatures. The procedure also records similar measurements of the regeneration air stream and the energy inputs are also recorded. The results of the test are validated by performing a heat balance (Heat rejected in the regeneration air path = Heat removed in the process air path + Energy added to the system).

Experimental

Laboratory Data

Experimental data used for this analysis were taken using a desiccant dehumidification system as shown in Figure 2. This system was not installed in ambient climate chambers simulating indoor and outdoor temperature/humidity levels as specified in the draft MOT, but was equipped with instrumentation to measure air dry bulb and dew point temperatures at the principal points and process air and regeneration air flow rates at system exits p3 and r5 in Figure 2. Requirements for accuracy of test instrumentation were in accordance with ASHRAE Standard 139-1998 (ASHRAE 1998). A chilled mirror was used to measure dew point temperatures because these measurements have accuracy and precision advantages over relative humidity and simultaneous dry bulb temperature measurements or wet-bulb temperature measurements (Jalalzadeh-Azar et al. 1996).

The term “process” air is used throughout this paper to describe the air stream dehumidified by the desiccant dehumidification system and eventually used for building conditioning. This is a term more appropriate to desiccant-dried air used for “process” applications like mothballing sensitive equipment, pharmaceutical or confectionery production, and the manufacturing of lithium batteries. But process air is used here because it would also be inappropriate to constantly refer to desiccant-dried air used for comfort conditioning as “supply” or “conditioned” air, because these terms have also taken on implied meaning not applicable to the desiccant system output. Process air is commonly used to describe air dried by a desiccant system and should generate less confusion in the subsequent descriptions.

The inlet dry bulb temperature for the process air stream was maintained within ± 0.5ºF(± 0.3ºC) of the desired set point by using a 10-kW duct heater and through controlled adjustment of a ducted by-pass from the regeneration outlet duct to the process air inlet used when the heater, alone, is inadequate. A 30-kW duct heater was used to regulate the air temperature for the regeneration air stream. Wet bulb temperatures on both air streams were maintained within ± 0.5ºF (± 0.3ºC) by manually controlled introduction of steam from process lines.

Wheel speeds for the desiccant and thermal heat recovery wheels were determined by marking the perimeter of the rotors and counting the revolutions while measuring time with a stopwatch. Motors, equipped with variable frequency drives, were used to control these rotational speeds.

The uncertainties of measurements at the process air inlet and exit air streams are primarily due to instrumental precision, bias, or drift rather than spatial nonuniformities. These uncertainties and their effect on values calculated from them are shown in Tables 1 and 2. The inlet properties of the process and regeneration air streams are uniform. Spatial nonuniformities of properties at the process air exit are minimized through the use of mixing baffles and duct fittings upstream of the instrumented plane. However, dry bulb and dew point temperature measurements made downstream of the desiccant wheel on the process side (p2 in Figure 2) and those made downstream of the recovery wheel on the regeneration side (r3 in Figure 2) are highly susceptible to variations caused by spatial nonuniformity (Jalalzadeh-Azar, et al, 1998). These nonuniformities are due to the cyclic operation of the desiccant and regeneration components and the large volumes of air moving through the system. When fluid state points are required at one of these intermediate locations, an analytical approach is used to determine the fluid properties (Jalalzadeh-Azar, et al, 2000).

For each set of inlet conditions, an experimental run is conducted with the thermal recovery wheel deactivated (0 rpm). The properties of the dried air exiting the desiccant wheel (state p2) when the wheel is activated, are set to be equal to those of the process air exiting the system (state p3) when the recovery wheel was not turning. Any uncertainty stemming from this assumption is considerably less than that associated with the spatial nonuniformities encountered by direct measurements of these properties (Jalalzadeh, et al, 2000).

Baseline system tests were performed at two ambient conditions. One with process and regeneration air at a 95ºF (35ºC) dry bulb temperature, 75ºF (23.9ºC) wet bulb temperature and another with both inlets at a 80ºF (26.7ºC) dry bulb, 75ºF (23.9ºC) wet bulb temperature. These are two rating conditions specified in the ARI Desiccant Dehumidification Component rating standard (ARI 940-1998). The desiccant wheel was rotated at 80 rph and both process and regeneration airflow rates were adjusted to 2500 cfm with inverter driven fans. This volume flow rate results in an approximately 400 ft/min. (≈ 120 meter/min.) face velocity with a 6.3 ft^2 (0.58 m^2) process-side wheel surface area. The desiccant wheel regeneration air temperature was held at 190ºF (87.7ºC) as described in previous publications (Vineyard, et al, 2000).

Modeling Data

A proprietary, desiccant system-modeling program provided by the manufacturer of the desiccant equipment, was used to compare modeled system performance to experimentally measured performance at the same inlet conditions, flow rates and operating conditions.

Method of Test Details

The draft method of test being evaluated is written in standard ASHRAE format. A spread sheet format is provided to facilitate calculation of the performance of the packaged desiccant system from the test data. The spreadsheet also projects the overall performance when the desiccant system is coupled with conventional HVAC equipment. These projections are then compared to a baseline case (conventional HVAC equipment with reheat) to show relative performance factors. Imbedded formulas in cells perform the necessary calculations. Experimental inputs required are dry bulb and wet bulb temperature inputs for outdoor air, process and regeneration air inlets, and process and regeneration air outlets. Wet bulb temperatures for this work were calculated from measured dew point and dry bulb temperatures for reasons previously stated. Additional experimental inputs include process and regeneration air flow rates (volumetric) in standard cubic feet per minute (SCFM), regeneration energy input (either fuel consumption + efficiency or Btu/hour), and parasitic electric power (fans and wheel motors – watts) inputs.

In order to make performance and economic comparisons between desiccant-assisted and vapor compression systems with reheat, input values for:

- indoor air conditions,
- air conditioner evaporator air flow rate (scfm/ton),
- sensible heat ratio,
- energy efficiency ratio (EER – [Btu/hr]/watt),
- electric source-fuel efficiency (%),
- fuel-source efficiency (%),
- fuel cost per therm ($/therm - HHV), and
- electric energy cost ($/kW)

must also be supplied. The automated spreadsheet model calculates desiccant system performance in terms of total cooling capacity (Btu/hour), sensible cooling capacity (Btu/hour), and latent cooling capacity (Btu/hour). Performance of a combined desiccant/conventional, electrically driven AC system is calculated and compared to an all-electric air conditioning system with reheat. The relative performance of the desiccant/conventional system to an air conditioner with reheat is calculated as a series of ratios including a source-fuel consumption ratio, energy cost ratio, air conditioning capacity ratio, and supply air flow ratio.

Results and Discussion

MOT Input Data

Laboratory and modeling data used to evaluate the proposed MOT are listed in Table 3. A matrix of input/output conditions with corresponding operating data for laboratory and modeled data at two inlet conditions, including runs where the heat exchange wheel was at 0 and 10 rpm result in eight (8) of the sets given in this table.

Since the experimental plan under which the initial laboratory data was taken focused primarily on process side performance and did not require rigorous measurement of regeneration side leaving conditions, an analytical methodology for calculating an "ideal" regeneration outlet condition was formulated. This ideal regeneration outlet temperature and dew point (wet bulb temperature) are calculated assuming that the more carefully controlled measurements across the inlet and outlet on the process air stream (p1 and p3 on Figure 1) are accurate and valid enough to be used as a basis to calculate what regeneration outlet conditions should be if conservation of energy and mass are assumed.

Dealing with only the desiccant wheel, conservation of energy requires that, at the steady state operating conditions when this data was taken, the $\Delta \boldsymbol{h}_{\text{process}}$ across the more carefully measured process stream (p1 to p2) has to be equal to the $\Delta \boldsymbol{h}_{\text{regen.}}$ across the regeneration side (r4 to r5) at equivalent dry air mass flow rates. Process air leaving the desiccant wheel will be dryer and warmer than the incoming air stream because water removed by the desiccant imparts its latent heat of vaporization to the air in the form of a temperature increase. Data taken when the heat recovery wheel is not turning gives an accurate indication of process airside conditions after the desiccant wheel without measurement uncertainties related to spatial nonuniformities. (Jalalzadeh-Azar, et al 1998) In this experiment, the knowledge that the regeneration air was heated precisely to 190°F (87.8°C) prior to entering the desiccant wheel plus an accurate measurement of regeneration air inlet conditions, establishes a computable state point condition at r3, prior to entering the desiccant wheel on the regeneration side.

Knowing the absolute enthalpy at point r3, the change in enthalpy needed to balance the conditoned side change and the amount of moisture being added to the regeneration stream to achieve mass balance allows explicit calculation of the air state point conditions at r5. Conditions at r3 are identical to those at r1 when the heat recovery wheel is not turning. The direct evaporative cooler in the regeneration air stream, Figure 2, was inactive in these experiments.

When the heat recovery wheel is rotated at 10 rpm, the conditions at r3 can be calculated assuming a thermal effectiveness value of 0.84 for the heat exchanger wheel, which was established, in previous work (Jalalzadeh-Azar, et al, 2000). Knowing the absolute enthalpies at the entrance and exit of the desiccant wheel and assuming equivalent mass flow rates permits

$$\varepsilon = \frac{h_{r5} - h_{r4}}{h_{p1} - h_{r4}} \qquad \textbf{(2)}$$

calculation of an "absolute desiccant wheel effectiveness" which combines latent and sensible effectiveness (equation 2). The "absolute desiccant wheel effectiveness", ε, which also can be used to estimate conditions at point r5 as shown in equation 2. It is gratifying to report that such good agreement was seen between "measured" and calculated regeneration outlet values that only the measured results were used to calculate the validation of test results and system capacity results that follow. Regeneration outlet conditions for the modeled data were calculated by this method however, because the model does not provide these results.

Also, the enthalpy change from r3 to r4 together with the air mass flow rates make it possible to analytically calculate the minimum energy needed to heat the regeneration air stream to the190°F (87.8°C) regeneration temperature used for these experiments. This calculation can serve as a check against experimentally measured regeneration heat input rates.

Enthalpy/Mass Balance Considerations

Ideally, the laws of conservation of energy and mass can be applied to the analyses of thermomechanical systems like this in the form of heat balance and mass balance calculations to serve as an internal check on the quality of experimental measurements and testing procedure. A conceptual illustration of the enthalpy changes across a thermally-regenerated, desiccant-based dehumidification system like the one tested here are depicted in Figure 3. In comparing this drawing to the one shown in Figure 2, please note that the direct evaporative cooling unit, which is a component in the regeneration air path of this desiccant system, was not used during these tests because of the complications it would cause with heat and mass balance considerations.

In Figure 3, enthalpy changes are indicated by:

- Heat and mass transfer between air streams effected by the desiccant wheel,
- Heat transfer from process to regeneration air by the heat recovery wheel,
- Heat input to the regeneration air by the regeneration heating coil, and
- Heat input to both air streams from turbulence and heat from electrically driven fans.

The draft MOT calculates the total capacity, latent plus sensible, of the process side of the desiccant dehumidifier system (primary) by multiplying the absolute enthalpy change of process air entering and leaving the unit by the mass flow rate of air, equation 3. A process-side capacity (confirming) is also calculated based upon regeneration-side enthalpy

$$\dot{\mathbf{Q}}_{\mathbf{Process\,(primary)}} = \dot{\mathbf{m}}_{\mathbf{process}} \left[\mathbf{h}_{\mathbf{Process\,Inlet}} - \mathbf{h}_{\mathbf{Process\,Outlet}} \right] \tag{3}$$

$$\dot{\mathbf{Q}}_{\mathbf{Process\,(confirming)}} = \dot{\mathbf{m}}_{\mathbf{regeneration}} \left[\mathbf{h}_{\mathbf{regen.Outlet}} - \mathbf{h}_{\mathbf{regen.Inlet}} \right] - \dot{\mathbf{Q}}_{\mathbf{regeneration}} - \dot{\mathbf{Q}}_{\mathbf{fans,\,etc}} \tag{4}$$

change data with the regeneration energy input and electric power input to the dehumidifier for both fans, drive motors, etc. subtracted according to equation 4. To be accepted as valid test data the primary total system capacity as calculated by equation 3 must agree with the confirming system capacity (equation 4) to within ± 6%, equation 5.

$$0.94 \langle \frac{\dot{\mathbf{Q}}_{\mathbf{Process(primary)}}}{\dot{\mathbf{Q}}_{\mathbf{Process(confirming)}}} \langle \mathbf{1.06} \qquad \mathbf{(5)}$$

A similar confirming test could be specified based on a mass balance of the water gained by the regeneration side to water lost on the process side. A water mass balance confirmation is used in ASHRAE Standard 139-1998 for component, desiccant wheels.

If the total system capacity "primary" is within ±6% of the total system capacity "confirming" the MOT establishes the total system capacity as $\dot{\mathbf{Q}}_{\text{process (primary)}}$ in equation 3. Sensible cooling capacity for the system is calculated by multiplying the heat capacity of air at standard conditions

$$\delta\,\dot{\mathbf{Q}}_{\mathbf{sensible}} \approx \dot{\mathbf{m}}_{\mathbf{a}}\,\bar{\mathbf{c}}_{\mathbf{p}}\,(\mathbf{T}_{\mathbf{p}3} - \mathbf{T}_{\mathbf{p}1}) \qquad \mathbf{(6)}$$

by the mass flow rate of process air and the temperature change for inlet to outlet across the dehumidifying side of the system, equation 6, where $\bar{\mathbf{c}}_{\mathbf{p}}$ is the average constant-pressure specific heat of the process air.

With desiccant systems, the sensible cooling capacity may be negative, i.e., dried, process air leaving the system is at a higher dry-bulb temperature than the inlet air. Obviously, this is the case for data given in Table 2 when the heat recovery wheel is not turning. Under these conditions, the desiccant dehumidifier imposes an extra sensible cooling load for the building's air conditioning system. Direct and/or indirect evaporative cooling components are often included with packaged desiccant dehumidifiers to provide a means of converting latent capacity to sensible cooling. If this evaporative cooler is on the regeneration side of the system and a heat recovery heat exchanger is used, this can constitute essentially "free" sensible cooling of the process air stream.

The latent cooling capacity for the desiccant dehumidification system is calculated by subtracting the sensible cooling capacity from the total capacity, equation 7.

$$\dot{\mathbf{Q}}_{\mathbf{latent}} = \dot{\mathbf{Q}}_{\mathbf{Process(primary)}} - \delta\,\dot{\mathbf{Q}}_{\mathbf{sensible}} \qquad \mathbf{(7)}$$

A desiccant-based product can be operated in one of the following operating modes:

Operating Mode	**Source of Process Air**	**Source of Regeneration Air**
Ventilation/Exhaust	Outdoor Air	Indoor Air
Ventilation	Outdoor Air	Outdoor Air
Recirculation/Exhaust	Indoor Air	Indoor Air
Recirculation	Indoor Air	Outdoor Air

For each mode selected, the method of test provides the following performance parameter for the packaged dehumidifier:

Total cooling capacity, Btu/h
Sensible cooling capacity, Btu/h
Latent cooling capacity, Btu/h
Regeneration fuel input rate, Btu/h
Electric input rate, W
Water input rate, gpm (if product includes an evaporative cooling component)
Process airflow rate, cfm
Process leaving-air dry bulb temperature, ºF
Process leaving-air dew point or wet bulb temperature, ºF

The calculation of overall efficiency in terms of COP for the dehumidifier are not included in the method of test due to the problems associated with dealing with multiple energy inputs (fossil fuel and electricity). Also, this product must be combined with a conventional air conditioner to constitute a complete HVAC system design for most applications. Therefore, procedures are provided in an appendix to the method of test for comparing the performance of the dehumidifier used in conjunction with a unitary air conditioner to a base case. For example, a base case consisting of a unitary air conditioner with electric reheat that provides comparable performance. The comparison ratios that can be provided are:

Relative source fuel consumption
Relative energy cost
Relative capacity of the air conditioner required for comparable performance
Relative quantity of the supply air required.

Heat Balance Validation of Test Results

MOT validation results based on an assumption of heat balance between process and regeneration air paths are shown in Figure 4 for the laboratory and modeled test results given in Table 3. Please note that the 0.0140 and 0.0176 absolute humidity ratios used as abscissa values in this plot and in Figure 5 are the 95ºF (35ºC) dry bulb temperature, 75ºF (23.9ºC) wet bulb temperature and the 80ºF (26.7ºC) dry bulb, 75ºF (23.9ºC) wet bulb temperature conditions, respectively, chosen as process air and regeneration air inlet states for this study.

Using a value of $\pm 6\%$ as the limit for "valid" versus "invalid" results, Figure 4 shows that two of the laboratory data sets would pass the validation test criteria for acceptable, useable rating data. The two data sets that failed to qualify as "valid" include one where the regeneration heat transfer wheel, Figure 1, was activated and is turning at 10 rpm, and one in which the wheel was not activated. The authors were encouraged by these results, since the tests were not performed in climate control chambers as specified in the test procedure. Presumably, conducting the testing under more carefully controlled environmental conditions like those used for other ASHRAE and ARI testing and rating procedures would help correct the relatively small measurement variations that resulted in deviations of the magnitude shown in Figure 4.

Interestingly, one of the modeled desiccant system results would also fail to qualify as a "valid" set of results, if this verification criterion were used, Figure 4.

System Capacity Results

Sensible, latent, and total system capacity results calculated from this same set of data are summarized and displayed in Figure 5. These results clearly show that the total (sensible + latent) capacity of desiccant dehumidification systems can often be negative and how the system's heat recovery wheel dramatically improves overall system performance. Figure 5 also illustrates that, while one modeled and two laboratory data sets fail to meet the validity criterion set in the MOT, they do give system capacity results that are quite consistent and interpretable in terms of the general operating characteristics of desiccant dehumidification, i.e., enhanced latent capacity and diminished sensible capacity with increasing ambient humidity levels in the process air.

Capacity results plotted in Figure 5 re-enforce the benefits of a heat recovery heat exchanger on these systems that can help post-cool desiccant dehumidified air by transferring heat to the regeneration air stream where it productively pre-heats air needed for desiccant regeneration. In the example shown, this heat exchanger adds roughly 120,000 Btu/h (35 kW) to the total capacity of this system under these conditions.

The increase in system latent capacity evident in Figures 5a and 5b when going from 0 rpm to 10 rpm for the HX wheel raises an apparent contradiction. When the heat recovery (HX) wheel is turning at equivalent inlet conditions and the regeneration temperature and air flow rates are fixed, the HX wheel is replacing a small amount of dried air on the process side of the dehumidifier with relatively moist air from r2. This can be seen when comparing the actual dew points or absolute humidity ratios of the process leaving air stream, p3 – Figure 2, with and without the HX wheel turning. These values show that the process air stream contains more moisture per pound of dry air when the HX wheel is turning as compared to when the wheel is held at 0 rpm for both the experimental and modeled results in Table 3.

This apparent contradiction can be explained as a result of the manner in which and position at which process air flow rates are measured and controlled in this laboratory testing work. Equivalent 2,500 cfm flow rates for this set of experimental runs were obtained by matching the a calibrated pressure difference across the multi-point, self-averaging, Pitot-tube air flow measuring device by controlling blower speeds. When the HX wheel is in operation, the process air leaving temperature is much lower (≈ 50 – 60 °F [28 – 33 °C]) and its density is increased by approximately 9 – 10%. Because of the manner in which process air flow rate is measured, this results in a 9 – 10% increase in <u>mass</u> flow rate of process air (see relative density and "SCFM" values in Table 3). So, the increase in latent capacity shown in Figures 5a and 5b results from a corresponding increase in the <u>mass</u> flow rate of process air even though a 2,500 cfm (as opposed to SCFM) flow rate is being held constant at the process outlet. The extent to which each pound of air is dried is actually less when the HX wheel is operated at 10 rpm.

A similar set of assumptions was applied to the modeled results (an acfm measurement was assumed at the process outlet) so these data show a similar result in Figure 5b. The model predicts less moisture carryover by the HX wheel than measured experimentally because the difference between latent capacities at rotation rates of 0 and 10 rpm is higher for the modeled results.

Figure 6 shows these same sensible, latent, and total system capacity results as percentage deviations of the laboratory results from modeled results. These histogram plots also indicate that the 80ºF (26.7ºC) dry bulb temperature, 75ºF (23.9ºC) wet bulb temperature (0.0176 humidity ratio) condition with the heat recovery wheel turning at 10 rpm that did not meet the test data validation criterion, shows a large negative latent capacity discrepancy when compared to the modeled results, Figure 4. Interestingly, the 95ºF (35ºC) dry bulb temperature, 75ºF (23.9ºC) wet bulb temperature condition (0.0140 humidity ratio) with the heat recovery wheel deactivated (0 rpm), shows large positive latent, sensible, and total capacity deviations from modeled results in Figure 6, but is quite close to meeting the test validation criterion, Figure 4. This laboratory data set may be “invalid” by the criterion selected because the sensible capacity (temperature of the regeneration outlet steam) is too high.

Significant air leakage between the process and regeneration air streams would be expected to dramatically alter and invalidate system test results. Since cooling capacities are calculated by a psychrometric method, it is very critical that the air delivered to the conditioned space is measured and not the quantity that enters the process air side of the dehumidifier. Airflow rates for this work are measured at the process air and regeneration outlets. No attempt was made to measure leakage rates between process and regeneration air streams in these experiments.

A provision is offered in the draft MOT for cabinet heat loss adjustments if these are required to meet the validation of test data criterion. Cabinet heat loss represents the amount of heat loss by radiation and convection from the surface of the equipment cabinet to ambient. This loss may be determined from appropriate surface temperature readings in accordance with the procedures in Sub-section 8.6.1 of ANSI/ASHRAE 103-1993. (ASHRAE 1993) Cabinet heat loss corrections were not applied to these results.

Conclusions

Desiccant-based dehumidifier systems are growing in popularity because of their ability to independently control humidity levels (latent loads) in buildings, thereby allowing conventional air-conditioning systems to primarily control dry-bulb temperature (sensible loads). Properly applied, desiccant ventilation air pretreatment systems have the potential to reduce building energy consumption, decrease greenhouse CO_2 emissions to the atmosphere, and significantly improve the indoor air quality experienced by building occupants. Thermally regenerating desiccant dehumidification HVAC systems with waste heat from distributed power generation applications fits well with the recent emphasis on Combined Heating and Power (CHP) and/or Cooling, Heating, and Power (CHP) for Buildings approaches as a means to save energy, minimize environmental pollution, and improve utility reliability. Waste heat from power producing technologies like microturbines, fuel cells, or IC engines used in distributed generation applications is captured to regenerate desiccants which are employed to reduce the latent air conditioning load in buildings.

An industry-accepted method of test (MOT) and product certification rating system for desiccant-based products would allow consumer comparisons of products from different manufacturers, rate the relative performance of these systems to other more conventional products, and facilitate integration of desiccant-based options with conventional building heating, ventilating, and air conditioning (HVAC) equipment.

A standard method of test for desiccant-based dehumidifying systems has been proposed which is analogous to current procedures used for conventional packaged HVAC products.

A method for validation or rejection of experimentally obtained data based upon a calculated heat balance has been included, and this criterion appears both necessary and reasonable when applied to data from a highly instrumented laboratory facility.

Latent, sensible, and total system cooling capacity in Btu/h are the proposed system performance parameters generated by the MOT. This is the form that is most familiar and ultimately useful to HVAC designers and system engineers.

Experimental procedures, such as alternately operating and not operating the sensible heat transfer wheel in this system, can be employed to better assess the performance of individual system components despite the spatial non-uniformity of the air streams thermal properties inherent in large, cyclic, air circulating equipment like this.

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NOMENCLATURE

δ	"delta" or "change in"
ε	effectiveness
$\bar{c}_p$	average specific heat of moist air
h	specific enthalpy per unit mass of dry air
$\dot{m}$	mass flow rate
$\dot{Q}$	capacity (latent or net)
$\dot{Q}_{regeneration}$	Time rate of thermal energy transfer to regeneration air
T	dry bulb temperature

Subscripts

a	dry air
l or **latent**	pertaining to latent capacity
p or **process**	process air
r or **regeneration**	regeneration air
s or **sensible**	pertaining to sensible capacity
t or **total**	total capacity

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Table 1. System Measurements, Sensor Precision/Accuracy

Measurement	Sensor	Precision/Accuracy
Temperatures	- Averaging RTDs - Thermocouples	± 0.24 % at 70º F (21.1º C) ± 1º F (0.5º C)
Dew Points	Chilled Mirror	± 0.4º F (± 0.2º C)
Volumetric Air Flow Rates	Fan Evaluator multiple Pitot tube	± 2% (500 – 5000 cfm range) (14.2 – 141.6 m^3/min)
Power	Watt transducers	± 2.5 watts
Liquid Flow Rates (hot water desiccant regeneration source)	Turbine flow Meter	± 0.5% of reading (± 0.0075 gpm) (± 0.0005 l/sec.)

Table 2. Estimated Uncertainties in Calculations

Values	**Symbol**	**Estimated Uncertainty**
Process/Regeneration air mass flow rate	$\dot{m}_a$	± 3.8 lb/min (± 1.7 kg/min)
Latent Capacity	$\dot{Q}_{latent}$	± 3,100 Btu/h (± 0.9 kW)
Sensible Capacity	$\dot{Q}_{sensible}$	± 1,400 Btu/h (± 0.4 kW)
Total Capacity	$\dot{Q}_{total}$	± 4,500 Btu/h (± 1.3 kW)
Regeneration Heat input	$\dot{Q}_{regeneration}$	± 7,300 Btu/h (± 2.1 kW)
Overall Effectiveness of Regeneration and Desiccant Wheels	ε	± 5% (relative)

Table 3. Laboratory and Modeling Data Used to Check Desiccant System MOT

Laboratory Data	SCFM	Dry Bulb Temp (F)	Wet Bulb Temp (F)	Density (lb./cu. ft.da)	Abs. Enthalpy (Btu/lb.da)	Regeneration Heat Input (Btu/hour)
95 F dry bulb, 75 F wet bulb, 0 rpm - Heat Recovery Wheel						
Process Air In	2,120	95.1	75.2	0.06993	38.467	
Process Air Out (point of measurement)	2,120	157.2	83.6	0.06355	46.139	
Regeneration Air In	2,177	95.3	75.0	0.06992	38.297	
Regeneration Air Out (point of measurement)	2,177	130.2	88.1	0.06527	52.449	199,900
95 F dry bulb, 75 F wet bulb, 10 rpm - Heat Recovery Wheel						
Process Air In	2,312	95.3	74.9	0.06994	38.188	
Process Air Out (point of measurement)	2,312	105.7	70.7	0.06929	34.200	
Regeneration Air In	2,182	94.7	74.9	0.07000	38.149	
Regeneration Air Out (point of measurement)	2,182	128.0	88.2	0.06542	52.939	94,590
80 F dry bulb, 75 F wet bulb, 0 rpm - Heat Recovery Wheel						
Process Air In	2,117	79.9	74.8	0.07153	38.400	
Process Air Out (point of measurement)	2,117	156.0	85.4	0.06346	48.246	
Regeneration Air In	2,209	80.1	74.8	0.07151	38.383	
Regeneration Air Out (point of measurement)	2,209	118.7	88.6	0.06623	53.324	241,400
80 F dry bulb, 75 F wet bulb, 10 rpm - Heat Recovery Wheel						
Process Air In	2,351	80.2	75.0	0.07148	38.541	
Process Air Out (point of measurement)	2,351	93.7	71.0	0.07046	34.577	
Regeneration Air In	2,213	80.4	74.9	0.07147	38.548	
Regeneration Air Out (point of measurement)	2,213	117.9	88.4	0.06633	53.956	106,550
Modeling Data with Heat and Mass Balanced Regeneration Outlet Conditions						
95 F dry bulb, 75 F wet bulb, 0 rpm - Heat Recovery Wheel						
Process Air In	2,108	95.0	75.0	0.06996	38.272	
Process Air Out	2,108	160.0	84.7	0.06325	46.918	
Regeneration Air In	2,108	95.0	75.0	0.06996	38.272	
Adusted Regeration Air Out (Adjusted for heat and mass balance)	*2,108*	*127.2*	*89.3*	*0.06540*	*53.978*	*233,000*
95 F dry bulb, 75 F wet bulb, 10 rpm - Heat Recovery Wheel						
Process Air In	2,310	95.0	75.0	0.06996	38.272	
Process Air Out	2,310	108.0	71.2	0.06039	34.534	
Regeneration Air In	2,103	95.0	75.0	0.06996	38.272	
Regeration Air Out (Adjusted for heat and mass balance)	*2,103*	*128.1*	*89.4*	*0.06310*	*54.131*	*103,700*
80 F dry bulb, 75 F wet bulb, 0 rpm - Heat Recovery Wheel						
Process Air In	2,118	80.0	75.0	0.07150	38.547	
Process Air Out	2,118	155.0	85.4	0.06354	48.280	
Regeneration Air In	2,217	80.0	75.0	0.07150	38.547	
Regeration Air Out (Adjusted for heat and mass balance)	*2,217*	*113.0*	*90.5*	*0.06650*	*56.032*	*270,300*
80 F dry bulb, 75 F wet bulb, 10 rpm - Heat Recovery Wheel						
Process Air In	2,374	80.0	75.0	0.07150	38.547	
Process Air Out	2,374	95.0	70.2	0.07040	33.887	
Regeneration Air In	2,211	80.0	75.0	0.07150	38.547	
Regeration Air Out (Adjusted for heat and mass balance)	*2,211*	*114.9*	*90.5*	*0.06634*	*55.943*	*121,000*

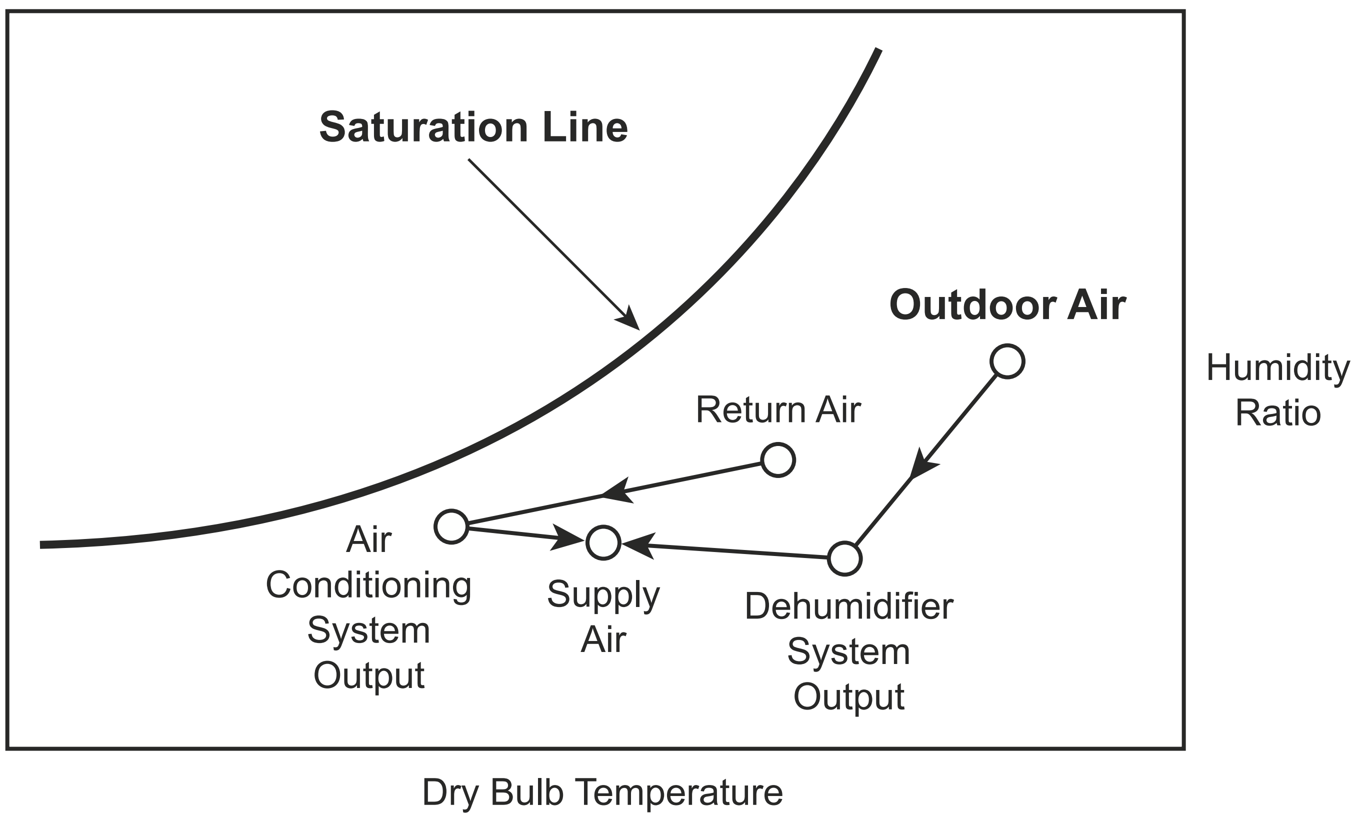

Figure 1. Mixing conditions showing relationships between outdoor, return, air conditioning outlet, dehumidifier system outlet (process), and supply air for a desiccant-based HVAC system.

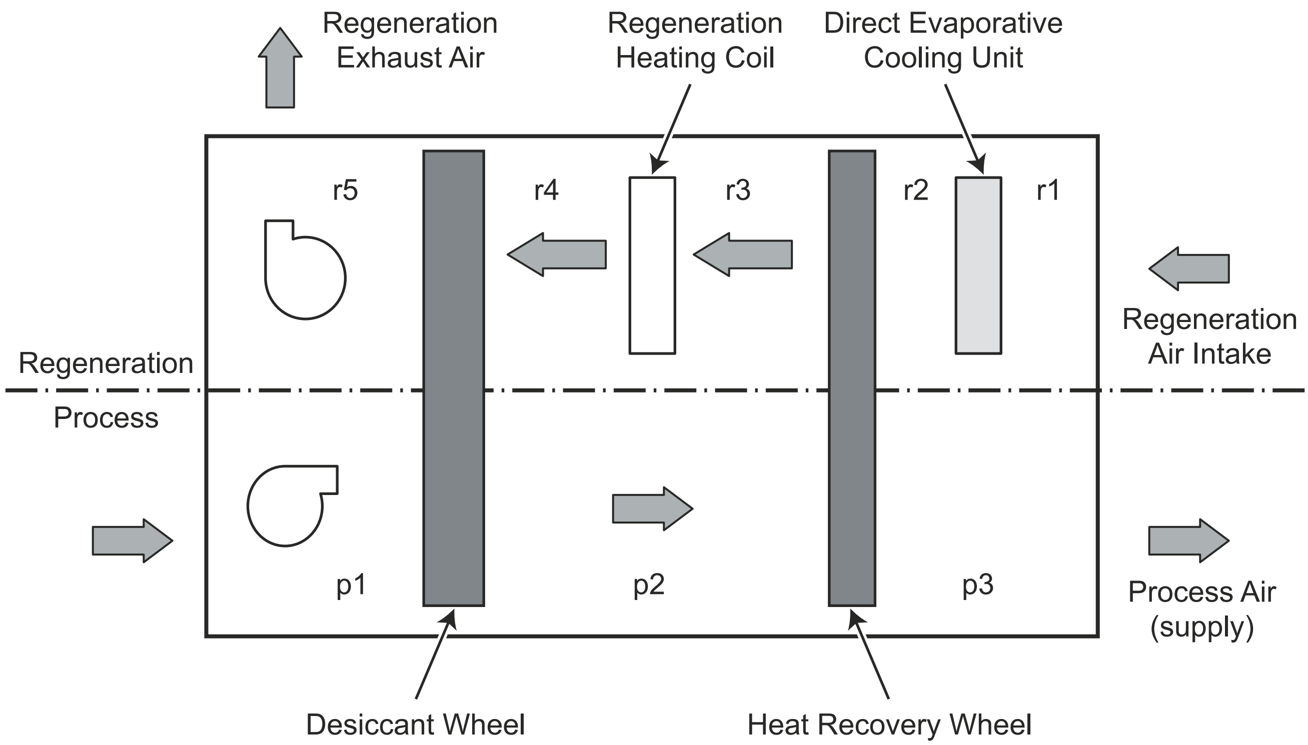

Figure 2. Schematic of desiccant system.

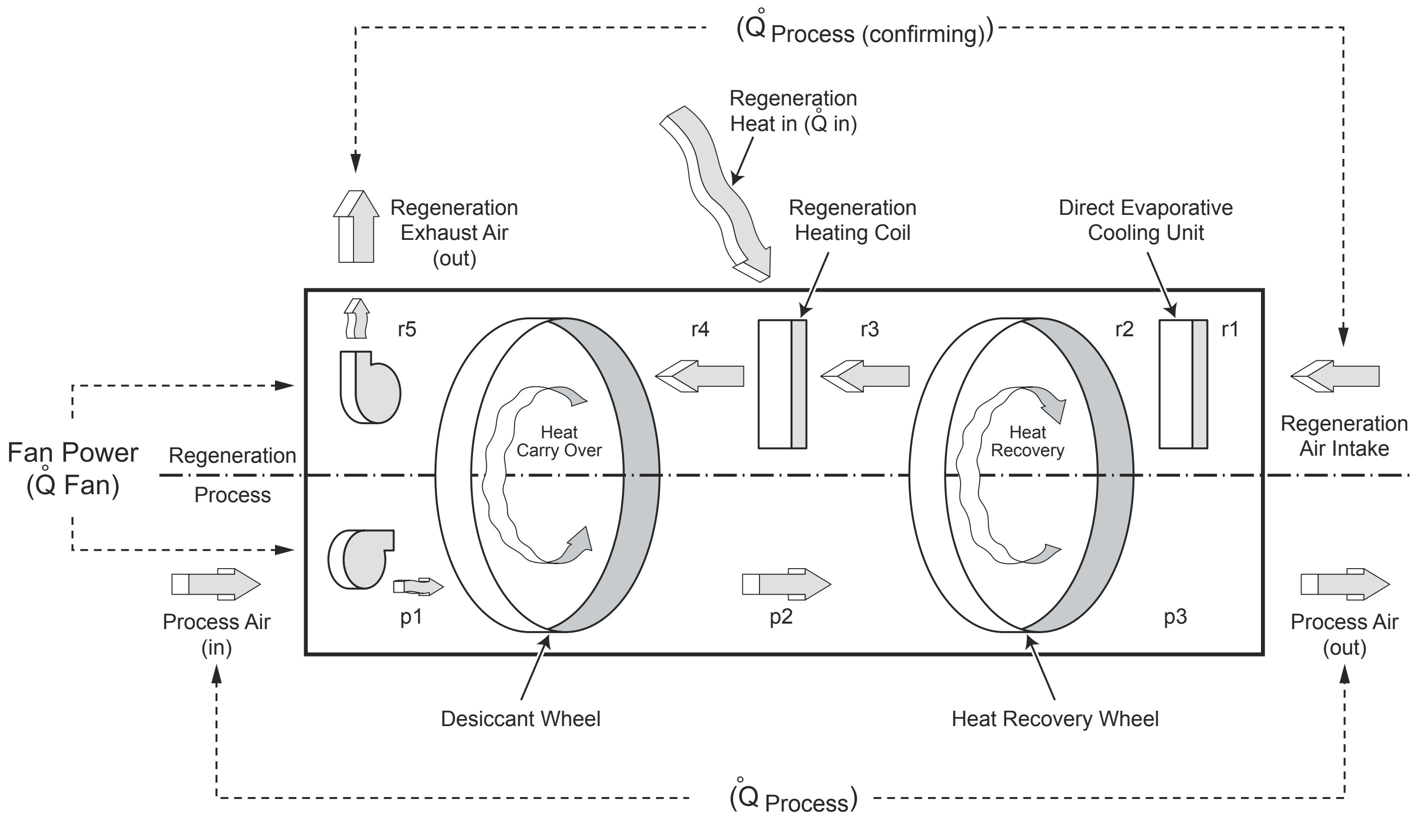

Figure 3. Enthalpy changes in a desiccant system.

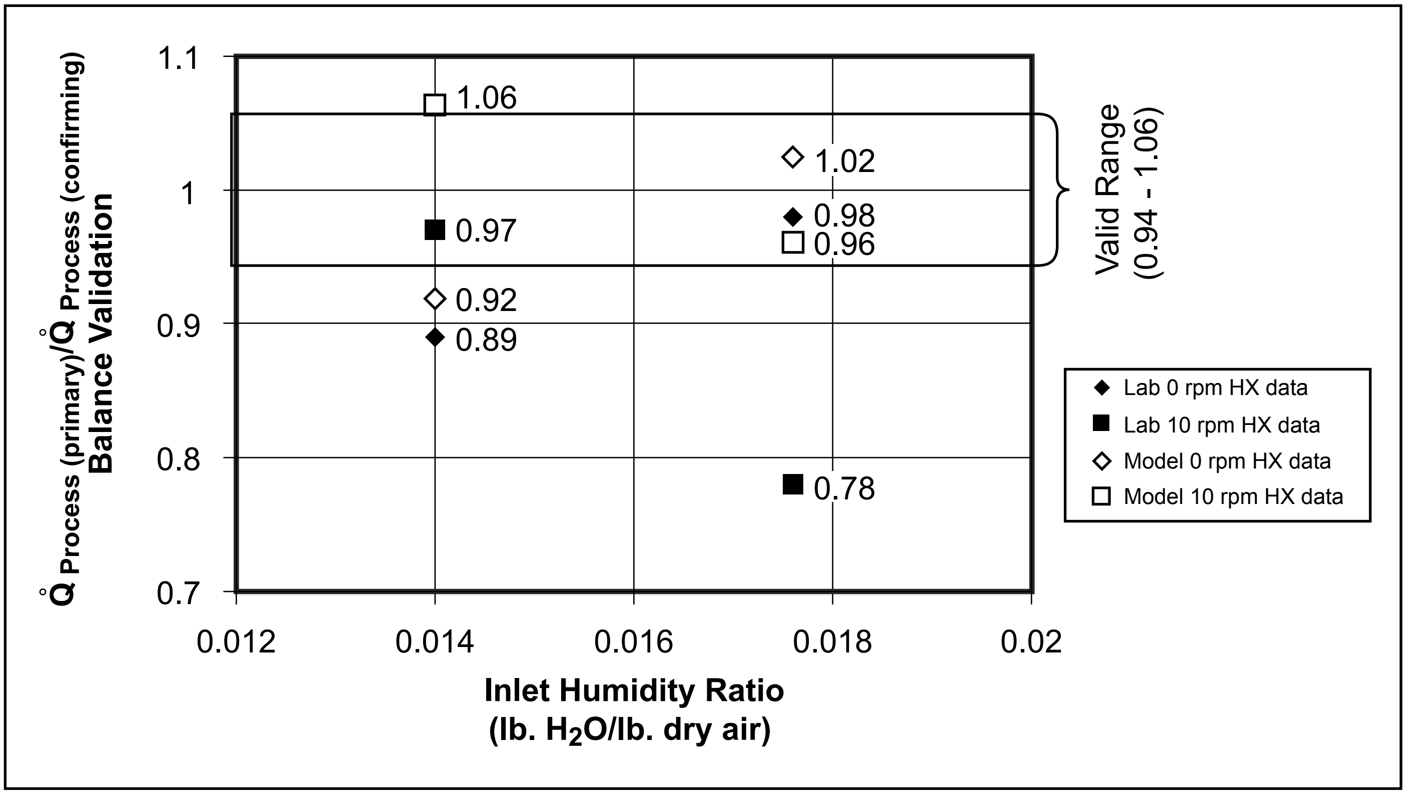

Figure 4. Method of test validation data.

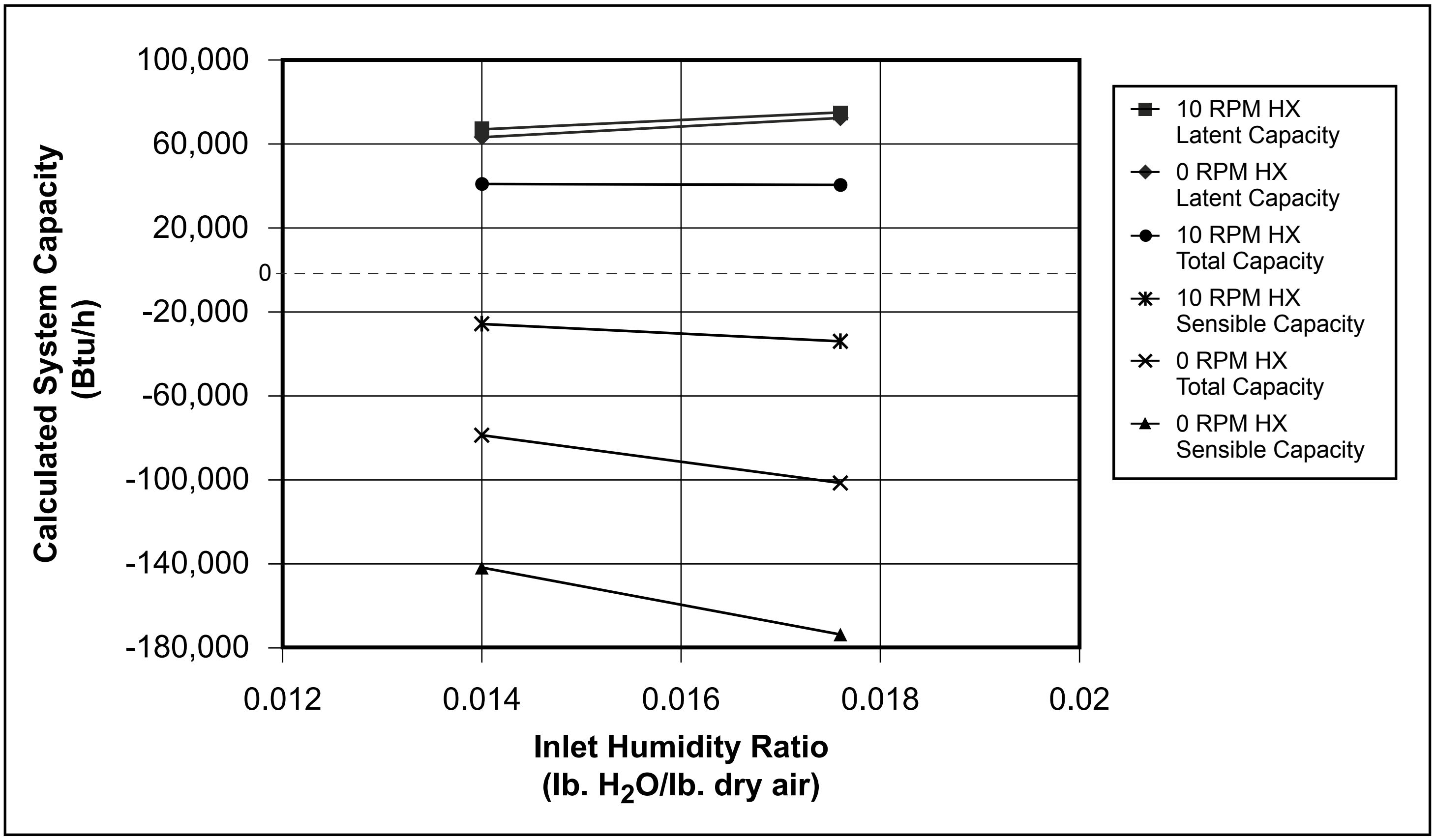

Figure 5a. Desiccant system capacity calculations - Laboratory results.

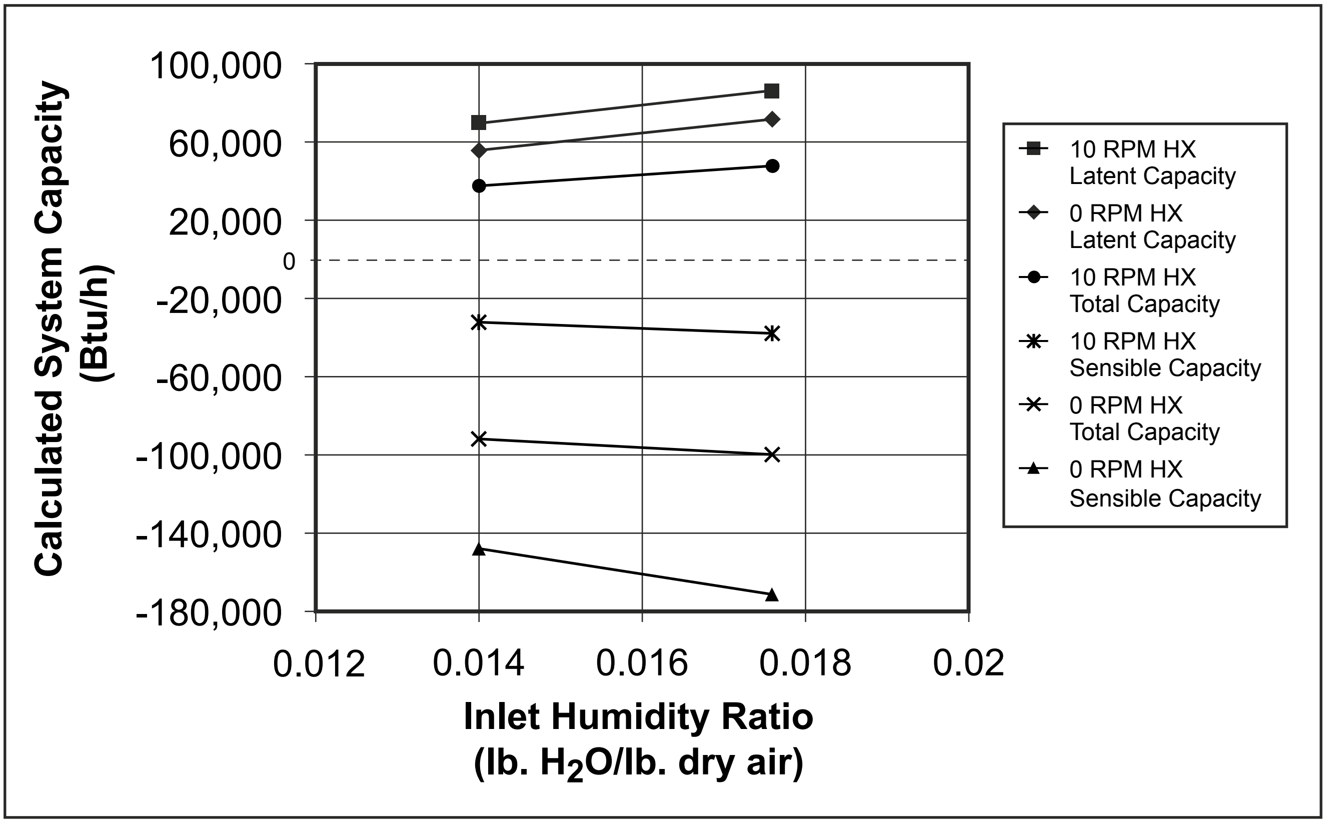

Figure 5b. Desiccant system capacity calculations - Modeled results.

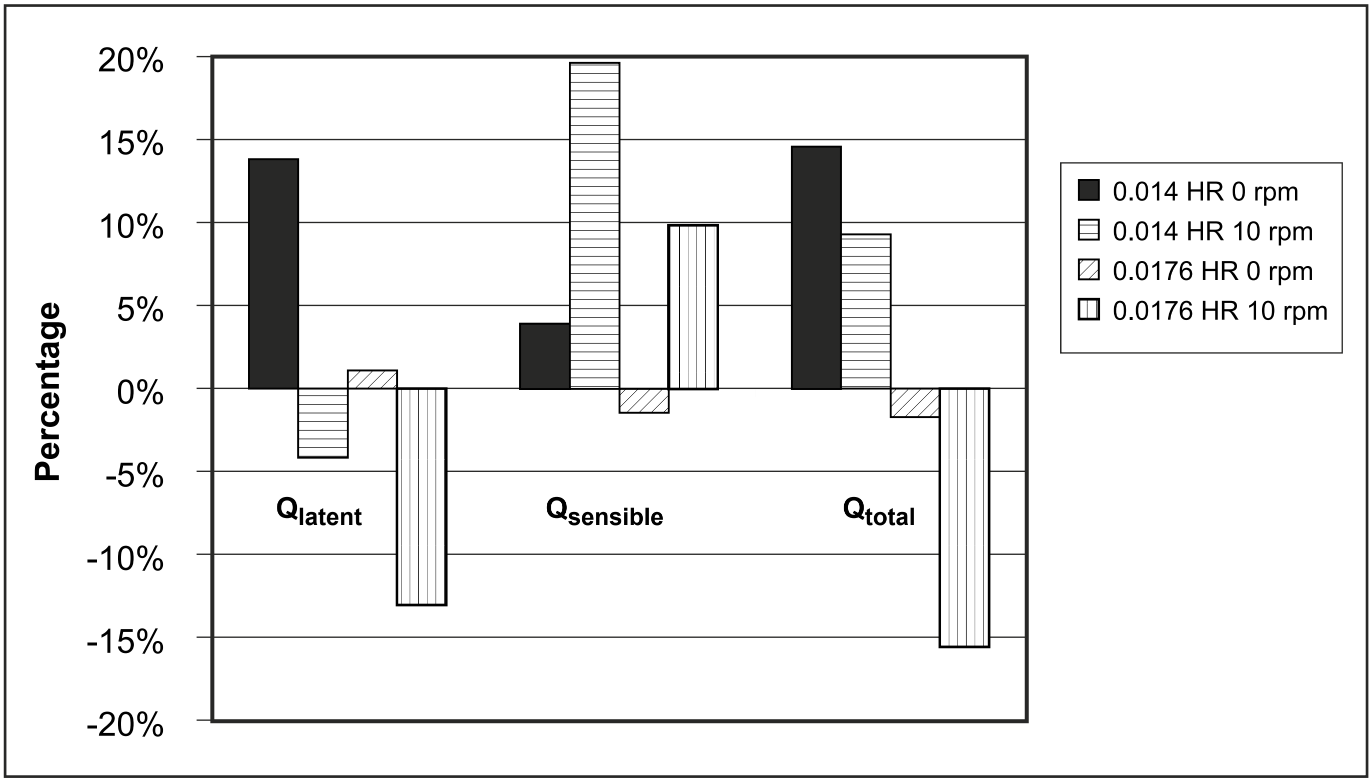

Figure 6. Deviation of laboratory from modeled capacity results.