Indoor Humidity Analysis of an Integrated Radiant Cooling and Desiccant Ventilation System

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Abstract: Radiant cooling is credited with improving energy efficiency and enhancing the comfort level as an alternative method of space cooling in mild and dry climates, according to recent research. Since radiant cooling panels lack the capability to remove latent heat, they normally are used in conjunction with an independent ventilation system, which is capable of decoupling the space sensible and latent loads. Condensation concerns limit the application of radiant cooling. This paper studies the dehumidification processes of solid desiccant systems and investigates the factors that affect the humidity levels of a radiantly cooled space. Hourly indoor humidity is simulated at eight different operating conditions in a radiantly cooled test-bed office. The simulation results show that infiltration and ventilation flow rates are the main factors affecting indoor humidity level and energy consumption in a radiantly cooled space with relatively constant occupancy. It is found that condensation is hard to control in a leaky office operated with the required ventilation rate. Slightly pressurizing the space is recommended for radiant cooling. The energy consumption simulation shows that a passive desiccant wheel can recover about 50% of the ventilation load.

1. INTRODUCTION

Radiant cooling panels cool the surrounding air by convection and cool objects within their direct view by radiation. In a radiant cooling system, the chilled water supply temperature can be increased by more than 10^{o}F compared with that required by the cooling coil in an air handler. This will significantly reduce chiller electricity consumption. The radiant cooling system can also save energy by cutting the supply fan power. Stetiu [1] simulated a radiant cooling system in a 700 square meter building and reported 30% energy savings compared with an all air system. Niu et al. [2] compared a chilled ceiling combined with a desiccant cooling system with a conventional constant volume all air system, and reported 44% primary energy savings in hot and humid climates such as Hong Kong.

However, condensation is a major problem that restricts the application of radiant cooling. Because radiant cooling systems lack the capability to remove moisture and ventilation is required, a radiant cooling system must be used in parallel with a dedicated outside air system. The dedicated outside air system can be a 100% outside air AHU or a desiccant wheel combined with a chilled water coil.

Several studies have examined the moisture condensation problem in radiantly cooled offices. Mumma [3][4][5][6] explored the condensation issues related to chilled ceilings combined with a dedicated outside air system. He studied the mechanisms of water formation on chilled panels when occupancy in the space exceeds the design, and necessary control measures. Zhang et al. [7] studied the indoor relative humidity behavior of all air systems with total heat recovery, chilled ceilings with an AHU, and chilled ceilings with desiccant cooling. They reported that a system combining a chilled ceiling with air dehumidification has more annual hours in the comfort region than with other ventilation system. They concluded that condensation can be avoided if the AHU ventilation unit begins operating one hour earlier than the chilled ceiling. One aspect that has received little attention from previous research is the impact of infiltration on condensation in a space where radiant cooling is integrated with a desiccant ventilation unit.

In the design of a radiant cooling system, the capacity of the ventilation system is decided by either space latent load or the indoor fresh air requirement, whichever is larger. When the latent load fluctuates widely such as in a meeting room, or a classroom, the design value of ventilation air flow is often very large to meet peak conditions. In these cases, the radiant cooling can only meet 10% to 30% of the space cooling load, which substantially reduces the energy savings from radiant cooling. How does the ventilation rate affect the indoor humidity and moisture condensation on chilled panels? This is another question has not received enough attention in previous studies.

This paper studies the hourly absolute humidity ratio in a radiantly cooled space with an integrated passive desiccant ventilation system. It illustrates the interaction between infiltration and mechanical ventilation rate on indoor humidity level, condensation and energy consumption of a radiantly cooled space. The possibility of condensation on the surface of radiant panels under different operation conditions, and the space load distribution between ventilation system and radiant cooling system have been investigated. Operating strategies to control condensation are recommended.

2. SIMULATION CASE STUDY OF A RADIANTLY COOLED SPACE

The space simulated is an advanced building technology testing site in Mid-Atlantic area. The test site is a small university office area, which includes faculty, graduate student and staff offices and a meeting room. The 580m^2 (6228 ft^2) space has a radiant heating and cooling system combined with a passive desiccant ventilation system. The space uses two types of radiant panels as shown in Figure 1. The first type is a radiant mullion system, which is installed vertically along the window frames. The mullion system is used to offset the heating and cooling load from the windows and increase the indoor comfort levels. Another function of the mullion system is that grouped mullions can provide flexible heating and cooling set points based on the preference of the occupants. The second radiant system is overhead ceiling panels, which are used for spaces away from the windows. These two types of radiant panels are used for cooling in summer and heating in winter. The chilled water and hot water are switched in the same piping system between summer and winter, in a two-pipe system.

The exterior walls of the space are metal with 4-inch insulation inside, providing an R-value of 20 ft^2-hr-^{o}F/Btu. The double pane windows account for 58.3% area of the exterior wall area. The open trussed sloped roof has the same thermal resistance as the metal wall. The roof includes 648 ft^2 of skylights, which have the same R-value as the windows. Moveable shades are installed on all the skylights. Because of the large window area, the lighting load in the space is relatively small, 0.9W/ft^2. Average equipment average load is 0.3 W/ft^2. 32 people are assumed as the normal maximum occupancy. A sensible load of 230 Btu/hr per person is assumed with a latent load of 0.13lb/hr-person.

The space is simulated by DOE2.1 software. The simulation model is carefully calibrated according to the procedures of Claridge et al. [8]. The calibrated simulation model matches the measured consumption data very well. Then the calibrated model is used to predict the system load at different infiltration and ventilation conditions.

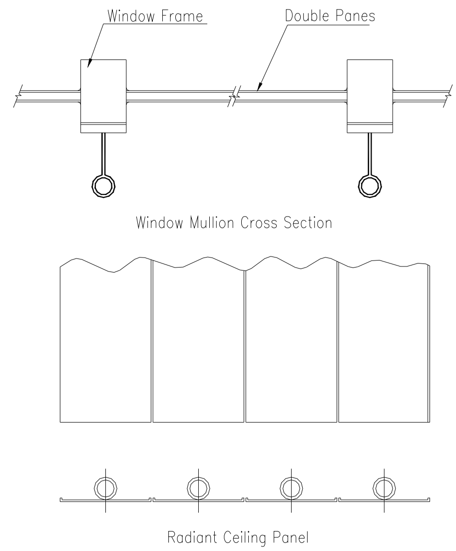

Fig.1 Two types of radiant panels in case study space

3. DESICCANT VENTILATION SYSTEM

To increase energy efficiency, a radiant cooling system is typically integrated with a solid desiccant ventilation system, which can be a passive system or an active system. The most commonly used desiccant systems are single wheel passive desiccant systems or dual wheel active desiccant systems. The desiccant wheel absorbs moisture from the fresh outside air and the wheel is regenerated with either hotter or dryer air. A passive desiccant wheel uses dry air, which is usually the building's exhaust air. An active desiccant wheel uses heated air produced by gas combustion or a heating coil. Active desiccant wheels can deeply dry the fresh outside air in all weather conditions regardless of the moisture content of the exhaust air. However, an active wheel requires heat input to dry the air, which increases the system energy consumption. A passive desiccant wheel cannot remove as much moisture as an active desiccant wheel. The moisture level of the supply air leaving the passive desiccant wheel depends on the dryness of exhaust air and its flow rate. Cooling is required after the passive wheel (henceforth called "post cooling") to remove additional moisture and maintain the humidity level sufficiently low inside the space when integrated with radiant cooling. Exhaust air reactivates the desiccant in a passive wheel adiabatically without additional heat input. The operating cost of a passive wheel is considerably lower than that of an active wheel [9].

3.1 Passive Desiccant System

Available commercial passive desiccant systems include single enthalpy wheel systems and duel wheel systems (enthalpy wheel plus sensible wheel). The enthalpy wheel removes both latent load and sensible load, while the sensible wheel removes only sensible load. The structure of these two wheels is similar. The key component is the "honeycomb like" transfer core, which utilizes an aluminum substrate. A commercial enthalpy wheel is normally coated with desiccant materials such as 3Å or 4Å molecular sieve or silica gel. The sensible wheel is a rotating heat exchanger without desiccant coating.

Commonly used desiccant materials in HVAC applications are silica gel and molecular sieves. Silica gel can absorb up to 40% of its own weight in water. A typical value for its specific microporic surface area is ~600 m^2/g (Babus'Haq et al.) [10]. The adsorption characteristics of silica gel function over a wide range of relative humidity. Molecular sieves are crystalline metal alumino-silicates (basically ceramic materials). The most commonly used molecular sieve for air dehumidification is known as type A zeolite. Zeolite can absorb water up to 20% of its own weight. Molecular sieves are porous crystals, with large specific surface areas and uniform pore sizes, and have a specific microporic surface area of ~700m^2/g (Babus'Haq et al.) [10]. A molecular sieve usually is used for low-temperature applications. A special property of molecular sieves is their ability to "selectively adsorb" materials based on their kinetic diameter, pulling in materials smaller than the size of their pore openings while excluding materials that are larger. This property can help to reduce the contaminants carried over from exhaust air to supply air.

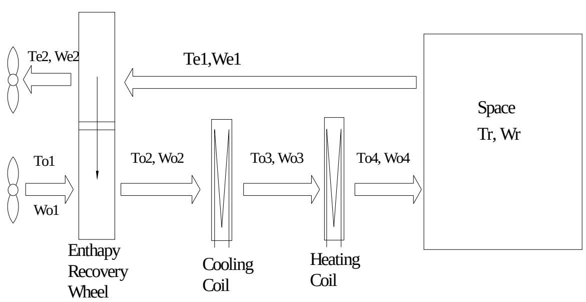

Fig.2. Passive desiccant system

Enthalpy wheels normally use an aluminum substrate coated with a molecular sieve material or silica gel. The effectiveness of an enthalpy wheel depends on the load of desiccant materials, the diameter and depth of the wheel, face flow velocity, rotational speed and other operating conditions. Bulk et al. [11] proposed $\varepsilon - NTU$ correlations for design calculation of latent and total effectiveness of enthalpy wheels coated with silica gel. Simonson et al. [12][13] developed more accurate complex correlations for the sensible, latent and total effectiveness of enthalpy wheels. Their model works well on balanced flow silica gel and molecular sieve enthalpy wheels. In 2000, Simonson et al. [14] modified the above correlations to make them apply for unbalanced flow. Freund et al. [15] developed a simple and generalized method to predict enthalpy wheel performance based on the classical $\varepsilon - NTU$ approach. Jeong and Mumma [16] proposed a group of correlations to calculate the sensible, latent and total effectiveness of enthalpy wheels at non-standard conditions based on statistical methods.

At design rotational speed and face velocity, latent heat transfer effectiveness, ε_L , and sensible heat transfer effectiveness, ε_S , can be found in manufacturers' manuals. The parameters of supply and exhaust air can be calculated according to the following equations based on energy and mass balance, once when the effectiveness and inlet conditions on both sides of the wheel are known.

$$T_{o2} = T_{o1} - \varepsilon_s \frac{(\dot{m}C_p)_{\min}}{(\dot{m}C_p)_s}(T_{o1} - T_{e1}) \tag{1}$$

$$w_{o2} = w_{o1} - \varepsilon_l \frac{\dot{m}_{\min}}{\dot{m}_o}(w_{o1} - w_{e1}) \tag{2}$$

$$T_{e2} = T_{e1} - \varepsilon_s \frac{(\dot{m}C_p)_{\min}}{(\dot{m}C_p)_e}(T_{o1} - T_{e1}) \tag{3}$$

$$w_{e2} = w_{e1} - \varepsilon_l \frac{\dot{m}_{\min}}{\dot{m}_e}(w_{o1} - w_{e1}) \tag{4}$$

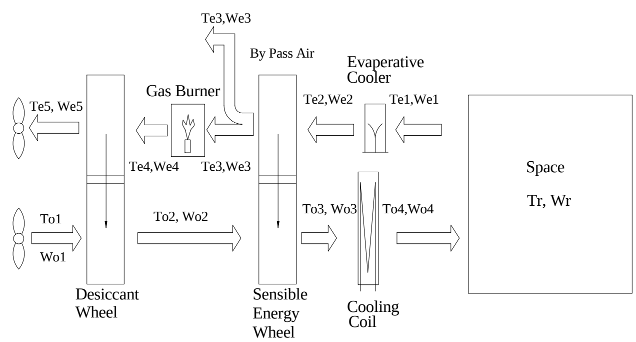

Fig.3. Active desiccant system

3.2 Active Desiccant System

An active desiccant wheel is made up of a fiberglass, paper, or sometimes aluminum substrate coated with silica gel. The most common active ventilation system is shown schematically in Figure 3. It is a combination of one desiccant wheel and one sensible energy wheel. The regeneration air can be exhaust air or outside air. The supplied outside air first passes through the desiccant wheel where the outside air (OA) is dried and the temperature increases. Then OA passes through the sensible wheel where it is cooled. Finally, the OA is cooled further by the cooling coil and its temperature is adjusted to the required temperature. The exhaust air first passes through the evaporative cooler and the sensible energy wheel to cool the supply air. After passing through the sensible wheel, part of the exhaust air is heated by the heating coil or a natural gas burner to 150°F-225 °F and used to regenerate the desiccant wheel. The other part of the exhaust air is discharged to ambient. In a typical configuration, 75% of the desiccant wheel face area is in the fresh outside air path while the remaining 25% is in the regeneration air path.

The effectiveness of an active desiccant wheel depends on structural parameters and operating conditions such as the depth of the wheel, the type and quantity of the desiccant, the surface area of the honeycomb, and the temperature and humidity ratio of the outside air and regeneration air, wheel rotational speed, face flow velocity, etc. Adjusting the regeneration temperature is the most approach commonly used by commercial manufacturers to change the wheel's moisture removal capacity. The higher the regeneration air temperature, the more moisture is removed by the desiccant wheel. When moisture is removed from the desiccant wheel, the latent heat of the moisture is converted to sensible heat. About 80% to 90% [9] of the temperature rise of the outside air comes from the conversion of latent heat, while the remainder is the sensible heat carried over by the wheel. Jurinak [17] developed the following model to evaluate the effectiveness of a silica gel active desiccant wheel by curve fits to the derived wave front propagation characteristics.

$$F1 = \frac{-2865}{T^{1.49}} + 4.344 w^{0.8624} \quad (5)$$

$$F2 = \frac{T^{1.49}}{6360} - 1.127 w^{0.07969} \quad (6)$$

$$\varepsilon_{F1} = \frac{F1_{o,2} - F1_{o,1}}{F1_{r,1} - F1_{o,1}} \quad (7)$$

$$\varepsilon_{F2} = \frac{F2_{o,2} - F2_{o,1}}{F2_{r,1} - F2_{o,1}} \quad (8)$$

F1 and F2 correspond to isopotential lines of enthalpy and relative humidity. ε_{F1} and ε_{F2} are the effectiveness of total energy and moisture removal at optimum rotary speeds. The subscripts "o" and "r" mean OA and regeneration air respectively. The subscripts "1" and "2" mean inlet and outlet. T is temperature in K and w is humidity ratio in kg (moisture)/kg (dry air). The outlet temperature and humidity of OA and regeneration air can be found iteratively by using this model.

Because of the temperature increase when the OA passes through the desiccant wheel, the sensible energy wheel is integrated into the system to cool down the outside air and increase the energy efficiency. The amount of heat removed from outside air depends on the temperature on the other side of the heat exchanger. The maximum efficiency can be obtained when the system takes exhaust air from the space and cools it by evaporative cooling. When exhaust air is not available, outside air can be used and cooled by an evaporative cooler; then passed through the sensible wheel to cool down the fresh outside air as shown in Figure 3.

The advantage of the active desiccant system is that this system can dry the outside air continuously and deeply in all weather conditions regardless of the moisture content of the exhaust air. The desiccant wheel can be regenerated with either exhaust air or outside air, which provides installation flexibility for places where exhaust air is not available.

4. TRANSIENT MODEL OF DEHUMIDIFICATION

When the dew point of indoor air is higher than the chilled panel surface temperature, water starts to condense on the surface of cooling panels. The dew point of indoor air is decided by the moisture balance among indoor latent heat, infiltration moisture and mechanical ventilation moisture. The transient study is necessary to decide the relationship among these parameters and to determine how much earlier the dehumidification unit needs to be started before the chilled panels are operated to avoid condensation. The steady model used in the latter section of the condensation study is derived from the transient model.

4.1 Dehumidification of Passive Desiccant System

If we take a space as shown in Figure 2 as a control space, the moisture balance inside the space can be described by the following equation:

$$V_r \rho_r \frac{dw_r}{dt} = \dot{V}_s \rho_s (w_s - w_r) + ach_i V_r \rho_o (w_o - w_r) + \dot{m}_{gen} \quad (9)$$

w_r, w_o, w_s are room humidity ratio, outside air humidity ratio and supply air humidity ratio respectively. For the passive desiccant system in Figure 2, the parameters of the supply air after the desiccant wheel can be calculated by changing equations (1)-(4) as follows:

$$T_{o2} = T_{o1} - \varepsilon_s \frac{(C_p \dot{m})_{\min}}{(C_p \dot{m})_s} (T_{o1} - T_{e1}) = T_{o1} - \varepsilon_s \beta (T_{o1} - T_r) \quad (10)$$

$$w_{o2} = w_{o1} - \varepsilon_l \frac{\dot{m}_{\min}}{\dot{m}_s} (w_{o1} - w_{e1}) = w_{o1} - \varepsilon_l \beta (w_{o1} - w_r) \quad (11)$$

$$\beta \approx \frac{\dot{m}_{\min}}{\dot{m}_s} \approx \frac{V_{\min}}{V_s} \quad (12)$$

The subscript "min" in equations (10) – (12) means the smaller of supply air flow rate and exhaust air flow rate. The subscript "s" indicates supply air while β is the ratio of the smaller flow rate to the larger flow rate. In a balanced system, the volume flow rate of supply air is equal to the exhaust air, and $\beta = 1$.

Case 1. No post cooling used

In this case, $w_s = w_{02}$. Then substitute equation (11) into equation (9), to obtain the following equation.

$$\frac{dw_r}{dt} = -w_r \frac{1}{V_r \rho_r}\left(\dot{V}_s \rho_s (1-\varepsilon\beta) + ach_i V_r \rho_o\right) + \frac{1}{V_r \rho_r}\left(ach_i V_r \rho_o w_o + \dot{V}_s \rho_s w_0 (1-\varepsilon\beta) + \dot{m}_{gen}\right) \tag{13}$$

Considering the initial conditions, $w_r|_{t=0} = w_{r0} = w_o$, the above equation can be solved as

$$w_r = (w_{r0} - \frac{b}{a})e^{-at} + \frac{b}{a} \tag{14}$$

In equation (14),

$$a = \frac{1}{V_r \rho_r}\left(\dot{V}_s \rho_s (1-\varepsilon\beta) + ach_i V_r \rho_o\right) \tag{15a}$$

$$b = \frac{1}{V_r \rho_r}\left(ach_i V_r \rho_o w_o + \dot{V}_s \rho_s w_0 (1-\varepsilon\beta) + \dot{m}_{gen}\right) \tag{15b}$$

In the equilibrium state, $w_r = \dfrac{b}{a}$ (16)

Case 2. Post cooling is used with, the supply air condition: 55°F, 0.0092lb/lb

Equation (9) can be written as

$$\frac{V_r \rho_s dw_r}{dt} = -\dot{V}_s \rho_s (0.0092 - w_r) + ach_i V_r \rho_o [w_o - w_r] + \dot{m}_{gen} \tag{17}$$

Solving the above equation, we can obtain an equation similar in format to Equation (14):with

$$a = \frac{1}{V_r \rho_r}\left(\dot{V}_s \rho_s + ach_i V_r \rho_o\right) \tag{18a}$$

$$b = \frac{1}{V_r \rho_r}\left(ach_i V_r \rho_o w_o + 0.0092 \dot{V}_s \rho_s + \dot{m}_{gen}\right) \tag{18b}$$

At the equilibrium state, $w_r = \dfrac{b}{a}$ (19)

Using equations (14) (15) and (18), the transient processes of dehumidification with and without post cooling can be plotted as shown in Figure 4 and Figure 5, when the outside air condition is 61°F and $w_o = 0.0107$ lb/lb. This transient process assumes the initial humidity ratio in the space equals the outside air humidity w_o and 25 people are using the space when the ventilation system starts. 0.13lb/hr per person is assumed for moisture generation. An infiltration rate of 0.45 air changes per hour is assumed. This value is based on the calibrated simulation model and site measurements.

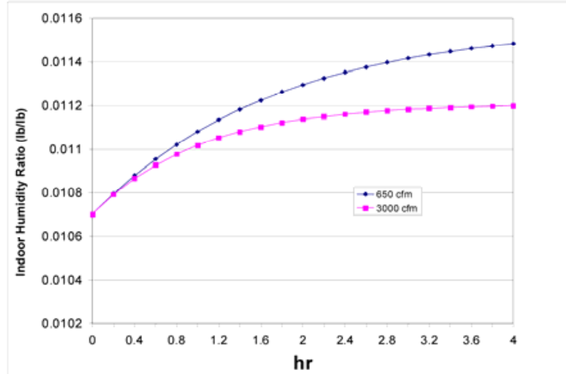

Fig.4 Transient behavior of indoor humidity for a passive desiccant system with No post cooling.

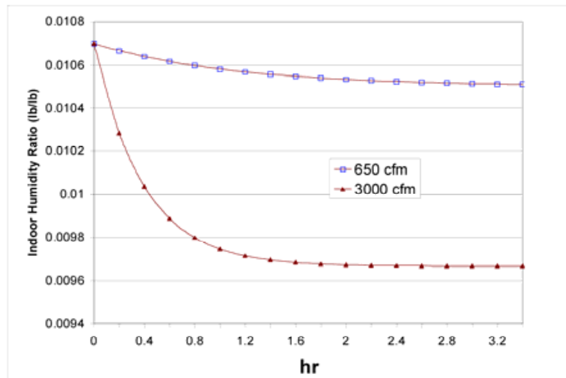

Fig. 5. Transient behavior of indoor humidity for a passive desiccant system with post cooling.

Figure 4 shows that the passive desiccant wheel actually adds moisture to the space instead of removing moisture from the space when the cooling coil is turned off. This means the dehumidifying function of a passive desiccant wheel depends on the dryness of the exhaust air. For a balanced flow passive desiccant system, the higher the ventilation rate; the lower the indoor humidity level. When the indoor humidity level reaches equilibrium, the inside humidity level is higher than the outside level. The desiccant wheel actually absorbs moisture from the exhaust air and releases it to the supply air when the humidity ratio of the exhaust air is higher than the supply air. This is the reason that the passive desiccant wheel does not lower space humidity or even increases humidity levels when post cooling is not available, and there are no other dehumidification sources.

Figure 5 shows that the high ventilation rate dries the space very quickly when post cooling is on (the supply air is cooled to 55°F, 0.0092lb/lb). However, it also creates another problem. To cool or heat a large volume of outside air will consume more energy. Figures 4 and 5 indicate that the moisture removing ability of a passive desiccant wheel requires the presence of post cooling.

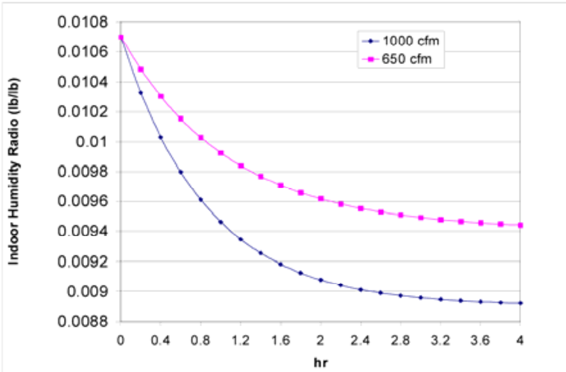

Fig.6. Transient behavior of indoor humidity for an active desiccant system

4.2 Dehumidification with an Active Desiccant System

The active desiccant system of Figure 3 can dry a space more effectively than a passive system, because the humidity ratio after the desiccant wheel can be set to a relatively low level by adjusting the regeneration air temperature to a high value. The transient behavior of an active desiccant system is the same as that of a passive desiccant unit with post cooling as analyzed in the previous section. When the humidity ratio after the desiccant wheel is set to 0.007 lb/lb (dew point 48°F), the space humidity decreases as shown in Figure 6. The humidity ratio of the supply air is decided by the regeneration air temperature and humidity. The active desiccant system can reduce the supply air humidity ratio to a lower level than a passive desiccant system. The reason is that the chilled water temperature at the inlet of the post cooling coil in passive desiccant units depends on the operating conditions of the chiller or DX coil. Normally this temperature can not be lower 40°F in a university campus loop. Figure 7 shows the relationship between humidity ratio after the wheel and regeneration air temperature based on Jurinak's model [17] (Equations (5)-(8)). Desiccant wheel inlet air conditions of 61°F, 0.0107lb/lb; $\varepsilon_{F1} = 0.3$, $\varepsilon_{F2} = 0.85$ are assumed in Figure 7. w_{r1} in figure 7 is the regeneration air humidity ratio. It can be seen that the ideal supply air humidity ratio of the air leaving the desiccant wheel has a nearly linear relationship with the regeneration air temperature.

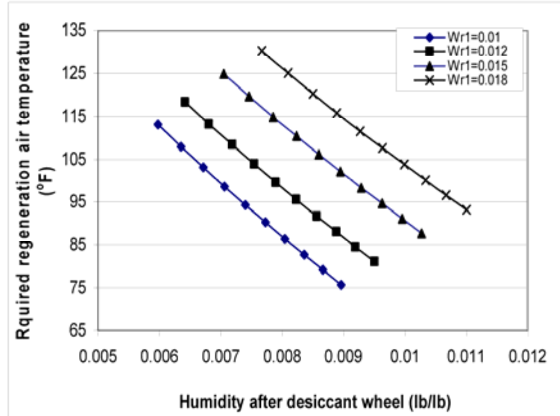

Fig.7 Relationship between humidity ratio and regeneration air temperature

5. CONDENSATION AND ENERGY CONSUMPTION ANALYSIS

Condensation is often a major problem when applying the radiant cooling system. An indoor humidity ratio higher than the saturation humidity ratio at the radiant panel surface temperature will cause water to condense on the surface of the radiant cooling panels which results in shutting down of cooling panels by the control system and overheating of the space. To avoid condensation, the dew point of the indoor air must be below the surface temperature of the radiant cooling panels. The normal design condition for indoor air is 75°F and 50% relative humidity ratio which corresponds to a dew point of 55°F and an absolute humidity ratio of 0.0092 lb (water)/lb (dry air). ASHRAE Standard 55-2004 [18] recommends an upper limit for indoor humidity of 0.012 lb/lb which corresponds to a dew point of 62°F. To ensure there is no condensation, the design surface temperature of a radiant panel is normally 1-3°F higher than the dew point of the indoor air. Therefore the inlet chilled water temperature of radiant panels is often set to 62-65°F. Increasing the panel surface temperature will increase the safety, but will decrease the cooling capacity. To avoid water condensation, the indoor humidity ratio needs to be controlled below 0.012 lb/lb, which corresponds to a dew point of 62 °F.

Radiant heating and cooling of a space is typically integrated with a ventilation system that provides humidity control. The ventilation system can be a 100% outside air handling unit, a simple passive desiccant system as shown in Figure 2 or an active desiccant system as shown in Figure 3. An air-handling unit is easier to control, but is less efficient. An active desiccant system usually is considered when the dew point of the supply air is required to be below 45°F (humidity ratio below 0.0063lb/lb) according to Gatley [19]. An active desiccant system often has a life cycle cost advantage when the dew point temperature of ventilation air is required to be below 40°F. Radiant cooling integrated with a passive desiccant ventilation system has been shown to be a cost effective way to maintain a healthy and comfortable indoor air environment [20]. Several factors in this type of system have important impact on indoor humidity ratio. These include infiltration ratio, outside air flow rate, and outside humidity ratio.

The tested radiantly heated and cooled space uses a passive desiccant system as shown in Figure 2. Chilled water is available only from June to September each year. Consequently, radiant cooling with post cooling of the passive desiccant wheel is only used from June to September. The infiltration rate has been estimated to average 0.45 air changes per hour on a yearly basis as noted earlier. Condensation occurs on the surface of the mullion system in summer. Hourly indoor humidity has been simulated under eight different conditions by using the model of equations (9) to (19). These eight conditions are shown in Table 1. Figures 8-11 show the simulated indoor humidity ratio over a one year period. The occupancy is assumed to be 25 from 9:00am to 8:00pm. The corresponding moisture generation in the space is about 3.25lb/hr. The ventilation system is assumed to run continuously in order to clearly show the humidity trend in an hourly time series over the year. Equilibrium conditions are assumed in the hourly simulation.

Tab.1 Simulation conditions

	Infiltration (ACH)	Supply (CFM)	Supply Air Humidity Ratio (lb/lb)	Return (CFM)
Case 1	0.001	650	0.0092	650
Case 2	0.450	650	0.0092	650
Case 3	0.001	1600	0.0092	1600
Case 4	0.450	1600	0.0092	1600
Case 5	0.001	650	0.008	650
Case 6	0.450	650	0.008	650
Case 7	0.000	650	0.0092	0
Case 8	0.000	850	0.0092	650

When the ventilation air flow rate is equal to the exhaust air flow, the space pressure is neutral. The infiltration rate has a great impact on the indoor humidity ratio. Figures 8a (case 1) and 8b (case 2) indicate when OA is 650 CFM and the supply air humidity ratio is 0.0092 lb/lb (dew point of 55°F), the indoor humidity level can be controlled below 0.011lb/lb over the whole summer in a tight building (ACH equals 0.001). In a leaky condition (ACH equals 0.45), a significant number of hours have a humidity level higher than 0.012 lb/lb (dew point of 62°F) during the summer. During these periods, water will condense on the surface of the radiant cooling panels when the panel surface temperature is 62°F or lower. Condensation has been observed at times during the summer. Another important trend to be noted is that the indoor humidity level during some hours in April, May and October is much higher than 0.012 in a tight building, because the post cooling is turned off. High humidity may cause indoor comfort problems during these periods. The tested space has operable windows, so this problem is not as serious as shown in the graphs.

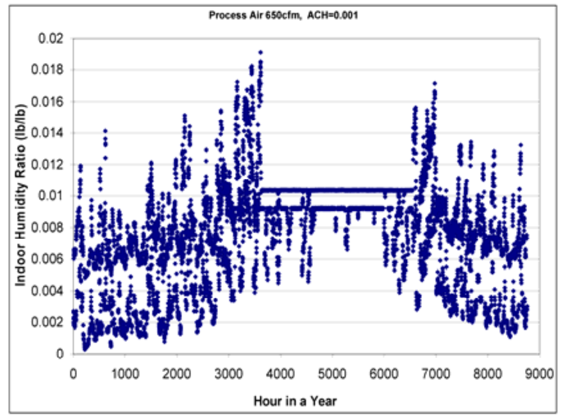

Fig.8a.Case1.OA: 650CFM, ACH:0.001, Supply air humidity ratio: 0.0092 lb/lb.

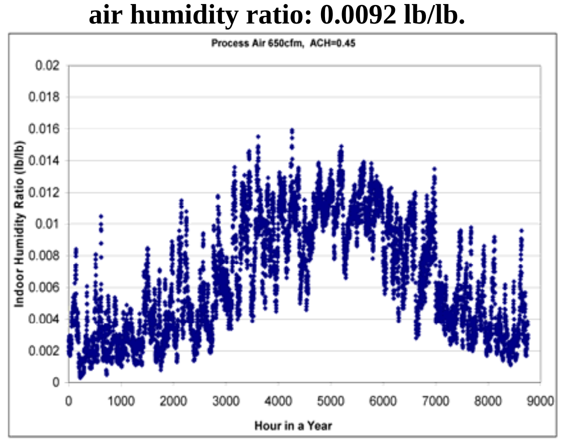

Fig.8b.Case2. OA: 650CFM, ACH:0.45, Supply air humidity ratio: 0.0092 lb/lb.

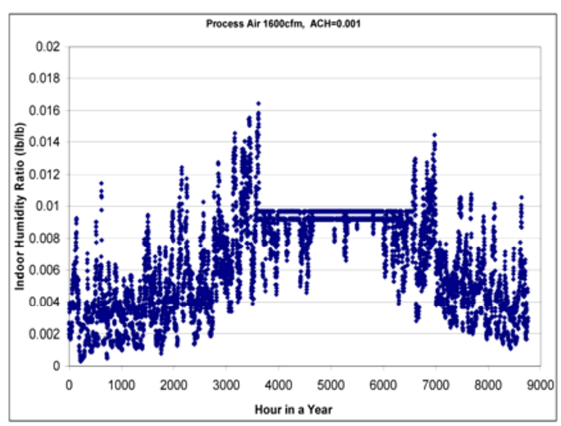

Fig.9a.Case3.OA: 1600CFM, ACH:0.001, Supply air humidity ratio: 0.0092 lb/lb.

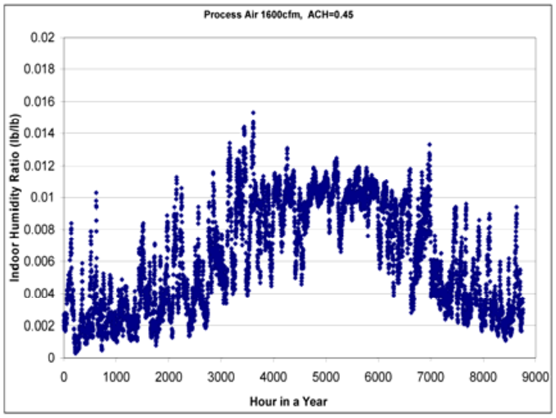

Fig.9b.Case 4. OA: 1600CFM, ACH:0.45, Supply air humidity ratio: 0.0092 lb/lb.

In order to reduce the indoor humidity level during summer, two approaches can be taken. One is supplying more dried outside air to the space as shown in case 3 and case 4. The other is to further reduce the humidity ratio of supply air as in case 5 and case 6. Indoor humidity ratios for an increased ventilation rate are shown in Figures 9a and 9b. 1600 CFM is the potential maximum outside air requirement. If the desiccant ventilation unit runs at 1600 CFM with an infiltration rate of 0.001 ACH, the summer indoor humidity ratio can be controlled under 0.01lb/lb (dew point of 60^{o}F). At the current leakage level of 0.45ACH, the humidity ratio can also be controlled under 0.012lb/lb (dew point of 62^{o}F) during most summer hours at the ventilation rate used in Figure 9b. However, energy consumption needs to be considered. The higher ventilation rate will increase the energy consumption in conditioning the outside air, even though the heat recovery by the enthalpy wheel also increases.

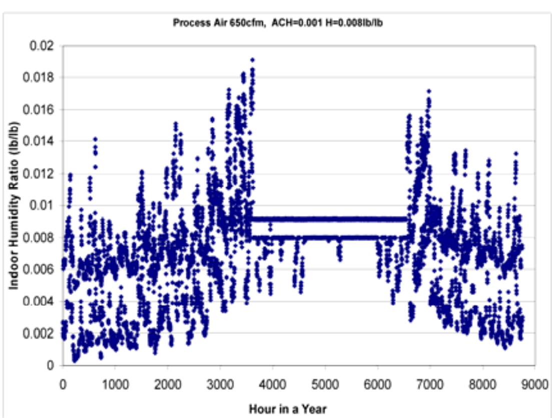

Fig.10a.Case5. OA: 650CFM, ACH:0.001, Supply air humidity ratio: 0.008 lb/lb

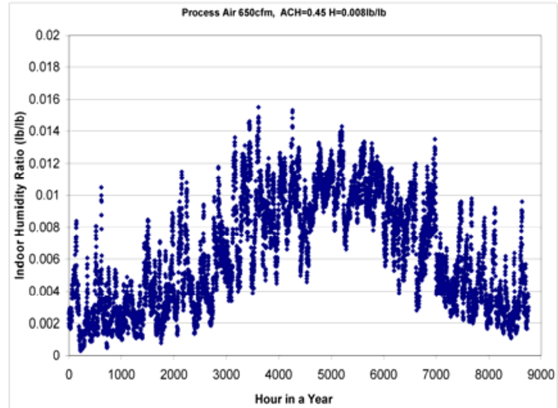

Fig.10b.Case6. A: 650CFM, ACH:0.45, Supply air humidity ratio: 0.008 lb/lb.

Another option for reducing indoor humidity level in summer is to reduce supply air humidity ratio as shown in Figures 10a (case 5) and 10b (case 6). The humidity ratio of the supplied ventilation air is set to 0.008 lb/lb (dew point of 52^{o}F). Comparing Figure 10a with Figure 9a, it can be seen that the indoor humidity level in summer is even lower at an OA rate of 650 CFM with a humidity ratio of 0.008lb/lb than at 1600 CFM with a humidity ratio of 0.0092 lb/lb in a tight building. Figure 10b illustrates that the condition of case 6 can not effectively control the humidity level below 0.012 lb/lb during the summer season. Moisture condensation on cooling panels can not be avoided under these conditions. Figures 10a shows that the humidity level in the tight space of case 5 is frequently higher than that in the leakier space of Case 6 (Figure 10b) during the winter, spring and fall when post cooling is not available. The reason for the difference is that when a building is leaky, the drier outside air removes indoor moisture in winter, spring and fall in a dry climate.

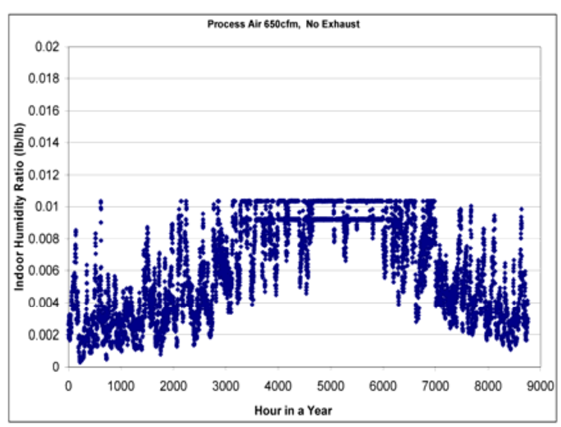

Fig.11a.Case 7. OA: 650CFM, No Exhaust air, ACH:0, Supply air humidity ratio: 0.0092 lb/lb

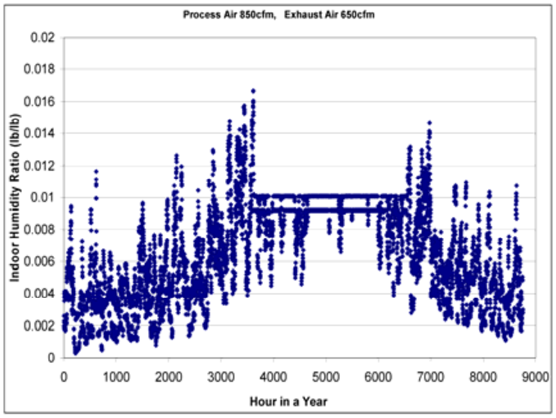

Fig.11b. Case 8. OA: 850CFM, Exhaust Air 650CFM, ACH:0.001, Supply air humidity ratio: 0.0092 lb/lb.

Because the infiltration has a significant impact on indoor humidity ratio in a radiantly cooled space, measures must be taken to reduce the infiltration. One method is to pressurize the building to reduce or stop outside air entering the space in summer. Two cases (cases 7 and 8) are simulated. Case 7 assumes 650CFM of OA is supplied without exhaust air and heat recovery in order to pressurize the space as shown in Figure 11a. Case 8 assumes 850CFM of OA is supplied and 650 CFM air is exhausted with heat recovery to slightly pressurize the space as shown in Figure 11b. Figures 11a and 11b indicate that indoor humidity ratio can be easily controlled under 0.011 lb/lb (dew point of 52°F) in both conditions if the space is pressurized and infiltration is reduced to close to zero. However, there is another drawback. In Case 7 of Figure 11a; there is no heat recovery because there is no exhaust air. In Case 8 of Figure 11b, increasing ventilation air from 650 CFM to 850 CFM will increase energy consumption.

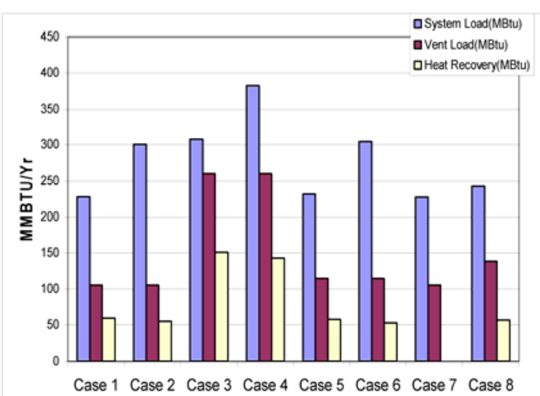

Fig.12a System load, ventilation load and heat recovery of different cases.

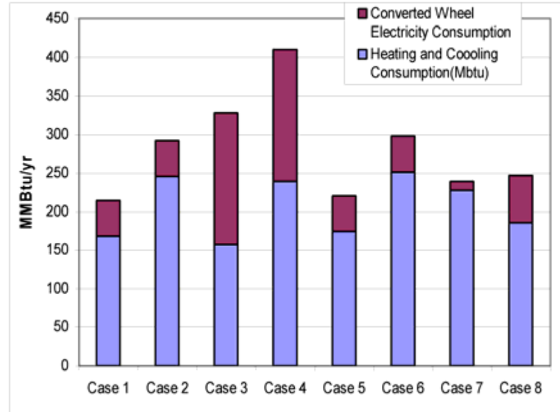

Fig.12b Estimation of net energy consumption per year

Energy consumption values for each of Cases 1 (Figure 8a) through 8 (Figure 11b) have been simulated. The results are shown in Figure 12. The conditions corresponding to each of cases 1 to 8 are given in Table 1. The ventilation system is assumed to run from 6:00am to 8:00pm. The total building system load is compared with the ventilation load and heat recovery in each case. The results are shown in Figure 12a. The total system load is calculated by assuming that the OA is conditioned by a normal air-handling unit without heat recovery. The ventilation load is calculated by assuming that 100% ventilation air is conditioned by a cooling or heating coil in the desiccant unit without heat recovery in order to compare with the heat recovered by the desiccant wheel. From Figure 12a, it can be seen that the higher the ventilation rate, the higher the building system load and ventilation load. The heat recovery is also higher. The amount of heat recovered by a passive desiccant wheel accounts for 50% of the ventilation load. Figure 12 shows that infiltration and ventilation ratio are two important factors affecting the total system load. These two factors explain why the system load and ventilation load of cases 3 and 4 is largest as shown in Figure 12a. By considering the electricity used by the passive desiccant wheel itself, the net energy consumption is as shown in Figure 12b. Electricity consumption (kWh) is converted to thermal energy (MMBTU) and multiplied by the chiller's COP of 2.5. The result shows that case 1 (OA 650CFM, ACH 0.001) uses the least energy. The net energy consumption of cases 1, 5, 7 and 8 are relatively close to each other. Compared with the indoor humidity levels in Figures 8 to 11, it can be concluded that case 8 is the best solution for a leaky space with a radiant cooling system. That means pressurizing the space or sealing leakage sites is very important to control condensation in a radiantly cooled space.

6. OPERATING STRATEGIES TO CONTROL CONDENSATION

To avoid condensation in a radiantly cooled pace, the following operating strategies are recommended

1) Occupants and infiltration air are the main sources of moisture for the indoor environment in summer. Infiltration air can lead to condensation on radiant panels. Checking and caulking leakage points in the space will reduce infiltration and eliminate condensation if infiltration is sufficiently reduced. Window opening should be restricted in summer when radiant cooling is running. The supply air rate should be higher than the exhaust air flow as shown in case 8 to pressurize the space. Although some energy is lost when pressurizing the space, this can be an effective means for controlling indoor humidity level.

2) The desiccant ventilation system should start at least one hour before the space is occupied. When the space is highly occupied, the humidity sensor in the space should be able to modulate the cooling coil control valve in the desiccant unit to reduce the supply air temperature and humidity ratio. Meanwhile, when the measured dew point of the indoor air is close to the inlet water temperature of the radiant panel, the inlet valve of the radiant panel should be shut down.

3) In a radiantly cooled space with an integrated desiccant ventilation system, space cooling is provided by two sources, radiant panels and ventilation air. At low loads, cooling should be provided by ventilation air. As cooling load increases, the temperature of the supplied ventilation air should be adjusted to match the load. When the supplied ventilation air temperature drops to a low limit of 55^{o}F or 52 ^{o}F at high cooling loads, the inlet control valve of the radiant panels starts to modulate to maintain the room air temperature at 76^{o}F. The radiant panels will not be enabled until indoor dew point is below a safe limit such as 0.011lb/lb (dew point of 60^{o}F). Then the temperature of the chilled water entering panels is modulated to meet the space sensible load. The inlet water temperature should be controlled to be 1-2^{o}F higher than the space dew point temperature to avoid water condensation.

4) Ventilation systems in radiantly cooled space can be oversized in the design phase. Sometimes, the oversized ventilation system can satisfy the cooling load alone even on a hot day. However, the higher ventilation rate will increase energy consumption. The supply air fan should be a variable speed fan to match the ventilation air with the space fresh air requirement so the space will not be over-ventilated.

5) A passive desiccant ventilation system increases the indoor humidity in spring and fall when there is no cooling load in the space and post cooling is shut off to save energy. The indoor humidity ratio will be too high to be comfortable in a tight space as shown in Figures 8a, 9a, and 10a. Windows should be allowed to open. The drier outside air can remove indoor moisture.

7. CONCLUSIONS

A passive desiccant system and an active desiccant system have been compared in this paper. The transient processes of dehumidification in a radiantly cooled space have been studied. A transient model is set up. Hourly indoor humidity at eight different operating conditions is analyzed based on the steady state of the transient model. The corresponding energy consumption values of the different cases have been simulated. Comparing energy consumption and yearly indoor humidity trends of the eight cases, the following conclusions can be reached:

- An active desiccant system dries a space deeply and continuously, while a passive desiccant system dries a space more energy efficiently. The moisture removal capacity of a passive desiccant system depends on the dryness of the exhaust air. When a passive ventilation system is the only source of dehumidification, the system cannot remove moisture without post cooling.
- High infiltration is one of the main causes of condensation in a radiantly cooled space in summer. Radiant panels cannot work without condensation in a leaky space (ACH: 0.45) even if the supply air is conditioned to 52^{o}F, 0.008lb/lb as shown in Figure 10b. A passive desiccant system may cause some humidity problems in a tight space in spring and fall when post cooling is not necessary because of no cooling load as shown in Figures 8a, 9a, 10a. Opening windows during this period of time can solve this problem.
- Pressurizing the space with ventilation air is one of the possible solutions to avoid water condensation on the surface of radiant cooling panels in a leaky building.
- An optimized cooling control sequence is necessary for condensation control, such as starting the ventilation system one hour before the space is occupied and cooling the space by stages.
- A passive desiccant ventilation system can recover about 50% of the energy of the ventilation load and provides reasonable humidity control in a tight, radiantly cooled space.
- The ventilation rate has a great impact on energy consumption in an oversized ventilation system.

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NOMENCLATURE

ach_i = Hourly air exchange ratio

F1 = Ideal isopotential line of enthalpy

F2 = Ideal isopotential line relative humidity

C_p = Specific heat capacity (Btu/lb*F)

$\dot{m}$ = Mass flow rate lb/min

$\dot{m}_{gen}$ = Moisture generation lb lb/hr

T = Temperature, ^oF or K

V_r = Space volume ft^3

$\dot{V}_s$ = Supplied outside air flow volume rate ft^3/hr

ρ = Air density lb/ft^3

W = Absolute humidity ratio, lb/lb

ε_l = Moisture transfer effectiveness

ε_s = Sensible heat transfer effectiveness

ε_{F1} = Effectiveness of total energy transfer at optimum rotary speed

ε_s = Effectiveness of moisture transfer at optimum rotary speed

Subscripts

s = Supply air

o = Outside air

r = Room air

o1 = Inlet of supply air at passive or active wheel

o2 = Outlet of supply air at passive or active wheel

e1 = Inlet of exhaust air at passive desiccant wheel

e2 = Outlet of exhaust air at passive desiccant wheel

r,1 = Inlet of regeneration air at active desiccant wheel

r,2 = Outlet of regeneration air at active desiccant wheel

min = minimum

REFERENCES

[1] Stetiu, Corina, 1999, Energy and Peak Power Savings Potential of Radiant Cooling Systems in US Commercial Buildings, *Energy and Buildings*, Vol 30, 127-138.

[2] Niu, J.L, Zhang, L.Z, Zuo, H.G., 2002, Energy

[3] savings potential of chilled ceiling combined with desicant cooling in hot and humid climates, Vol 34. *Energy and Buildings*, 487-495.

[4] Mumma, S.A., Jeong, Jae-Weon, 2005, Direct Digital Temperature, Humidity, and Condensate Control for a Dedicated Outdoor Air-Ceiling Radiant Cooling Panel System, ASHRAE Transactions, Vol 111, Part 1, 547-558.

[5] Mumma, S.A., 2002, Chilled ceilings in Parallel with dedicated outdoor air systems: Addressing the concerns of condensation, capacity, and cost, ASHRAE Transactions, Vol 108, Part 2, 220-231.

[6] Mumma, S.A., 2003, Chilled Ceiling Condensation Control, ASHRAE *IAQ Application*, Fall Issue, 22-23.

[7] Mumma, S.A., 2001, Condensation Issues with Radiant Cooling Panels, *ASHRAE IAQ Application*, Fall Issue, 1-3.

[8] Zhang, L.Z, Niu, J.L., 2003, Indoor Humidity Behaviors Associated with Decoupled Cooling in Hot and Humid Climates, *Building and Environment*, Vol 38, 99-107.

[9] Claridge, David E., Bensouda, N., Lee, S.U., Wei, G., Heinemeier, K., Liu, M., Manual of Procedures for Calibrating Simulations of Building Systems, Lawrence Berkley National Laboratory, October, 2003.

[10] Harriman, Lewis G. III, Czachorski, Marek, White, M.J., Kosar, D.R., 1999, Evaluating Active Desiccant Systems For Ventilation Commercial Buildings, *ASHRAE Journal*, Vol 41, n10, 28-33.

[11] Babus'Haq, R.F., Olsen, H., Probert, S.D., 1996, Feasibility of Using an Integrated Small Scale CHP unit plus Desiccant Wheel in a Leisure Complex, *Applied Energy* Vol 53, 179-192.

[12] Bulk, E. Van den, Mitchell, J.W., Klein, S.A., 1985, Design theory for rotary heat and mass exchangers-II. Effective-number of transfer units method for rotary heat and mass exchangers, *International Journal of Heat and Mass Transfer*, Vol. 28, 1587-1595.

[13] Simonson, C.J., Besant, R.W., 1999, Energy Wheel Effectiveness. Part 1-development of dimensionless groups. *Int. J. Heat Mass Transfer*, Vol 42(12), 2161-2170.

[14] Simonson, C.J., Besant, R.W., 1999, Energy Wheel Effectiveness. Part 2-Correlations, *Int. J. Heat Mass Transfer*, Vol 42 n12, 2171-2185.

[15] Simonson, C.J., Shang, W., Besant, R.W., 2000, Part-Load Performance of Energy Wheels: Part I-wheel speed control, *ASHRAE Transactions* Vol 106 part 1, 286-300.

[16] Freund, S., S.A. Klein, D.T. Reindl, 2003, A semi-empirical method to estimated enthalpy exchanger performance and a comparison of alternative frost control strategies, *HVAC&R Research* Vol 9 (4), 493-508.

[17] Jeong, Jae-Weon, Mumma, Stanley A., 2005, Practicl thermal performance correlations for molecular sieve and silica gel loaded enthalpy wheels, *Applied Thermal Engineering*, Vol 25, 719-740.

[18] Jurinak, J.J., Mitchell, J.W., Beckman, J.W. 1983, Open cycle desiccant air conditioning as an alternative to vapor compression cooling in residential applications. Proceedings of the ASME Solar Energy

[19] Division Fifth Annual Conference, Orlando, FL, USA, 374-381.

[20] ASHRAE. 2004 *ANSI/ASHRAE Standard 55-2004, Thermal Environment Conditions for Human Occupancy* . Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

[21] Gatley, Donald P., Dehumidification Enhancements for 100% Percent Outside Air AHUs, Donald P. Gatley, Heating/Piping/Air Conditioning Engineering, September 2000, 31-35.

[22] Conroy, C.L., Mumma, S.A., 2001, Ceiling radiant panels as a viable distributed parallel sensible cooling technology integrated with dedicated outdoor air systems. ASHRAE Transactions, v107, part 1, 2001, p578-585.