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Condensation Risk in a Room with High Latent Load and Chilled Ceiling Panel and with Air Supplied from Liquid Desiccant System

Yonggao Yin * **Xiaosong Zhang, Ph.D.** **Qingyan(Yan) Chen, Ph.D.**

Fellow ASHRAE

ABSTRACT

Radiant cooling has shown great advantages of improving thermal comfort and can also reduce energy demand. But the system may have a high condensation risk if used with a high latent load. This study proposed a new ventilation system with radiant cooling panel and air supplied from a liquid desiccant dehumidification system, which provided very dry supply air and chilled water for radiant cooling. This study used an office with a high latent load to demonstrate the application of such a system in hot and humid climate. By using a validated Computational Fluid Dynamics (CFD) program to calculate the distributions of air temperature, humidity, and velocity as well as the Predicted Mean Vote in the office, the results showed that the new system could reduce the risk of condensation on the cooling panels and offer a comfortable indoor environment. The new system has a great potential for use in hot and humid climate regions.

INTRODUCTION

Radiant cooling has shown great advantages of improving thermal comfort due to reduced vertical temperature gradient in the occupied zone and to a low air speed (Behne 1999; Imanari et al. 1999, Hao et al. 2007). Radiant cooling can also reduce energy demand because the energy used by its pump is lower than that by fan in all air systems (Sodec 1999; Novoselac and Screbric 2002). With radiant cooling, the room air temperature in summer could also be higher than that in all air systems so that the cooling load would become smaller. As a result, the energy demand can be further reduced (Feustel and Stetiu 1995). Since radiant cooling alone cannot remove latent load and pollutants, such as volatile organic compounds, it should be supplemented with ventilation. In the 1990s, radiant cooling was used in combination with displacement ventilation in Europe. Experimental investigation and practical experiences in Europe (Givoni 1991; IEA 1998) have proved that radiant cooled ceilings are able to remove high cooling loads with good thermal comfort. For example, Niu (1994) found that cooled ceiling combined with displacement ventilation can produce a thermally comfortable environment at a cooling load up to 50 W/m^2, compared with 40 W/m^2 with only displacement ventilation. However, Vangtook and Chirarattananon (2006) studied radiant cooling with natural ventilation under hot and humid climate in Thailand. The temperature of radiant cooled water was limited to 24ºC to avoid condensation. They found that the low heat reception

* **Yonggao Yin** is a Ph.D. candidate, **Xiaosong Zhang** is a professor in the School of Energy and Environment, Southeast University, Nanjing, China. **Qingyan Chen** is a professor of mechanical engineering in the School of Mechanical Engineering, Purdue University, West Lafayette, IN. Yonggao Yin is a visiting scholar at Purdue when this study was performed.

capacity of the panel would limit its use only to situations when loads were low. Obviously, the system is good for dry climate and could be problematic for humid climate. In the humid climate or high latent load space, such as operating theaters and certain hospital wards, a very high ventilation air requirement of a conventional system is essential or mandated, as pointed out by Ameen and Mahmud (2005), which would cause more energy consumption and cost, even discomfort.

Using desiccant dehumidified ventilation for the radiant cooling system could be competent for that situation. Desiccant dehumidified ventilation can provide very dry air to decrease the supply air flow rate, because the humidity ratio is lower than that obtained with a conventional vapor compression chiller used for all-air systems. This makes it desirable for buildings in hot and humid regions, like southeastern countries of Asia, where the risk of condensation is very high due to the low surface temperature in the radiant cooling panels. Another advantage is that sensible and latent cooling load are decoupled and controlled separately, which means that control of humidity can be achieved better than with conventional vapor compressive systems. Many investigations have been conducted on using desiccant for radiant cooling. Niu et al. (2002) proposed a hybrid air conditioning system using desiccant rotor to dehumidify ventilated air. Their results show that up to 44% primary energy demand could be reduced with the hybrid system, in comparison with a conventional all-air system. Ameen and Mahmud (2005) conducted experiment study on desiccant dehumidification with hydronic radiant cooling system in humid tropical climates and the trial run confirmed that the condensation problem was not insurmountable; furthermore the system could be easy to agree with the load changing. Tsay et al. (2006) experimentally studied a desiccant cooling system with a heat pump by using the condensation heat for the desiccant rotor regeneration to improve the system thermal performance. Their results show that the COP of the desiccant system was estimated to be 2.32, which was higher than the thermal performance of heat-driven desiccant system. However, energy consumed by the system is high-grade electricity.

Convectional radiant cooling uses chilled water from electricity-driven vapor compression chillers. The chillers have to use CFC or HCFC refrigerants at present and electricity. The use of such refrigerants is bad for the environment. Electricity is very high grade energy. Thus, it is necessary to find an alternative system that does not used vapor compressive chiller.

This study proposes a new liquid desiccant cooling system for hot and humid climate. The system uses chilled water for radiant panels by dehumidifying and evaporative cooling while supplies dry air for space ventilation.

DESCRIPTION OF THE NEW LIQUID DESICCANT SYSTEM

Fig.1 (a) shows our new liquid desiccant dehumidification/radiant cooling system. The new system consists of four units: (I) liquid desiccant ($LiCl$-H_2O) regeneration unit, (II) air dehumidification unit, (III) unit producing chilled water by evaporative cooling, and (IV) radiant cooling and dehumidified ventilation unit. The liquid desiccant regeneration unit is used to concentrate the diluted liquid desiccant from the dehumidifier by solar energy. Our early study (Yin et al. 2007) indicated that the regeneration of liquid desiccant could be driven by 60-80ºC low-grade heat sources, such as solar energy. The air dehumidification unit uses liquid desiccant to produce very dry air for evaporative cooling unit and to condition the space. Different dehumidification processes could be used depending on the type of dehumidifiers, such as internally-cooled dehumidifiers and adiabatic dehumidifiers. The one shown in Figure 1 used an adiabatic dehumidifier. The air conditioned space can be kept as a thermally comfortable environment by radiant cooling and ventilation with dry air. The dry air is a part of that from the dehumidifier. The other part of the dry air enters the evaporative cooler to produce chilled water for the radiant ceiling panels. The temperature of the chilled water can be as low as 12ºC with good thermal performance (Yin et al. 2006).

Fig.1 (b) depicts the air conditioning processes of this new system on a psychrometric chart. Indoor air quality standard requires fresh air (state 0) being supplied to the system. The fresh air is then cooled by an indirect evaporative cooler to state 0' before it is mixed with return air (state 1) to state 2. The mixed air (state 2) enters the dehumidifier, which dries the air with reduced enthalpy to state 3. The air in state 3 is cooled naturally to state 4 with a cooling water coil, and is further cooled down to state 5 by the low-temperature air from the evaporative cooler. In succession, the air at state 5 is divided in to two parts. The part enters the evaporative cooler (state 7) absorbs moisture and cools down to state 8. The other part (state 6) enters the room to condition the room air to state 1. Finally the air at state 8 absorbs the heat from process 4 to 5 and then continues to absorb more heat from the fresh air (point 0) in combination of indirect evaporative cooling.

The new system also has the following advantages:

(1) The system uses low-grade heat with temperature of only 60-80ºC to achieve refrigeration and air conditioning;

(2) Due to the use of radiant cooling, the room air temperature in summer could also be higher than that in all air systems so that the cooling load would become smaller with the same comfort level;

(3) The new system can decouple in removal sensible and latent heats, which provides flexibility for handling a large variation of sensible/latent heat ratio in the same building and convenience for multi-zone control.

CASE SETUP

In order to demonstrate the suitability of the new system for hot and humid climate or space with high latent load, it is essential to ensure all the four units in the system are feasible. Our early studies (Yin et al. 2006; 2007) have demonstrated that the liquid desiccant (LiCl-H_2O) regeneration unit and air dehumidification unit are feasible. The focus of this study was at the radiant cooling and dehumidified ventilation unit in an indoor space.

This study used a hypothesis conference room as shown in Fig. 2. The room dimension was 5.16 m long, 3.65 m wide and 2.43 m high with 10 occupants and a coffee maker. Each occupant produced 0.0236 g/s water vapor and the coffee maker 0.028 g/s (Novoselac and Srebric 2002). The sensible heat load in the room was from the six lights (30W each), the coffee maker (160W), the occupants (75W each) and the walls (6 W/m^2). The room used displacement ventilation that supplied conditioned air through the diffuser at the floor level and exhausted air from the ceiling. The flow rate of the supply air was 188.1m^3/h at 20ºC (4 ACH). Radiant ceiling panels covered the entire ceiling. The radiant ceiling temperature was 18ºC. Due to high occupant density and moisture generated from the coffee maker, the conference room has a very high risk of condensation in hot and humid climate. The risk is particularly severe if the conference room uses radiant cooling panels.

RESEARCH METHOD AND METHOD VALIDATION

This study used a CFD program to determine the condensation risk because it provides very detailed distribution of moisture in the room. Since the room airflow is turbulent, the CFD program used the RNG k-ε model, and Zhang et al. (2007) found that the model was robust and accurate. The CFD program solved the following governing equation:

$$\frac{\partial}{\partial t}(\rho\phi)+\frac{\partial}{\partial x_i}(\rho u_i\phi)=\frac{\partial}{\partial x_i}\left(\Gamma_\phi\frac{\partial\phi}{\partial x_i}\right)+S_\phi \tag{1}$$

More detailed information on the CFD model could be referred to the paper from Zhang et al. (2007). Our grid refinement study found that this conference room would need 69 by 53 by 41 cells. Fig. 3 shows the grid distribution.

This investigation used the approach suggested by Srebric and Chen (2002) to couple the convective and radiative heat transfer. The radiative heat from an object surface is first calculated as:

$$Q_r=\varepsilon_{obj}\sigma\left(T_{obj}^4-T_w^4\right)A_{obj} \tag{2}$$

In this study, the radiative sources are occupants, coffeemaker and lights. In order to calculate the radiative heat from the sources, the surface temperatures (T_{obj}) of the occupants, coffeemaker and lights are assumed respectively as 30 ºC, 55 ºC and 45 ºC. The cooling panel temperature is constant and the same as that of the chilled water. The convective heat of every heat source is the total heat minus the radiative heat.

$$Q_c=Q_t-Q_r \tag{3}$$

Since the CFD used approximations and simplifications on boundary conditions for diffusers, it is necessary to validate the CFD program before it could be used as a tool for this study. The validation was done by comparing the CFD results with experiment data in an environmental chamber with displacement ventilation done by Yuan et al. (1999). Fig.4 shows the three-dimensional view and plane view of the experimental chamber layout. The chamber had a displacement diffuser, a window, an exhaust, two occupants, two computers, two tables, two pieces of furniture, and six ceiling lamps. The case is very similar to the one described in the previous section, except no radiant ceiling panels. If the CFD program could predict Yuan's case, it can be a reliable tool for the room with the ceiling panels as described in the previous section. All boundary conditions, including surface temperatures, for the CFD program were obtained through the experimental measurements. The convective heat from a heat source, such as from the lighting, the occupants, and the computers, were calculated also evaluated according to equation (3).

Fig.5 compares the normalized air temperature, air velocity, and tracer-gas concentration distributions, respectively, along the height of the chamber at three different positions. Fig. 4(b) shows the three locations in the chamber, which are rather typical locations, among the nine locations where Yuan et al. conducted the measurements. Fig.5 (a) shows that the computed air temperature profiles agree well with the measurement data. Fig.5 (b) shows that the air velocity in the chamber was low. The computed and measured data agree with each other in most of the positions but discrepancies exist. Both the computational and experimental results were not free from errors. The errors from the computed results could be due to the approximations used in the turbulence model, while the false air velocity produced by the omni-directional anemometers due to the convection from the heated probes was of the same magnitude of the actual air velocity. Fig. 5(c) shows that the computed tracer-gas concentration contributions agreed reasonably with experiment data with rather notable errors. As pointed by Yuan et al. (1999), the measured tracer-gas concentration had a high uncertainty. Nevertheless, the general trend of the computed concentration still reflected the measured one. For our study, such accuracy on prediction air temperature, air velocity, and tracer-gas concentration is acceptable.

The tracer-gas concentration was used by Yuan et al. to simulate a gaseous contaminant. This investigation used it for simulating water vapor in the air. Thus, it is rather meaningful to compare the three different parameters, which are important in the new desiccant system presented in this paper. Both the computed and measured air temperature and tracer-gas concentration distributions indicate stratified flow in the environmental chamber.

This study also used standard BS EN ISO7730 (Moderate thermal environments-determination of PMV and PPD indices) to evaluate thermal comfort. The PMV stands for Predicted Mean Vote and PPD for Predicted Percentage Dissatisfied. The PMV is calculated from ASHRAE standard (2005):

$$\begin{aligned} PMV = &\left[0.303\times\exp\left(-0.036M\right)+0.028\right]\Big\{\left(M-W\right)-h_c f_{cl}\left(T_{cl}-T_a\right)-3.05\left[5.73-0.007\left(M-W\right)-P_v\right] \\ &-0.42\left[\left(M-W\right)-58.15\right]-0.0137\left(M-W\right)\left(5.87-P_v\right)-3.96\times10^{-8}\left[\left(T_{cl}+273\right)^4-\left(T_w+273\right)^4\right]f_{cl}\Big\} \end{aligned} \tag{4}$$

where

$$\begin{aligned} T_{cl} = &\,35.7\text{-}0.0275\left(M-W\right)\text{-}0.155\times I_{cl}\Big\{(M-W\text{-}3.05\left[5.73-0.007\left(M-W\right)-P_v\right] \\ &\text{-}0.42\left[\left(M-W\right)-58.15\right]\text{-}0.0173\times M\cdot(5.87\text{-}P_v)\text{-}0.0014\times M\cdot(34\text{-}T_a)\Big\} \end{aligned} \tag{5}$$

and

$$h_c = \begin{cases} 2.38\left(T_{cl}-T_a\right)^{0.25} & 2.38\left(T_{cl}-T_a\right)^{0.25} > 12.1\sqrt{U_a} \\ 12.1\sqrt{V_a} & 2.38\left(T_{cl}-T_a\right)^{0.25} < 12.1\sqrt{U_a} \end{cases} \tag{6}$$

$$f_{cl} = \begin{cases} 1.00+1.290 I_{cl} & I_{cl} \le 0.078 m\cdot°C/W \\ 1.05+0.645 I_{cl} & I_{cl} > 0.078 m\cdot°C/W \end{cases} \tag{7}$$

The PMV is a function of air temperature, environmental temperature, relative humidity, air velocity, clothing level, and metabolism. The CFD can calculate air temperature, environmental temperature, relative humidity, and air velocity for the PMV model. Loveday (2002) studied thermal comfort for sedentary workers in offices with displacement ventilation and chilled ceiling by using the PMV model and found that the model is good for such a system.

RESULTS AND DISCUSSION

Condensation Risk Analysis

The new liquid desiccant dehumidification and radiant cooling system (hereafter "desiccant system") supplies air to an indoor space with a humidity ratio as low as 6.2 g_v/kg_a for the displacement ventilation. However in conventional radiant cooling system (hereafter "conventional system") using cooling with dehumidification, the humidity ratio of the supply air is around 8.3 g_v/kg_a in hot and humid climate, which is higher than that in the new system. The two humidity ratios have been used in our study to investigate condensation risk on the ceiling panels.

Fig.6 illustrates the relative humidity contours on the radiant ceiling panels for the conventional and desiccant systems. In the conventional system, condensation would occur on the upper-left corner of the panels since the relative humidity in the area reached 100%. The relative humidity on the panels was mostly higher than 90%, which implies a very high risk of condensation. In the desiccant system, the relative humidity on the ceiling panels was lower than 90% so that the risk of condensation was minimal. Note that the high relative humidity regions on the ceiling panels were either above the coffeemaker or above the occupants due to their thermal plumes. Fig.7 shows the thermal plumes through the section of the occupants. The results clearly conclude that the condensation risk of the desiccant system is much lower than that of the conventional system.

Fig.8 compares the humidity ratio distribution of the conventional system with that of the desiccant system at three positions indicated in Fig. 2(b). Humidity ratio in the room was about 2 g_v/kg_a lower because the supply air from the desiccant system was 2.1 g_v/kg_a drier than that from the conventional system. Pole 2 represents typical humidity ratio distribution in the room that shows a small gradient of humidity ratio in the lower part. In the upper part of the room, the humidity ratio was nearly uniform. The distribution was different from that in displacement ventilation without radiant cooling panels as measured by Yuan et al. (1999) with tracer-gas. The main reason is likely caused by the downward airflow from the ceiling panels due to negative buoyancy. The higher humidity in the mid-height at pole 3 was caused by local recirculation that brought moisture to the region. The airflow pattern in the room was very complex. Pole 5 depicts a higher humidity near the ceiling because of the thermal plume from the coffee maker.

Thermal Comfort Analysis

As mentioned previously, air temperature, environmental temperature, air velocity and air humidity are important parameters to indoor thermal comfort. Also vertical gradient of air temperature around the occupants has also impact on thermal comfort (ASHRAE 2005). Fig.9 shows the vertical profiles of air temperature at those representative locations in the conference room. This study controlled the average air temperature in the room to be 25ºC. Both the desiccant and conventional systems had a vertical temperature gradient in the lower part of the room. The temperature gradient in the occupied zone was less than 2 ºC/m, which was within the limit for thermal comfort. The upper part appeared to be well mixed due to the strong convection from the thermal plumes from the coffee maker and the occupants and the downward flow from the radiant panels.

Fig.10 shows the relative humidity profiles in the conference room. The relative humidity in the room for the conventional system varied from 60% to 70%, while that for the desiccant system from 50% to 60%. In hot and humid climates, the 10% difference offers a significant benefit in removing excessive humidity. The exhaust relative humidity in the desiccant system is 55%, but 70% in the conventional system.

Fig.11 shows vertical profiles of airflow velocity in the room. Since humidity played little role in air distribution, the airflows in the two systems were identical. Near the ceiling panel, the air velocity was generally higher due to downward flow from the ceiling panel and also due to impinging of the thermal plumes from the coffee maker and the occupants. The high velocity near the floor in poles 1 and 2 was caused by the flow from the diffuser. Nonetheless, the air velocity in the occupied zone was lower than 0.1 m/s so that draft should not be a concern.

To check the thermal comfort or discomfort, PMV index was also computed to predict the mean value of votes, which relates to the conditions of indoor environment and human. The parameters of indoor environment, such as the air temperature, environmental temperature, air velocity, relative humidity, are available from the CFD program, and also the conditions of human should be concerned, here we assumed clothing level of 0.8 clo and metabolism of 1.2 Met. This investigation used the PMV model described by equation (3) to calculate the thermal comfort level in the room. Fig.12 illustrates the PMV values at 1.1 m above the floor in the five poles in the room. In the occupants area (pole 1 through 3), better thermal comfort is showed in the new desiccant system since PMVs at pole 1 to 3 seem close to zero, which is on account of low humidity air supply by the new system. At pole 4, it feels a bit cool because of the downward airflow from the cooled ceiling, where higher air velocity is illustrated in Fig.11. Therefore, appropriate design and operation parameters are significant to gain good thermal comfort for the new system. Pole 5 is very close to the coffee maker, which means that there is high temperature and humidity at that location, and therefore pole 5 shows very high PMV and indicates worst thermal comfort.

CONCLUSIONS

This study presented a new ventilation system with radiant cooling and air supplied from liquid desiccant dehumidification. By comparison with a conventional radiant cooling system using a water chiller, the new system can reduce the risk of condensation in air-conditioned spaces with high latent load in hot and humid climate, such as auditoria. The system also creates a slightly better thermal comfort level compared with a conventional system. Thus, the new system could be better for hot and humid climate regions.

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NOMENCLATURE

Symbols

A_{obj} = surface area of the objects
C = concentration
d = absolute humidity
f_{cl} = surface area ratio of clothed-man and nude-man
H = height
h_c = heat transfer coefficient of convection
I_{cl} = thermal resistance of clothing
M = rate of metabolic heat production
PMV = predicted mean vote
P_v = water vapor pressure
Q = heat exchange rate
RH = relative humidity
S_Φ = source
T = temperature
t = time
U = velocity
u_i = velocity component in i direction
W = active work of occupant
x, y = Cartesian coordinates
Z = dimensionless height

Greek Symbols

Γ_Φ = effective diffusion coefficient
ε_{obj} = surface emissivity of the heat object
ρ = air density
σ = Stefan-Boltzman constant
Φ = a general variable in Equation 1

Subscripts

a = air
c = convection
cl = cloth
i = Cartesian coordinate in x direction
obj = heat object
r = radiative
t = total
v = vapor
w = wall

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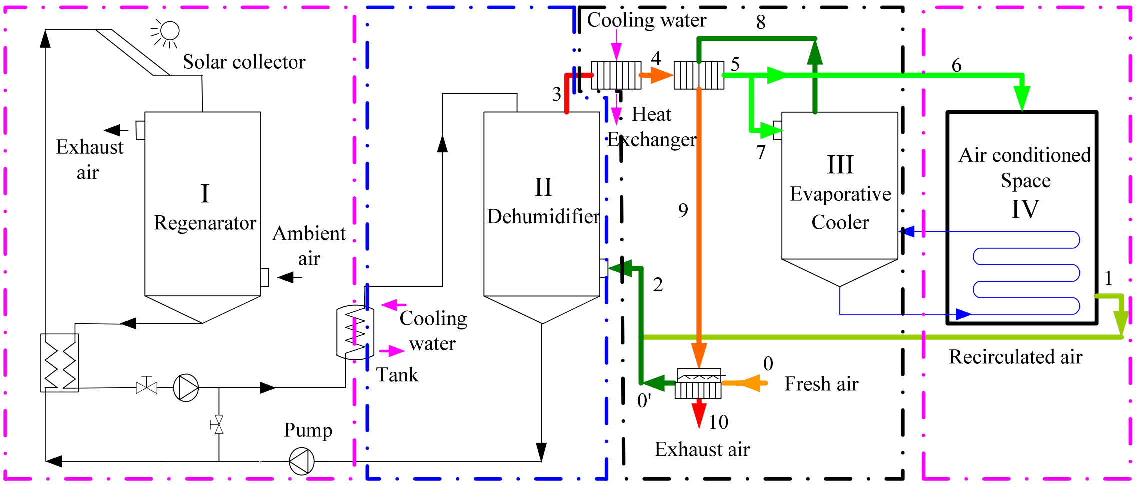
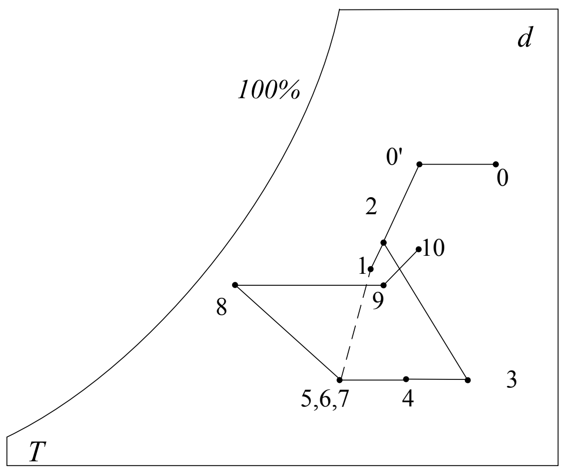
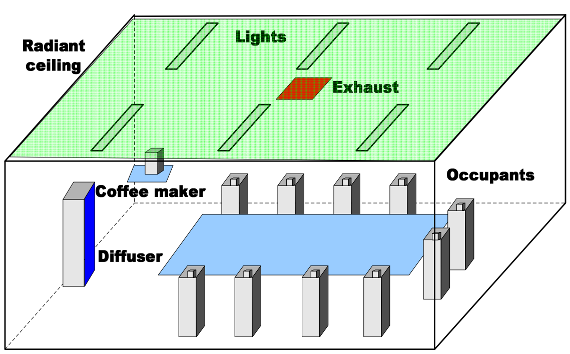


(a) System sketch

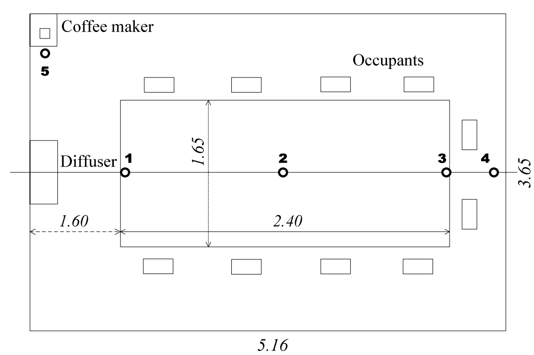


(b) Air conditioning processes in psychrometric chart

Fig.1 A new liquid desiccant dehumidification/radiant cooling system

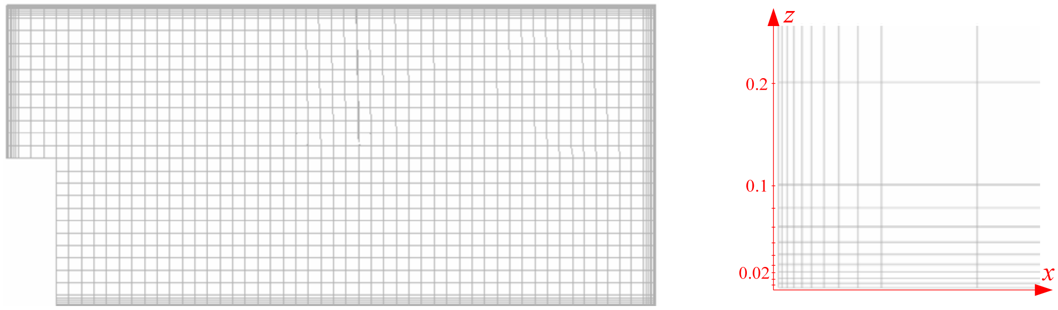


(a)



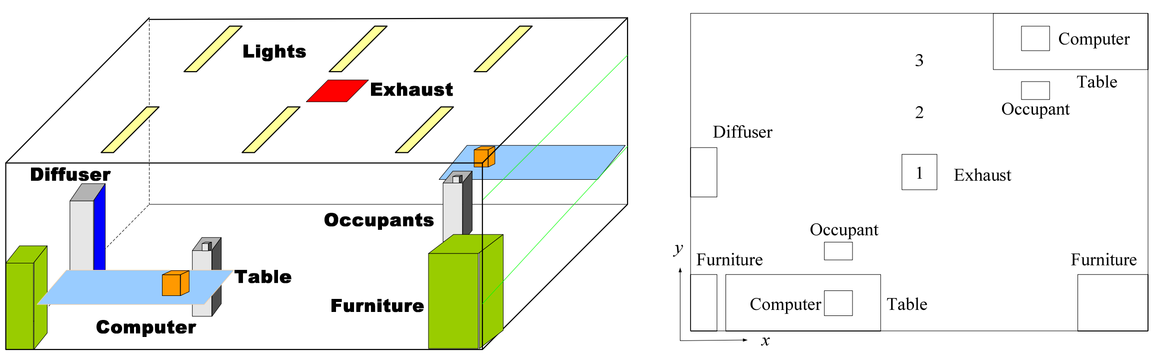
(b)

Fig.2 Conference room model for analysis: (a) conference room layout; (b) plan view of the conference room.



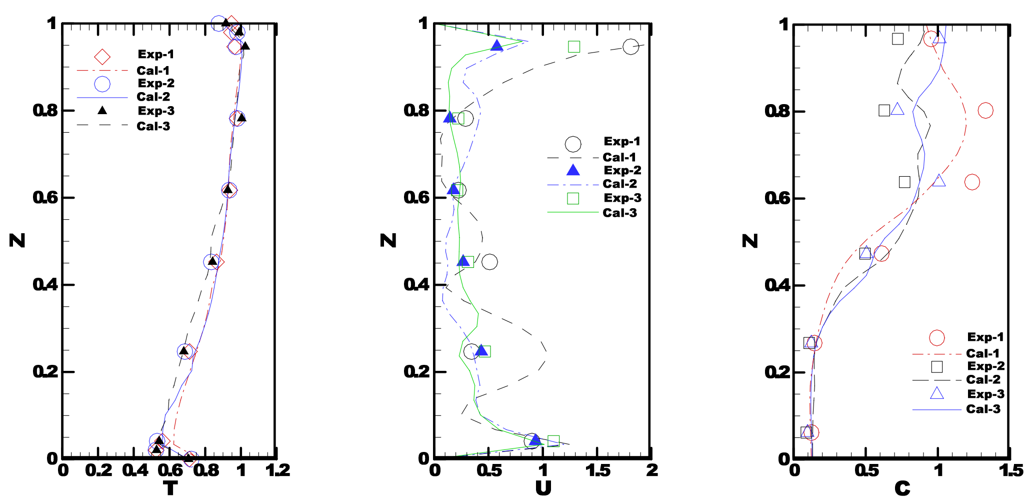
(a) (b)

Fig.3 Grid distribution used for the conference room: (a) mid-section in the longitudinal direction; (b) amplified view in the corner



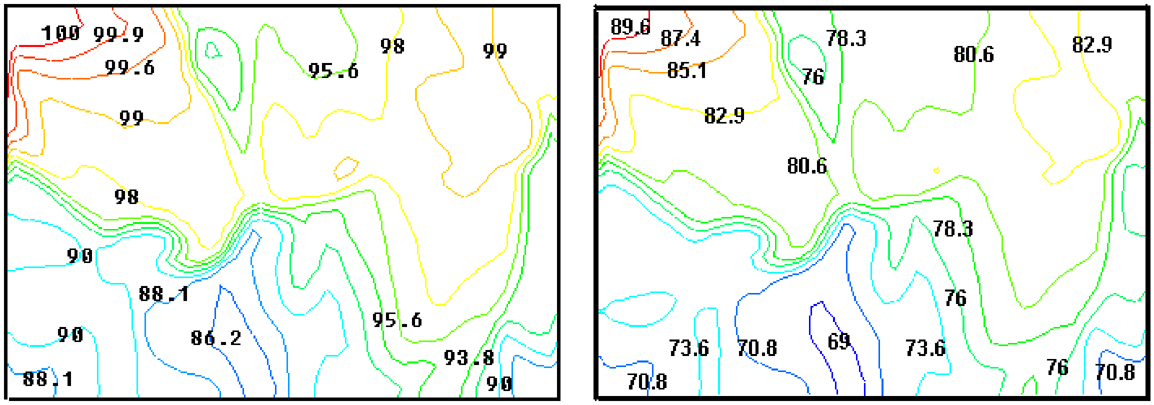
(a) (b)

Fig.4 Experimental room schematic map: (a) Diorama; (b) Planform



(a) (b) (c)

Fig.5 Comparison of the normalized air temperature (a), air velocity (b), and tracer-gas concentration (c) obtained by CFD with the experiment data (Exp -experiment data; Cal –calculated results; $T=(t-t_{in})/(t_{out} - t_{in})$;$U = u/u_{in}$; $C = (c - c_{in})/(c_{out} - c_{in})$; and Z = (height from floor)/(room height))



(a) **(b)**

Fig.6 Relative humidity distributions on the ceiling panels: (a) Conventional system; (b) Desiccant system

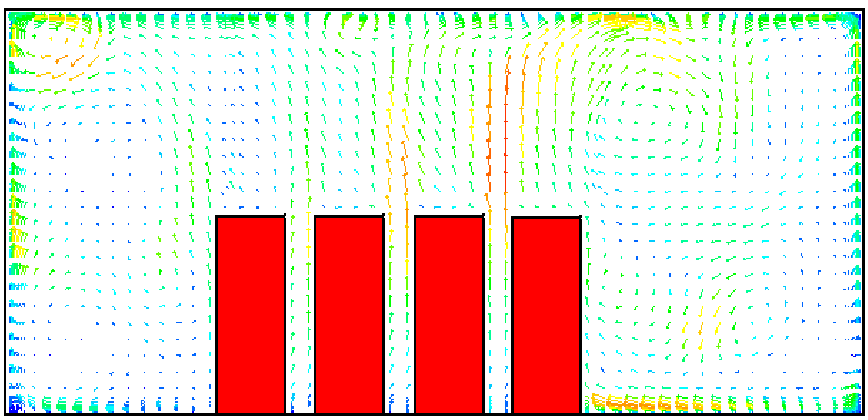
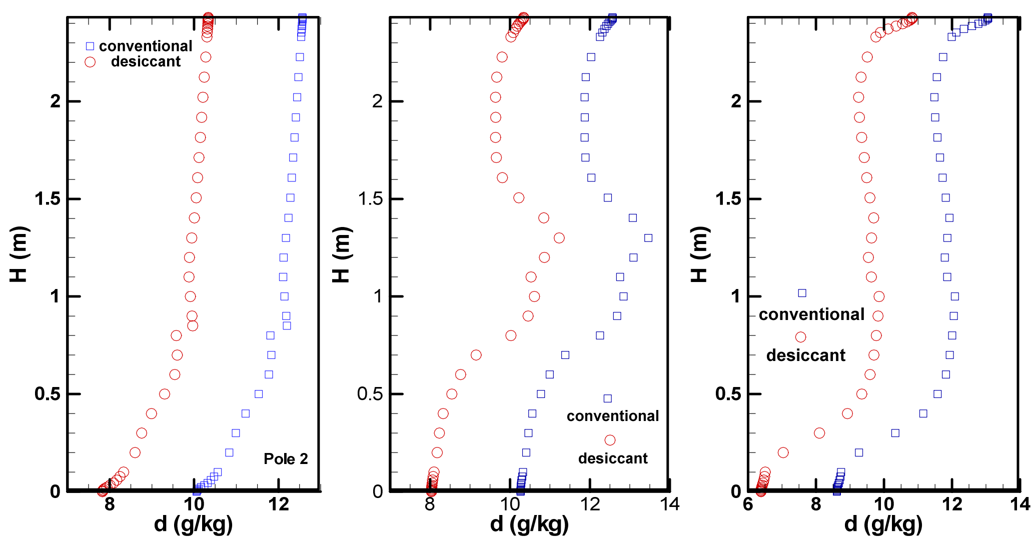
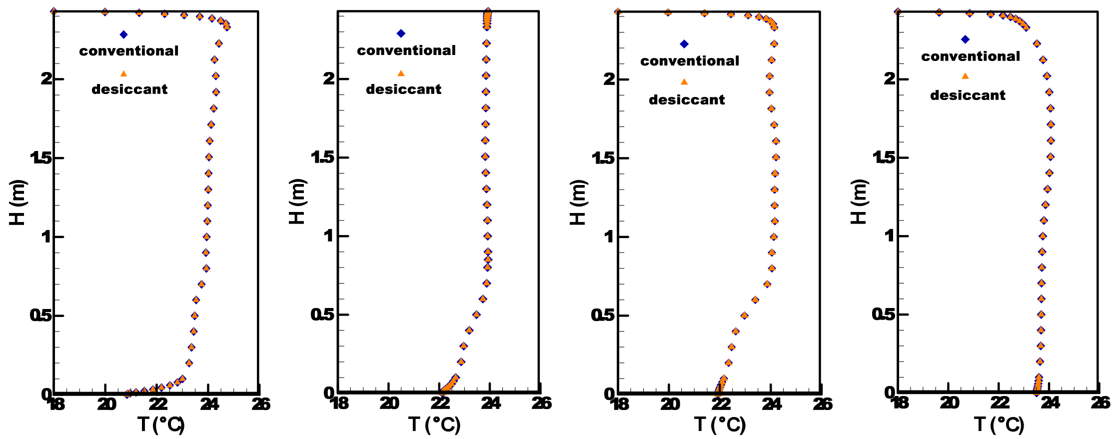


Fig.7 Thermal plumes generated by occupants in the conference room.



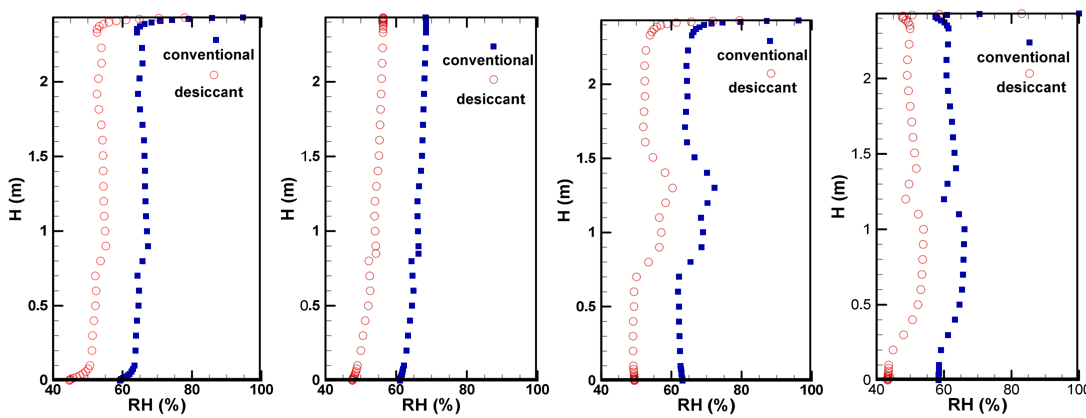
(a) Pole 2 **(b) Pole 3** **(c) Pole 5**

Fig.8 Vertical profiles of humidity ratio in several representative locations of the room.



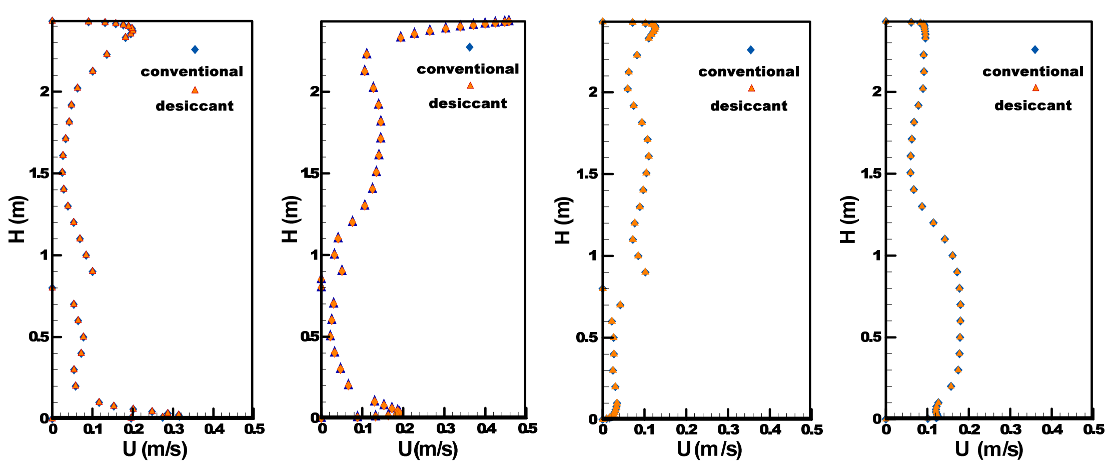
(a) Pole 1 (b) Pole 2 (c) Pole 3 (d) Pole 4

Fig.9 Vertical profiles of air temperature in several representative locations of the room.



(a) Pole1 (b) Pole 2 (c) Pole 3 (d) Pole 5

Fig.10 Vertical profiles of relative humidity in several representative locations of the room.



(a) Pole 1 (b) Pole 2 (c) Pole 3 (d) Pole 4

Fig.11 Vertical profiles of air velocity in several representative locations of the room.

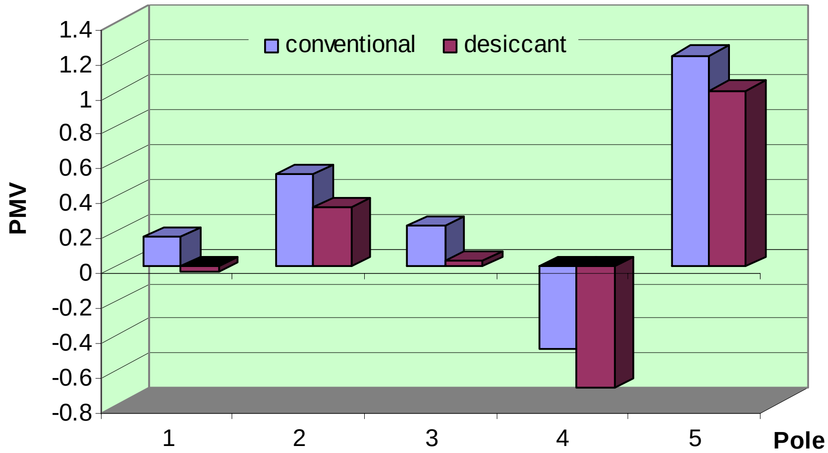


Fig.12 PMV at 1.1 m above the floor in the five locations in the room.