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Air Change Effectiveness in Laboratory Tests of Combined Chilled Ceiling and Displacement Ventilation

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SUMMARY (150 words)
Radiant chilled ceilings (CC) with displacement ventilation (DV) represent a promising integrated system design that combines radiant systems energy efficiency with the opportunity for improved ventilation performance typical of DV. The purpose of this study was to conduct laboratory experiments to investigate how the radiant surface temperature and the ratio of cooling load removed by CC over the total cooling load affects the air change effectiveness (ACE) measured according to ASHRAE Standard 129. Three experiments were carried out in a climatic chamber equipped with radiant panels covering 36.7% of the suspended ceiling, representing a typical installation of metal radiant panels. The results showed that ACE higher than one is maintained in the occupied zone even when more than half (54 percent) of the heat load is removed by a CC and the radiant surface temperature is 18.7°C.

IMPLICATIONS (50 words)
This study shows the ability of displacement ventilation to reduce the exposure to contaminants generated by heat sources even when coupled with a typical radiant chilled ceiling configuration, therefore the two systems combine the advantages of low energy consumption and high ventilation effectiveness.

KEYWORDS
Contaminant stratification, displacement ventilation, air temperature stratification, radiant panel, ventilation effectiveness.

INTRODUCTION

Displacement ventilation (DV) is a method of room air distribution that could provide improved indoor air quality for contaminants emitted by heat sources compared to the dilution ventilation provided by overhead mixing systems. In a DV system, which is applied mainly for cooling purposes, air is supplied at low velocity through supply devices located near floor level (the most common are low side wall diffusers), and is returned near ceiling level. The ASHRAE (Chen and Glicksman, 2003) and the REHVA (Skistad et al., 2002) methods are the most commonly used references for the design and operation of DV systems. Supplying cool air at floor level in a stratified environment may cause local thermal discomfort due to draft and excessive temperature stratification (ASHRAE Standard 55, 2010). Hydronic-based radiant systems are associated with energy savings, therefore there is strong interest in combining hydronic systems with the indoor air quality benefits of DV. There are two types of chilled ceiling designs: (a) radiant ceiling panels; and (b) thermally activated building systems (TABS) also known as hydronic or active slab. In this paper we focus on radiant ceiling panels. Radiant ceiling panels have a fast response time, thus they are easy to control and are able to adapt to rapidly changing loads, and they are relatively easy to design. They can also be used in retrofit applications, and are compatible with conventional suspended ceiling systems. The main drawbacks are related to the cost, the inability to store heat (peak-shave) and their low operating mean water temperature requiring thoughtful space dew point control to avoid condensation. Radiant ceiling panels, due to the high cost, usually cover less ceiling area than TABS.

Novoselac and Srebic (2002) did an extensive critical literature review of the performance and design of a combined chilled ceiling and displacement ventilation system. According to Novoselac and Srebic (2002) one of the key parameters of the design is the cooling load split between the CC and DV system. Tan et al. (1998) defined η as the ratio of the zone cooling load removed by the chilled ceiling to the total room cooling load. η may vary between zero and one. If η equals one, it means that a pure CC system is used. On the other hand, if η equals zero, a pure DV system is used. Reducing the amount of the cooling load removed by DV, i.e. increasing η, implies the likelihood of reduced stratification in the room, and this in turn implies a reduction in the ventilation effectiveness of the system. Schiavon et al. (2011), by changing the amount of chilled ceiling (active) area (from 73.5 to 36.7%) showed that average radiant surface temperature is a stronger predictor of air temperature stratification performance and they underlined that the design methods summarized in Novoselac and Srebic (2002) were developed for high percentage of active ceiling area typical of TABS and not for radiant panels. Schiavon et al. (2011) focused only on air temperature stratification and did not report air change effectiveness (ACE) results. ACE is a measure of the effectiveness of outdoor air distribution to the breathing level within the ventilated space (ASHRAE, 2002).

The purpose of this study is to conduct laboratory experiments for a typical U.S. interior zone office configuration with a CC/DV system to investigate how the radiant surface temperature and the ratio of cooling load removed by CC over the total cooling load affects the air change effectiveness.

METHODS

Experimental facilities and room description

The experiments were carried out in a climatic chamber (4.27 m x 4.27 m x 3.0 m) equipped with radiant panels located in a suspended ceiling placed at a height of 2.5 m above the floor. The climatic chamber is located within a large conditioned test hall. The area of the climatic chamber is 18.2 m^2. A detailed description of the room is reported in Schiavon et al. (2011).

The aluminum radiant panels installed in the suspended ceiling are 1.83 m long and 0.61 m wide (area equal to 1.11 m^2). Copper pipes are thermally connected to aluminum channels in panels with a spacing of 0.15 m. Cotton fiber insulation was present over the panels (2.288 m^2K/W). Six panels were used (6.7 m^2 of the ceiling equals 36.7 % of the ceiling area) and configured in series as shown in Figure 1. Figure 2 shows the locations of the two simulated workstations, typical office heat loads, and instrument station for measuring the vertical temperature profile. The inlet air was supplied to the room from a 0.6 m tall corner mounted displacement diffuser with a radius of 0.46 m. Heat sources are summarized in Table 1. Office heat sources were modeled using a floor-mounted tower computer, flat screen and desk lamp on the desk, and overhead lighting. Occupants were simulated with heated thermal manikins according to EN 14240 (2004). These simulators represent a load on the space by using light bulbs enclosed in a sheet metal cylinder. They try to match the radiant convective split of a person by using high emissivity paint and holes to allow air to pass through.

Measuring instruments and uncertainty

The air and water temperature and flow sensors, location and uncertainty are reported in detail in Schiavon et al. (2011). The air temperature accuracy was ±0.15°C or better. The water temperature accuracy was ±0.045°C or better. A vertical tree was used to measure air temperatures at seven heights (0.1, 0.25, 0.6, 1.1, 1.7, 1.9, 2.4 m) at the instrument station in the room (see Figure 2). All air temperature sensors were shielded against radiant heat transfer using a fabricated mylar cylinder. At this same location the mean radiant temperature was measured at the 0.6 m height with a black-globe thermometer. The black-globe thermometer fulfills the requirements of ISO 7726 (1998). The displacement ventilation airflow rate accuracy was better than ±3% of the reading. The cooled water mass flow rate accuracy was ±0.02% of the reading. Carbon dioxide (CO_2) was used as tracer gas. All CO_2 probes were calibrated using a two point calibration method. The first point was measured at 0 ppm of CO_2 and the second point was measured at 5050ppm of CO_2. The new calibration data was uploaded to each individual probe and a spot check was done using 2460 ppm CO_2. The CO_2 sensors as starting condition should measure the same values of CO_2. The sensors differed of less than ±30 ppm. The measured values have been corrected to take this difference into account. CO_2 sensor were located in the supply diffuser, in the exhaust and at three heights in the room (0.6; 1.1 and 1.7 m) at the CO_2 sensor tree (see Figure 2).

Table 1. Heat load summary

Heat source	Power [W]	Power per floor area [W/m^2]
Workstation 1	215	11.8
Workstation 2	231	12.7
Overhead lights	145	8.0
Sensors & DAQ	40	2.2
Total	631	34.7

Table 2. Experimental summary

Test #	Airflow rate [L/s]	Predicted η
Test 1	73.3	0
Test 2	60.4	0.25
Test 3	43.1	0.50

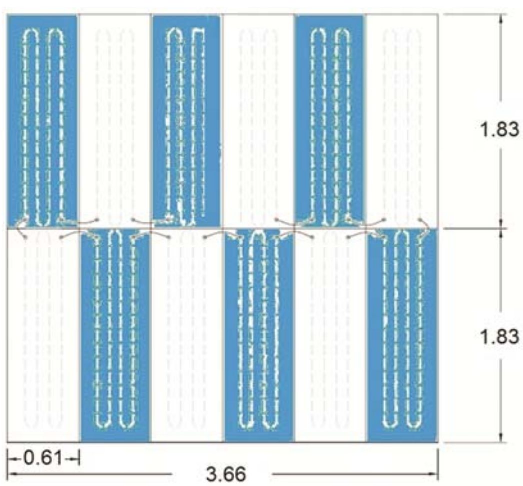

Figure 1. Active radiant panels (blue) configuration.

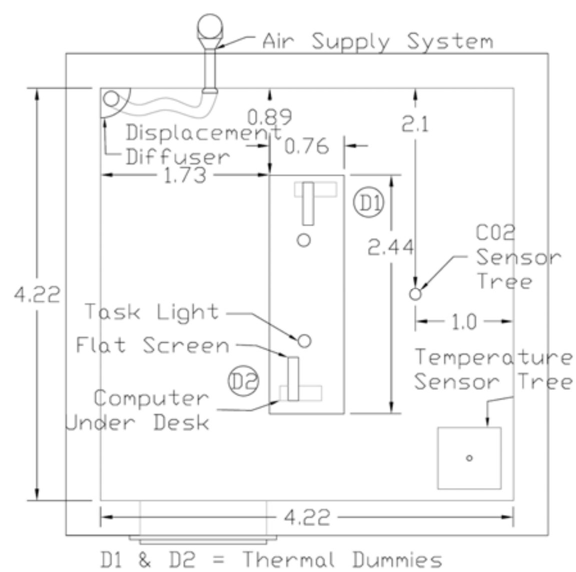

Figure 2. Layout of test chamber.

The data are analyzed in accordance with the ISO guideline (1993) for the expression of uncertainty. The sample uncertainty of the derived quantities (air and water temperature differences, cooling load removed by the panels, electrical load, and η (see definition below)) has been evaluated. The derived uncertainty of the air temperature difference is ±0.41°C, the water temperature difference is ±0.125°C, the cooling load removed by the chilled ceiling is ±25.5 W, the electrical total power is ±14.7 W), and η is ±0.04. The level of confidence is 95 percent. It has not been possible to calculate the uncertainty for the calculated ACE values.

Experimental conditions and procedure

η (eta) is the ratio of the cooling load removed by chilled ceiling over the total cooling load. The total cooling load is equal to the electrical power of the heat sources because the measurements were done in steady state conditions in an almost adiabatic room, thus the heat gains are equal to the cooling loads. The experimental test conditions are summarized in Table 2 and described in the following. The heat load in the room was kept constant and equal to 631 W (34.7 W/m^2). The heat loads are described in Table 1. The operative temperature was kept constant and almost equal to 24°C. The operative temperature was calculated as the average of the mean radiant temperature and the averaged seated air temperature according to ISO 7726 Annex G (1998). The averaged seated air temperature was the mean value of the air temperatures measured at 0.1, 0.6 and 1.1 m. The mean radiant temperature was measured at 0.6 m. The DV supply air temperature was kept constant and equal to 18°C. The air, water and mean radiant temperatures, the cooled water mass flow rate, and air flow rate were recorded for at least 30 min after steady-state conditions were obtained. The electrical power consumption was manually recorded before starting the experiments. Measurement of air change effectiveness was performed for all tests according to the tracer gas step-up method of ASHRAE Standard 129 (2002).

RESULTS

Figure 3 presents measured CO_2 concentrations vs. time for (a) η = 0, 100% DV; (b) η = 25%; and (c) η = 54%. Measurements are reported for supply, exhaust, and three heights in the room (0.6, 1.1, and 1.7 m). The reported concentrations have been adjusted with respect to outdoor average concentration. All three tests show variations in CO_2 concentration with height, indicating that preferential ventilation is maintained at lower heights in the room. Figure 3a shows that it was not possible to keep the CO_2 emission in the room constant, this is represented as a gradual reduction of the CO_2 concentration over time at the exhaust after steady-state is reached; this test does not comply with ASHRAE 129 (2002). Figure 3d shows the measured vertical temperature distribution in the room for the three tests. The results demonstrate how stratification is reduced in the upper portion of the room as η is increased. In fact, the temperatures suggest a well-mixed upper region in the room that increases in depth as more of the heat load is removed by the chilled ceiling. However, temperature stratification and ACE higher than one are both maintained in the lower region, corresponding to the occupied zone. The main performance parameters of the displacement ventilation and chilled ceiling systems obtained in the experiments are summarized in Table 3.

Table 3. Experimental summary

Test #	Calculated η [-]	Radiant panel surface temp. [°C]	Cooling load removed by CC [W]	ACE at 0.6 m [-]	ACE at 1.1 m [-]
Test 1	0	24.9	0	2.45	1.48
Test 2	0.25	22.6	160	1.75	1.23
Test 3	0.54	18.7	341	1.63	1.37

The operative temperature was controlled at 23.9°C, therefore we may conclude that the comparison was done with almost thermally equal comfort conditions (air velocity and relative humidity were constant as well). ACE was measured at a location far from thermal plumes. For a real occupant even if the breathing zone is roughly at 1.1 m, he/she would breathe air taken from his/her own thermal plume originating from a lower level (e.g., 0.6 m).

All ACE values are higher than one (mixing ventilation) and they decrease with the increase of η and the reduction of the ceiling radiant temperature, in particular at 0.6 m.

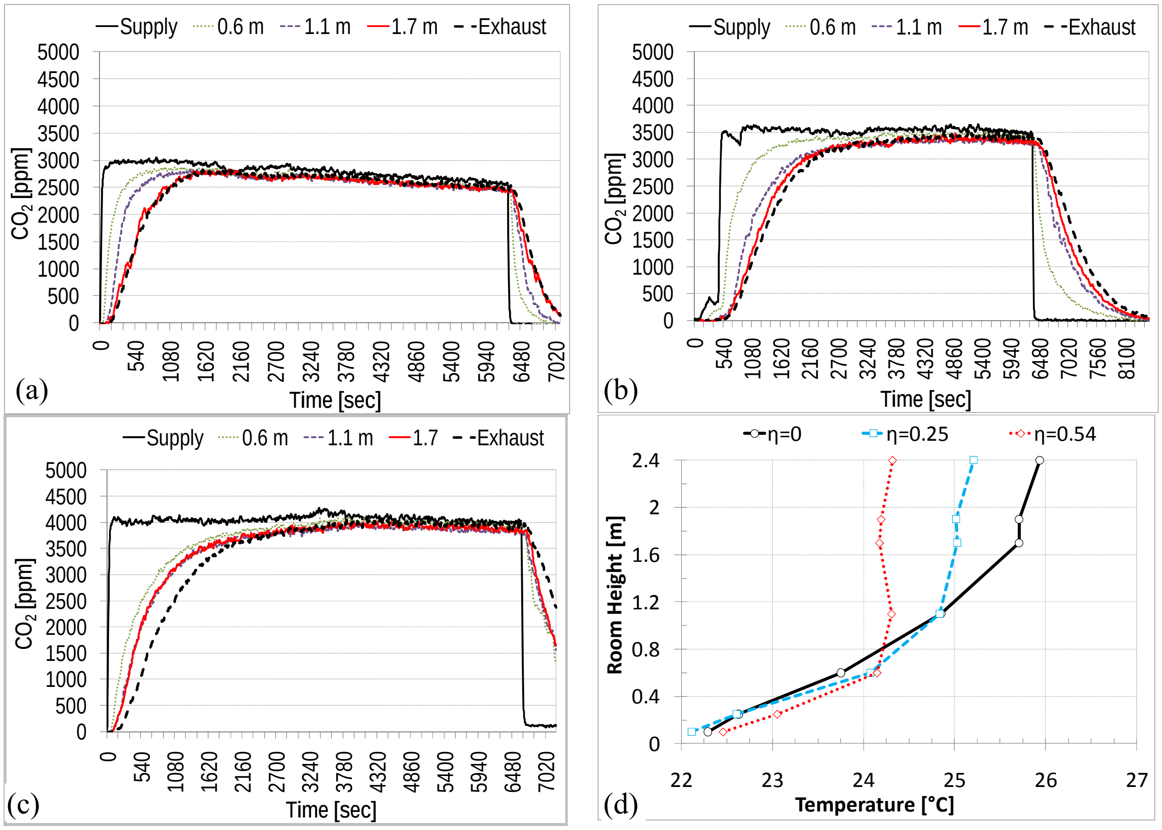

Figure 3. For (a) (b) and (c): CO_2 concentrations for the step-up method at the supply, exhaust and at 0.6, 1.1 and 1.7 m for: (a) test 1 where there is only DV; (b) test 2 where the radiant panels remove 25% of the load; and (c) test 3 where the radiant panels removed 54% of the load. (d) Air temperature profiles for the three tests described in Table 2.

DISCUSSION

The key finding from this study demonstrates that improved air change effectiveness (compared to a well-mixed system) is maintained in the lower occupied region of the room for a stratified displacement ventilation system, even when more than half (54%) of the heat load is removed by a chilled radiant ceiling and the radiant panel surface temperature is higher than 18.7°C. The results indicate that ACE and stratification are reduced compared to the 100% DV case (Test 1), as more heat load is removed by the chilled ceiling and its surface temperature is reduced. Due to the limited number of tests and the lack of compliance of test 1 with ASHRAE 129 (2002) it is not possible to develop a predictive model. While previous results (Behne, 1999 and Tan et al. 1998) for CC/DV systems have focused on systems with all or close to 100% of the ceiling serving as the active radiant chilled surface (typical of TABS), the present study is representative of a radiant panel installation in a suspended ceiling configuration, typically covering 30-50% of the ceiling area. The reduced active panel area requires a lower surface temperature to remove the same amount of load. Previous stratification results reported by Schiavon et al. (2011) suggest that ACE higher than one may be extended to higher values of η, but further testing is needed to determine the maximum limit before improved ACE is destroyed by the effect of the chilled ceiling.

CONCLUSIONS

There is great interest in improving the energy efficiency of space conditioning systems while still maintaining a high quality indoor environment. In this study, a laboratory experiment was conducted in a typical office space with displacement ventilation (DV) to investigate how air change effectiveness (a known benefit of stratified DV systems) is affected by the addition of a low-energy radiant chilled ceiling (CC) system. The results are promising for the combination of these two technologies. This paper describes test data from a chamber equipped with radiant panels covering 36.7% of the suspended ceiling, representing a typical installation of metal radiant panels. Due to the smaller active chilled ceiling surface area in panel systems, relatively cooler surface temperatures are required to remove the same amount of heat load compared to TABS (close to 100% active area). Therefore, it is quite likely that in a building with TABS, the disruption of stratification and improved ACE as the amount of load removed by the chilled ceiling (η) is increased will be no worse than the results presented in this paper.

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