TEMPERATURE STRATIFICATION AND AIR CHANGE EFFECTIVENESS IN A HIGH COOLING LOAD OFFICE WITH TWO HEAT SOURCE HEIGHTS IN A COMBINED CHILLED CEILING AND DISPLACEMENT VENTILATION SYSTEM

Stefano Schiavon, Ph.D., P.E.
Center for the Built Environment,
University of California at Berkeley

Fred Bauman, P.E.
Center for the Built Environment,
University of California at Berkeley

Brad Tully, P.Eng.
Price Industries
General Manager,
Price Mechanical West

Julian Rimmer, P.Eng.
Price Industries
Senior Product Manager,
Sustainable Technologies

ABSTRACT

Radiant chilled ceilings (CC) with displacement ventilation (DV) represent a promising integrated system design that combines the energy efficiency of both sub-systems with the opportunity for improved ventilation performance resulting from the thermally stratified environment of DV systems. Their combined cooling capacity is thought to be limited. The purpose of this study is to conduct laboratory experiments for a U.S. interior zone office with a very high cooling load (91.0 W/m^2) and with two different heat source heights represented by computer CPUs (at floor level and at 1.52 m) to investigate their influence on room air stratification and air change effectiveness. The experiments were carried out in a climatic chamber equipped with 12 radiant panels, covering 73.5% of the ceiling, installed in the suspended ceiling. The cooling load removed by the panels varied between 0 and 92 W/m^2 (based on radiant panel area) or between 0 and 68 W/m^2 (based on room area). The average mean water temperature of the panels varied between 14.1- 26.2°C. The displacement ventilation airflow rate varied between 4.0 and 9.9 l/(s m^2), and the supply air temperature was kept constant at 18°C. The results showed that displacement ventilation and chilled ceiling are able to provide a stable thermal stratification and improved ventilation effectiveness compared to mixing ventilation for a wide range of configurations and system design even for extremely high cooling load (91 W/m^2). Stratification and air change effectiveness decreases when a larger portion of the cooling load is removed by the chilled ceiling (surface temperature of the panel decreases). For every degree decrement of the panel the stratification decreases by 0.13°C and the ACE by 0.13. Moving the CPUs (representing 51% of the total room heat gain) from the floor level to 1.5 m height markedly increased the room median stratification (0.8°C) and the median air change effectiveness measured at 0.6 m (1.75). Therefore, increasing the height of heat sources reduced energy use and improved indoor air quality. When the CPUs were located in the higher location, the median stratification in the occupied zone was 2.95°C and the ACE at 0.6 m was 2.9. Moreover, it was found that the higher the stratification the better the air change effectiveness.

KEYWORDS

Displacement ventilation; Chilled ceiling; Air vertical temperature stratification; Radiant panel; thermally activated building system (TABS); contaminant stratification; high loads; office space design

INTRODUCTION

Displacement ventilation (DV) is a method of room air distribution that can provide improved indoor air quality for contaminants emitted by heat sources (ventilation performance) 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 very low velocity through supply devices located near floor level (the most common are low side wall diffusers), and is returned near ceiling level. A displacement flow pattern can also be obtained with horizontal discharge (low throw) floor diffusers in underfloor air distribution (UFAD) systems. The ASHRAE [1] and the REHVA [2] 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 [3]. Hydronic-based radiant systems are associated with energy savings [4-6] even if sometimes problems could arise [7], therefore there is strong interest in combining hydronic systems with the indoor air quality benefits of DV.

A review of the literature about displacement ventilation and radiant chilled ceiling until 2010 is reported in [8] . A short summary of the literature review and updates based on recently published papers are reported hereafter. The combination of chilled floor and DV was described in Causone et al. [9] . They concluded that the combination of DV with floor cooling, under a typical European office

room layout, may cause the air temperature difference between head and ankles to exceed the comfort range specified by ASHRAE Standard 55 [3]. They also noticed that, by increasing the air flow rate and thus raising the floor temperature, the vertical air temperature differences decreased. They also showed that the draft risk did not increase significantly. From the indoor air quality point of view they showed that the presence of the chilled radiant floor does not affect the contaminant removal effectiveness (a.k.a. ventilation effectiveness in Europe) of the DV system.

The combination of chilled ceiling (CC) and DV is more attractive for U.S. markets. There are two types of chilled ceiling designs: (a) radiant ceiling panels; and (b) thermally activated building systems (TABS) also known as hydronic slab. Radiant ceiling panels have several advantages: they have a fast response time, thus they are easy to control and are able to adapt to rapidly changing loads, they are relatively easy to design and the technology is well known. 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. TABS, usually fabricated as hydronic tubing embedded in slabs, are less expensive than radiant panels, have the ability of peak shaving and shifting, and usually operate at higher cooling temperatures, reducing the condensation risk. The main drawbacks are related to the complexity of the design and control, and the slow response of the thermally massive slab to the changing cooling loads [10] .

Alamdari et al. [11] described how adding CC to a DV system influences the air distribution characteristics of DV. Rees and Haves [12] developed a nodal model to represent room heat transfer in DV and CC systems that is suitable for implementation in an annual energy simulation program, but it cannot be applied as a stand-alone design tool. Novoselac and Srebic [13] did an extensive critical literature review of the performance and design of a combined chilled ceiling and displacement ventilation system and concluded that one of the key parameters of the design is the cooling load split between the CC and DV system. Tan et al. [14] 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 0 and 1. If η equals 1, it means that a pure CC system is used. On the other hand, if η equals 0, a pure DV system is used. Tan et al. [14] suggested that, to maintain a temperature gradient of at least 2°C/m, the DV system should remove a minimum of 33% of the cooling load (i.e., η = 0.67). Behne [15] stated that good thermal comfort and air quality could be maintained when the DV system removes at least 20-25% of the total cooling load.

Gheddar et al. [16] developed general design charts for sizing the CC/DV systems using a simplified plume-multi-layer thermal model of the conditioned space developed by Ayoub et al. (2006). The model developed by Ayoub et al. [17] was compared to CFD simulations. The main limitation of the method is related to the fact that the design charts were developed for a 100% ceiling coverage factor. A sensitivity analysis has been performed for 80% ceiling coverage factor. There are no data for lower ceiling coverage factors. Keblawi et al. [18] expanded Gheddar et al. [16] to operating sensible load ranges from 40 W/m^2 to 100 W/m^2. The model relates system load and operational parameters with comfort measured by vertical temperature gradient and indoor air quality measured by the stratification height.

Kanaan et al. [19] developed and experimentally tested a simplified model to predict carbon dioxide transport and distribution in rooms conditioned by CC and DV. Chakroun et al. [20] extended the model to transient conditions and applied it to study the energy savings potential during the cooling season for a simplified room (25 m^2) located in the Kuwait climate. To perform the energy simulation they used an algorithm developed internally to their research group.

Schiavon et al. [8] experimentally investigated the influence of percentage of ceiling active area and of the split of cooling load between displacement and chilled ceiling on stratification. It was found that the average radiant ceiling surface temperature is a better predictor of the temperature difference between the head (1.1 m) and ankle (0.1 m) of a seated person in the occupied zone compared to other parameters related to the fraction of the total cooling load removed by the radiant chilled ceiling. This result accounts for the fact that when smaller active radiant ceiling areas are used (e.g., for a typical radiant ceiling panel layout), colder radiant surface temperatures are required to remove the same amount of cooling load (as a larger area), which cause more disruption to the room air stratification. We also found that the room air stratification in the occupied zone (1) decreases as a larger portion of the cooling load is removed by the chilled ceiling, (2) increases with higher radiant ceiling surface temperatures, and (3) decreases with an increase in the ratio between the total cooling load and the displacement airflow rate. These results confirmed the ones summarized in [13] . We concluded that despite the impact that the chilled ceiling has on stratification, the results indicate that a minimum head-ankle temperature difference of 1.5°C in the occupied zone (seated or standing) will be maintained for all radiant ceiling surface temperatures of 18°C or higher.

Ventilation effectiveness is an indicator of the efficiency with which fresh air is delivered to the breathing zone in ventilated rooms and it is related to indoor air quality. It is a representation of how well a considered space is ventilated compared to a uniform wellmixed room

[21] . In the U.S. ventilation effectiveness is measured with the index named Air Change Effectiveness (ACE) according to the ASHRAE Standard 129 [22]. However, Rim and Novoselac [21] questioned the overall ability of ACE as an indicator of air quality and human exposure. With climatic chamber experiments and a calibrated CFD model they showed that for fine particles (1 μm), an increase in ACE reduces occupant exposure, while for coarser particles (7 μm), source location and airflow around the pollutant source are the major variables that affect human exposure. It is important to keep these findings in mind with the application of displacement ventilation, where pollutant sources located at floor level near an occupant could be drawn up to the breathing level by the rising thermal plume. In our previous work [23] we reported three ACE tests and we concluded that ACE higher than one is maintained in the occupied zone even when more than half (54%) of the heat load is removed by a CC and the radiant surface temperature is 18.7°C.

The purpose of this study is to conduct laboratory experiments for a U.S. interior zone office with high cooling load (91.0 W/m²) and with two different heat source heights, represented by computer CPUs (at floor level and at 1.52 m), to investigate their influence on room air stratification and air change effectiveness.

METHOD

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 laboratory space. The area of the climatic chamber is 18.2 m² and the volume is 54.7 m³. The room has no windows. The walls, the ceiling and the floor have similar construction and thermal properties. Starting from the exterior, the chamber wall is comprised of 3.522 m²K/W insulation, a stagnant 0.102 m air gap (0.352 m²K/W), aluminum extruded walls, and another layer of 0.102 m of polyurethane board (3.522 m²K/W). By adding up this assembly, the overall transmittance is 0.135 W/m²K.

The aluminum radiant panels installed in the suspended ceiling are 1.83 m long and 0.61 m wide (area equal to 1.11 m²). Copper pipes are thermally connected to aluminum channels in panels with a spacing of 0.15 m. The suspended ceiling is composed of radiant ceiling panels connected in series. Cotton fiber insulation was present over the panels (2.288 m²K/W). Twelve panels were used (13.4 m² of the ceiling equals 73.5 % of the ceiling area). Figure 1 shows the locations of the four simulated workstations, office heat loads, measuring station for recording the vertical temperature profile, CO_2 measuring tree and location of the globe thermometer. The inlet air was supplied to the room from a 1.2 m tall semi-circular wall-mounted displacement diffuser (radius = 0.6m). Heat sources are summarized in Table 2. Office heat sources were modeled using tower CPUs (computer processing units; sometimes referred to as PCs, or personal computers), representing 51% of the total heat gain, flat screens and desk lamps on the desks, and overhead lighting. A portion (36%) of the heat gains generated by the tower CPUs were generated with electrically heated 0.35 m by 0.35 m galvanized steel plates. The plates (2 mm thick) are heated by two silicon rubber strip heaters mounted with high temperature room temperature vulcanizing adhesive. The convective/radiative split and surface temperature of these plates are similar to those of tower computers. Occupants were simulated with heated thermal manikins according to EN 14240 [24]. 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. When fully installed, the test chamber represented a 4-person office with multiple computers (high heat gain) at each workstation.
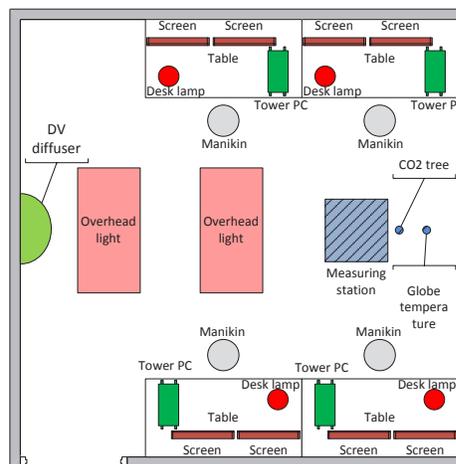

Figure 1. Layout of test chamber. All dimensions are in meters.

Measuring instruments and uncertainty

The air temperatures were monitored continuously with resistive thermal devices PT 100. The sensors were calibrated prior to the measurements. The obtained accuracy was ±0.15°C or better. The supply and return water temperatures, $t_{w,s}$ and $t_{w,r}$ were monitored continuously with resistive thermal devices PT 100. The sensors were calibrated prior to the measurements. The obtained accuracy was ±(0.03+0.0005·t_w), for the range of measured values the accuracy was ±0.045°C or better. The electrical power was measured with a power harmonic analyzer. The DV supply air temperature, $t_{air,s}$ was measured inside the diffuser. The exhaust air was leaving the room through a slot in the suspended ceiling and finally leaving the return plenum through a duct going out into the surrounding hall. The exhaust air, $t_{air,r}$ was measured in that duct. 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 1). All air temperature sensors were shielded against radiant heat transfer using a fabricated mylar cylinder. The globe temperature was measured at 0.6 m height with a black-globe thermometer. The black-globe thermometer fulfills the requirements of ISO 7726 [25], and the same standard was used to calculate the mean radiant temperature from the globe temperature. The displacement ventilation airflow rate, V_{air} was measured with a calibrated plate orifice having an accuracy of better than ±3% of the reading. The cooled water mass flow rate, m_w, was measured with a high quality Coriolis mass flow meter with an accuracy of ±0.02% of the reading. The data are analyzed in accordance with the ISO guideline [26] 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. When presented, the uncertainty is indicated by means of error bars. The level of confidence is 95% (coverage factor 2).

Carbon dioxide (CO_2) was used as the tracer gas for ACE measurements. 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 5050 ppm of CO_2. The new calibration data was uploaded to each individual probe and a spot check was done using 2460 ppm CO_2. CO_2 sensors 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 1). The stepup method according to the ASHRAE Standard 129 [22] was used and the measurements comply with its requirements.

Experimental conditions and procedure

η (eta) is the ratio of the cooling load removed by chilled ceiling, CL_{CC}, over the total cooling load and is expressed by the following equation:

$$\eta = \frac{cooling\ load\ removed\ by\ CC}{total\ cooling\ load} = \frac{CL_{CC}}{CL_{DV} + CL_{CC}} \qquad (1)$$

The total cooling load is equal to the electrical power of the heat sources because the measurements were done in steady state conditions, thus the heat gains are equal to the cooling loads. The cooling load removed by the radiant panels, CL_{CC}, has been calculated with the following formula:

$$CL_{CC} = m_w\, c_{p,w}\, (t_{w,r} - t_{w,s}) \qquad (2)$$

where the $c_{p,w}$ is the specific heat capacity of water. The cooling load removed by DV, CL_{DV}, was calculated indirectly as the difference between the total cooling load and the cooling load removed by the radiant ceiling panels. The cooling load removed by DV could also be calculated directly by measuring the airflow rate and the supply and return air temperature. This procedure was not used because the accuracy of the water flow sensor was much higher than that of the airflow rate sensor.

The experiments are summarized in Table 1. The experiments are identified based on a first order estimation of the airflow rate measured in L/s, the temperature setpoint (where "F" stand for "Free to change") and the location of the heat sources ("F" for floor and "H" for "at 1.52 m Height above the floor). The heat load in the room was kept constant and equal to 1657 W (91.0 W/m²). The heat loads are described in Table 2. The operative temperature, t_{op}, was kept constant and almost equal to 24°C, except in tests 140-FH and 75-F-H. The operative temperature was calculated as the average of the mean radiant temperature (0.6 m height) and the average seated air temperature according to ISO 7726 annex G [25]. The average seated air temperature was the mean value of the air temperatures measured at 0.1, 0.6 and 1.1 m. In a stratified environment there is no single height where the air temperature can be measured that represents the "perceived" air temperature. For this reason, the average of the air temperatures measured at the ASHRAE Standard 55 [3] heights was used. The DV supply air temperature, $t_{air,s}$, was kept constant and equal to 18°C. In order to keep the operative temperature setpoint equal to 24°C, the water mass flow rate and the cold water supply temperature were manually adjusted. In the experiments 140-F-H and 75-F-H the air and water flow rates and the supply air and water temperatures were kept constant and equal.

to the case 140-24-H and 75-F-H to study the influence on the air change effectiveness and thermal stratification of just moving the computer heat sources up to a higher part of the room. 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.

Table 1. Experimental tests summary.

Test	Airflow rate [L/s]	η^1	Operative temperature [°C]	PCs location
180-24-F	181.4	0.20	24	Floor
160-24-F	163.2	0.24	24	Floor
140-24-F	138.2	0.47	24	Floor
120-24-F	117	0.57	24	Floor
95-24-F	94.5	0.64	24	Floor
75-24-F	72.4	0.73	24	Floor
35-24-F[2]	36.6	0.89	24	Floor
130-24-H	131.6	0	24	At 1.52 m
100-24-H	102.3	0.34	24	At 1.52 m
75-24-H	74.4	0.57	24	At 1.52 m
140-F-H	142.2	0.49	Free to change	At 1.52 m
75-F-H	75.3	0.75	Free to change	At 1.52 m

[1] This parameter has been calculated after performing the experiment.

[2] The total power was 1803 W and not 1657 W, as in the other experiments, because we needed to add an extra pump in the room, above the radiant panels, to increase their water flow rate. In the calculation of η we included the power of the pump. If the pump was not included, then η would have been equal to 0.97. For this test it was not possible to perform the ACE test due to time constraints.

The CPUs are equal to 51% of the total heat gains and 71% of the heat gains coming from the office equipment (screen and CPUs). Screens cannot be moved from the desk, but the location of the CPUs is flexible. They are often located on the floor under the desk. We tested two locations, the first one (named "floor" or "F") in which the tower CPUs were located at floor level under the desk, and the second one in which they were placed on open shelves above the desks at 1.52 m (5 feet) above the floor.

Table 2. Heat load summary.

Heat source	Number	Power per unit [W]	Total power [W]	Power per floor area [W/m²]
CPUs	4	212	848	46.6
Screens and lamps	8	44.25	354	19.5
People	4	75	300	16.5
Instrument tree and datalogger	1	20	20	1.1
Overhead Lighting	2	67.5	135	7.4
Total			**1657**	**91.0**

The tests summarized in Table 1 were performed in June 2012. In the results and discussion sections results from previous CC/DV testing in the same lab will also be reported [8] . To verify consistency between separate lab tests, the experiment without radiant panels (only displacement ventilation) was repeated and compared for all the visits. The temperature profiles were found to be very similar. The average of air temperature differences between the cases calculated at each height was 0.30°C.

RESULTS

The main performance parameters of the displacement ventilation and chilled ceiling systems obtained in the experiments are summarized in Table 3. The operative temperature for the first ten experiments was controlled within the range of 24.0-24.2°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). The DV supply air temperature was precisely controlled at 18°C. The airflow rate varied between 36.6 to 181.4 L/s [2.4 – 11.8 air changes per hour].

Temperature stratification

The vertical air temperature profiles are shown in Figure 2. Figure 2a shows the temperature stratification when the PCs are located at the floor level below the desks. From part "a" of the figure it can be deduced that the temperature stratification in the occupied zone for a seated person (up to 1.1 m height) is not strongly affected by the change in the cooling load split between displacement ventilation and chilled ceiling. The stratification is reduced from 2.1°C to 0.8°C when the airflow is reduced from 181.4 L/s (η=0.20) to 36.6 L/s (η=0.89). At higher heights in the room, it can be seen that temperature stratification is reduced as the amount of load removed by the chilled ceiling increases. The suspended ceiling is located at 2.5 m from the floor. Figure 2 reports the air temperatures from floor to the suspended ceiling; between the suspended ceiling and the exhaust there is a void space. When the panels are activated, i.e. cooled, the exhaust air, $t_{air,r}$, is cooler than the temperature measured at 2.4 m by the panels. Figure 2a shows that most of the temperature stratification is occurring in the occupied zone. The relatively well mixed conditions (small temperature differences) at higher heights in the room is a good indication that these points fall above the stratification height that separates the two characteristic lower and upper zones of a stratified displacement ventilation system. Experiment 35-24-F was not fully successful. The aim of this experiment was to test the combination of DV and CC in extreme conditions, with the CC taking almost 90% of the load and providing only 36.6 L/s (that is a bit more than double of the minimum outdoor air flow rate (15.5 L/s) according to ASHRAE 62.1[27] for an office space). In order to obtain the operative temperature equal to 20°C the water supply temperature was reduced to 9.7°C (mean water temperature was 10.9°C), which is too low for almost any real application. Even at 9.7°C we were not able to obtain the desired operative temperature and we increased the mass flow rate from 419 kg/h to 575 kg/h. In order to do this we added an extra pump in the room, above the radiant panels. The obtained temperature profile was correct, but we were not able to perform an ACE test due to time constraints.

Figure 2b shows the temperature stratification when the PCs are located at 1.52 m above the floor. The effect is dramatic. After a lower layer from 0 to 0.6 m, where the air is relatively well-mixed, there is a strong stratification between 0.6 and 1.7 m. There are two groups of profiles. The ones on the left (dotted lines) when the temperature in the room was allowed to fluctuate, and the group with solid lines where the average operative temperature in the occupied zone was maintained at 24°C.

In only two cases (100-24-H and 75-24-H) was the vertical temperature difference between head (1.1 m) and ankle (0.1 m) for seated occupancy observed to exceed 3°C, the maximum acceptable stratification specified by ASHRAE Standard 55 [3]. In both these cases the CPUs were in the higher part of the room. CC/DV systems even with high cooling loads are able to maintain stratification lower than 3°C, if more than 50% of the heat gains are in the lower part of the room. In applications of CC/DV to spaces with stratification approaching 3°C, it is advisable to remove a high enough percentage of the total load by the chilled ceiling to maintain stratification at acceptable levels.

The lower stratification (0.8°C) was obtained for the experiment 35-24-F when η was equal to 0.89 and t_p was equal to 10.9°C.

Table 3. Experimental performance parameters.

			Displacement		Radiant panels					
Test	η	t_{op} [°C]	V_{air} [L/s]	$t_{air,r}$ [°C]	m_w [kg/h]	$t_{w,r^-} t_{w,s}$ [°C]	$t_{w,m}$ [°C]	CL_{CC} [W]	$CL_{CC}{}^1$ [W/m2]	$CL_{CC}{}^2$ [W/m²]
180-24-F	0.20	24.0	181.4	23.9	200	1.4	22.8	324	24	18
160-24-F	0.24	24.0	163.2	23.7	150	2.3	21.8	397	30	22
140-24-F	0.47	24.1	138.2	23.3	283	2.4	18.3	779	58	43
120-24-F	0.57	24.0	117.0	23.1	400	2.0	16.8	937	70	51
95-24-F	0.64	24.0	94.5	23.3	419	2.2	15.4	1069	80	59
75-24-F	0.73	24.0	72.4	23.1	400	2.6	14.1	1206	90	66
35-24-F[2]	0.89	24.0	36.6	23.5	575	2.4	10.9	1605	120	88
130-24-H	0.00	24.1	131.6	27.8	0	2.0	26.2	0	0	0
100-24-H	0.34	24.0	102.3	26.2	283	1.7	24.7	564	42	31
75-24-H	0.57	24.2	74.4	25.1	400	2.0	20.7	937	70	51
140-F-H	0.49	21.2	142.2	22.8	283	2.5	18.3	813	61	45
75-F-H	0.75	21.4	75.3	22.0	400	2.7	14.1	1237	92	68

[1] Panel capacity expressed per unit of panel area   [2] Panel capacity expressed per unit of floor area

(a)   *(b)*
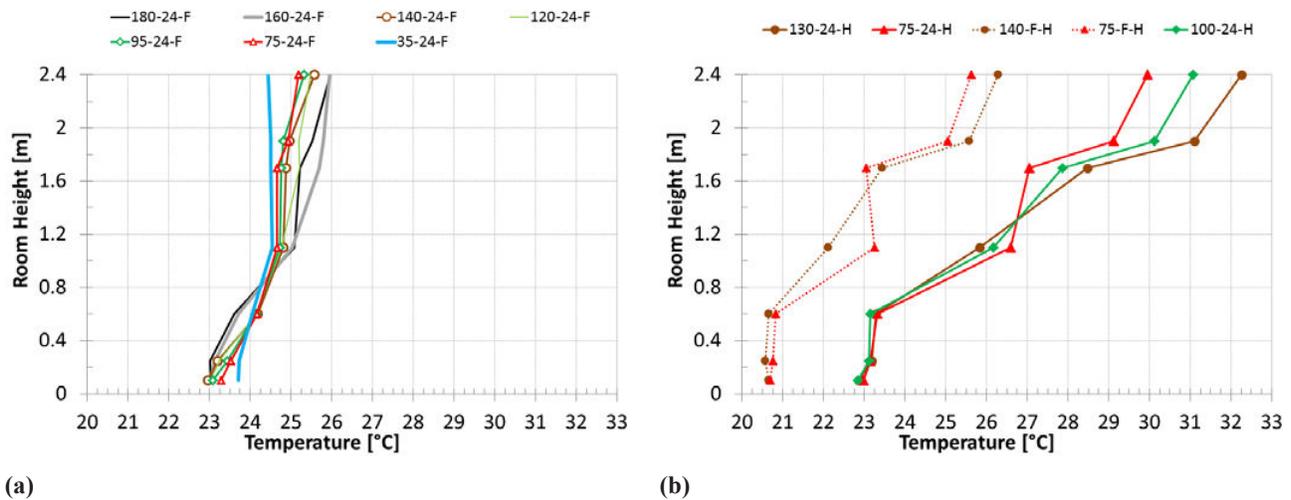
Figure 2. Air temperature profiles for twelve tests described in Table 3: (a) tests with the CPUs located at floor level; and (b) tests with CPUs located at 1.52 m height above floor.

Figure 3 compares the temperature profiles of three tests: 75-24-F, 75-F-H and 75-24-H. In all these tests the heat gains (91.0 W/m²), the airflow rate (~74 L/s), and the air supply temperature have been held constant. From test 75-24-F to 75-F-H the only thing that changed was the location of the CPUs. From under the desk at floor level, the CPUs, representing 51% of total heat gains and 71% of heat gains from the office equipment, were moved above the desk to 1.52 m above the floor. The effect on average temperature in the occupied zone and the amount of stratification is significant. The temperature at ankle level is reduced from 23 to 20.7°C and at 1.1 m from 26.6 to 23.2°C. This air temperature reduction produces a decrease in operative temperature equal to 2.6°C (from 24 to 21.4°C). The temperatures at the ceiling height are quite similar for these two tests. To compare the effect of moving the CPUs from the floor to 1.52 m height on energy use, a third test (75-4-H) was performed at similar thermal comfort conditions to the original floor-level load test (75-24-F). To accomplish this, the supply water temperature to the radiant panels was progressively increased from 12.8°C to 19.7°C. This implied that the average water temperature increased from 14.1 to 20.6°C. We obtained an operative temperature of 24.2°C, almost equal to the case 75-24-F.

We can conclude that increasing the height of the heat sources from the floor to about head-height, for the same thermal comfort conditions, allows a significant increase in radiant panel surface temperature, thereby saving cooling energy. This simple strategy has strong potential for reducing energy consumption in stratified systems (DV and UFAD), as well as implementation of passive or renewable energy sources, such as cooling tower, ground source heat pumps, etc.
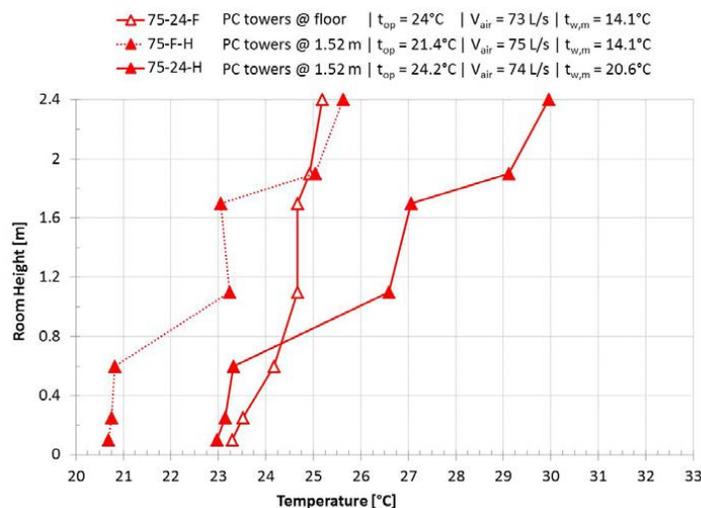

Figure 3. Temperature profiles for 75-24-F, 75-F-H and 75-24-H. Heat gains, airflow rate and supply air temperature were constant. From 75-H-F to 75-F-H only the CPUs location was changed. From 75-F-H to 75-24-H only the water temperature supplied to the radiant panels was increased.

Air change effectiveness

Air change effectiveness tests were performed for 11 of the 12 tests (35-24-F was not performed). Figure 4a presents a representative example of the measured CO_2 concentrations vs. time for test 180-24-F. 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 intake (before injecting the tracer gas) average concentration (continuously measured throughout the test). Figure 4b presents a representative example of the calculated ACE for the three heights vs. time for test 180-24-F. The air change effectiveness values calculated at 0.6, 1.1 and 1.7 m and three key performance parameters of the displacement ventilation and chilled ceiling systems are summarized for all completed tests in Table 4.

The median ACE at 0.6 m is 2.3 (max=3.2 and min = 1), the median ACE at 1.1 m is 1.5 (max=2.1 and min = 1), and the median ACE at 1.7 m is 1.2 (max=1.4 and min = 0.9). All the ACE median values are higher than one (mixing ventilation). Among the parameters reported in Table 4 the height of the heat sources has the strongest effect. When the heat sources are located in the higher part of the room ACE at 0.6 m is consistently higher than 2, ACE at 1.1 m in average equal to 1.6 and almost constantly equal to 1 at standing head height (1.7 m). This means that if we locate the heat sources in the higher part of the room we can create two separate zones, one of clean and fresh air in the lower part of the room (seated occupants) and one with mixed air in the higher part of the room. For the same heat source location the ACEs at 0.6 m and 1.1 m increase with the increase of the air flow rate, the decrease of η and the increase of the panel surface temperature (in these cases equal to the mean water temperature in the panels).

Table 4. Air change effectiveness results.

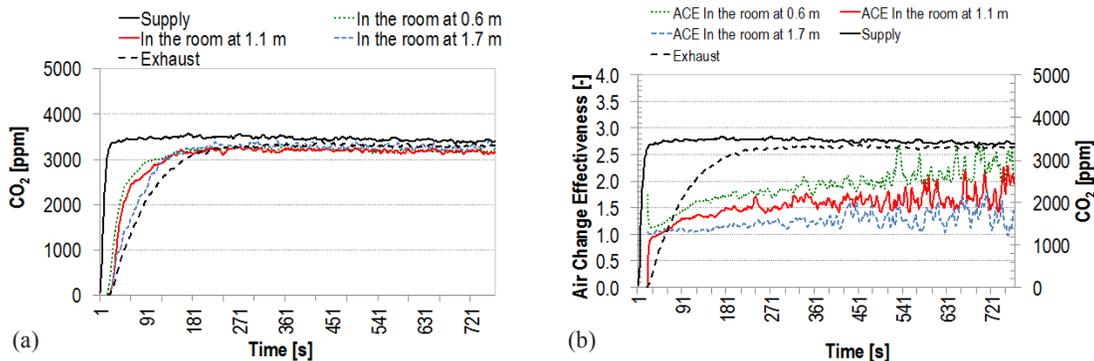| Test	Calculated η [-]	Panel surface temp. [°C]	CC/V_{air} [kW/(m³/s)]	ACE at 0.6 m [-]	ACE at 1.1 m [-]	ACE at 1.7 m [-]
160-24-F	24	21.8	10.2	1.8	1.5	1.4
120-24-F	57	16.8	14.2	1.0	1.1	1.3
75-24-F	73	14.1	22.9	1.2	1.3	1.4
95-24-F	64	15.4	17.5	1.1	1.2	1.3
140-24-F	47	18.3	12.0	1.1	1.2	1.3
180-24-F	20	22.8	9.1	2.3	1.9	1.3
100-24-H	34	24.7	16.2	2.9	1.7	0.9
75-24-H	57	20.7	22.3	3.2	1.5	1.0
130-24-H	0	26.2	12.6	2.7	2.1	0.9
140-F-H	49	18.3	11.7	2.6	1.8	0.9
75-F-H	75	14.1	22.0	2.9	1.0	1.2
Max				3.2	2.1	1.4
Min				1.0	1.0	0.9
Average				2.1	1.5	1.2
Median				2.3	1.5	1.2

Figure 4. (a) CO_2 concentrations for the step-up method at the supply, exhaust and at 0.6, 1.1 and 1.7 m for test 180-24-F; (b) Air change effectiveness calculated at 0.6, 1.1, and 1.7 m and CO_2 concentrations for the supply and exhaust for test 180-24-F.

DISCUSSION

The data analyzed in this paper have been obtained from the same climatic chamber previously described by Schiavon et al. (2012). It is therefore possible to compare and merge the two datasets. In this section the terms "mean water temperature" and "radiant surface temperature", t_p, are synonymous because in these tests the two values were almost the same. This would not be correct for TABS systems. We want to develop a model that could work for radiant panels and TABS, therefore our reference is the surface temperature of the radiant element.

Tan et al. [14] and Ghaddar el al. [16] stated that the ratio between the total cooling load, CC, and the displacement air flow rate, V_{air}, is relevant for prediction of the stratification in a room with DV and CC. In this paper we named this ratio CC/V_{air}. Previously, we demonstrated [8] that the ratio of the cooling load removed by chilled ceiling over the total cooling load, η, cannot be a unique parameter to predict the stratification, because cases with equal η may have different profiles when the active ceiling area is different. Moreover we found that the radiant surface temperature and CC/V_{air} are better predictors of the stratification than η. By looking at the new data we found that η is strongly correlated to t_p (Spearman's rank correlation coefficient, $r = -0.83$) and to CC/V_{air} ($r = 0.88$). This means that we can use these parameters instead of η. We prefer to use t_p and CC/V_{air} because they are the physical parameters that affect the fluid dynamics in the space. We also found a strong correlation between t_p and CC/V_{air} ($r = -0.71$); this could imply that only one of the two parameters is needed as the independent variable in a predictive model.[8]

Figure 5 presents air temperature differences between head and ankle of a seated (1.1 - 0.1 m) occupant as function of the mean surface radiant panel temperature for the data previously published and the tests reported in this paper. Figure 6 shows the same temperature differences as a function of the ratio between the total cooling load and the displacement airflow rate. Figure 5 and Figure 6 show that the previously published data and the data obtained when the CPUs were located under the desk have a similar behavior. It is possible to merge the two dataset and develop a more robust regression model.

Four variables (t_p, η, CC/V_{air} and a dummy variable that identifies if the CPUs are located in the lower or higher part of the room) were used to develop a predictive model. A multivariable regression linear model was developed. Regression models were selected based on R-squared adjusted values and authors' judgment of the maximum number of useful explanatory variables. R-squared, the coefficient of determination of the regression line, is defined as the proportion of the total sample variability explained by the regression model. Adding irrelevant predictor variables to the regression equation often increases R-squared; to compensate for this, R-squared adjusted can be used. R-squared adjusted is the value of R-squared adjusted down for a higher number of variables in the model. The statistical analysis was performed with R version 2.15.1. All the data points have been used except the ones with pure DV ($\eta = 0$). The best regression model, in SI and IP units, is reported below.

$$s = 0.127t_p - 0.528 + k_1 \qquad \text{(SI)} \quad (3)$$

$$s = 0.127t_p - 4.568 + k_2 \qquad \text{(I-P)} \quad (4)$$

Where s is the temperature difference between 1.1 and 0.1 m [43 and 4 in.]) (°C [°F]), t_p is the mean radiant panel surface temperature (°C [°F]), k_1=0.808 and k_2=1.4544 if at least 50% of the heat gains are located at 1.5 m (5 feet) or higher. The model is valid within the experimental conditions tested: 10.9°C (51.7°F)< t_p <24.9°C (76.4°F).

The ANOVA analysis of the regression model indicated that the model is significant ($p<0.001$) and the Adjusted R-squared is equal to 0.64. Visual evaluation of the plot of residuals indicated that the hypotheses of the linear regression model were met, and thus, the model is valid. The model reported in equation 3 and 4 does not include CC/V_{air} because this parameter was strong. Thanks to the data reported in this paper the applicability of the model has been expanded from 16.5°C < t_p <24.9°C to 10.9°C < t_p <24.9°C.

From equations 3 and 4 it can be deduced that the stratification decreases when the surface temperature of the panel also decreases (larger percentage of cooling load removed by chilled ceiling). For the same cooling load, ventilation and thermal comfort conditions, it is possible to increase stratification by increasing the active radiant surface area because this would allow a higher surface temperature to be used. In design, this could be accomplished by employing a larger area (TABS) radiant slab with a DV system, instead of a typically smaller-area radiant panel design. Stratification increases by 0.13°C for every degree increment of the radiant surface temperature. Moving at least 50% of the heat gains from the floor level to 1.5 m (5 feet) or higher produces an increment of the stratification of 0.8°C (1.44°F).

The developed regression model with the 95% confidence interval is shown in Figure 8 for the case with k_1=0 (heat sources located at the floor level). The clear influence of the location of the CPUs on stratification is shown in Figure 7, the difference between the two locations of the CPUs is included in the regression model with the constant k. When the heat sources are located in the higher part of the room the stratification increase significantly.
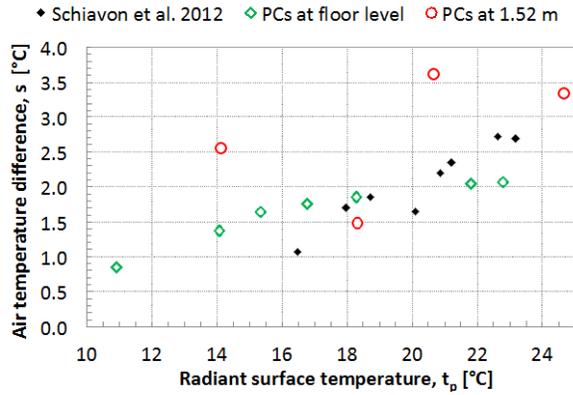
Figure 5. Air temperature difference between head and ankle for seated occupant (1.1 - 0.1 m) as function of the radiant panel average surface temperature for previously published data (Schiavon, 2012) and the tests with the CPUs at the floor level and at 1.52 m.
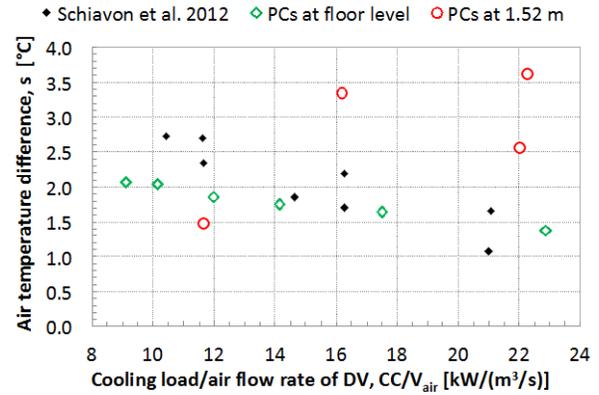

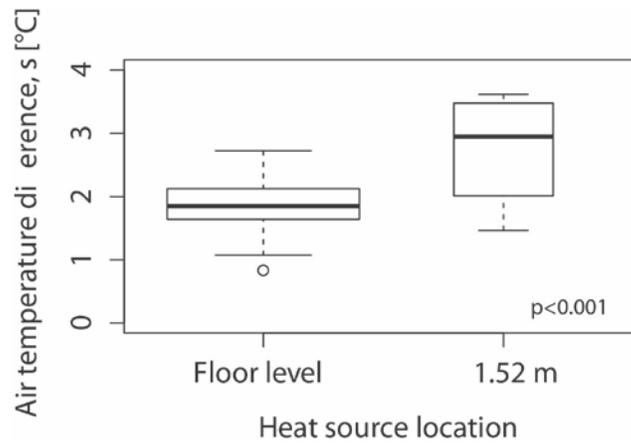Figure 6. Air temperature difference between head and ankle for seated occupant (1.1 - 0.1 m) as function of ratio between the total cooling load and the displacement air flow rate for previously published data (Schiavon, 2012) and the tests with the CPUs at the floor level and at 1.52 m.

Figure 7. Boxplot of the air temperature difference for the CPUs located at the floor level and at 1.52 m (high).
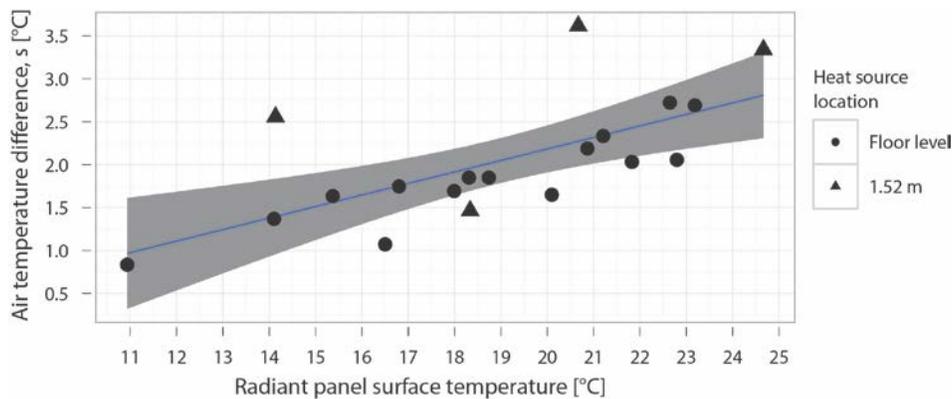

Figure 8. Regression model of radiant panel surface temperature versus air temperature difference (Equation 3) with 95% confidence intervals.

In a similar way to the previous model we developed a regression equation to predict the non-dimensional temperature measured at the floor level, $\phi_{0.1}$, expressed in equation 5. In this case all data points obtained with the CPUs located at 1.52 m have been removed from the database because the $\phi_{0.1}$ was almost constant and equal to 0.35. In addition, the datapoint obtained from test 35-24-F was removed because it was a leverage point for the regression.

$$\phi_{0.1} \quad \phi_{0.1} \quad \frac{t_{air\,0.1} - t_{air\,s}}{t_{air\,r} - t_{air\,s}}$$

$$\phi_{0.1} \quad \phi_{0.1} \quad \frac{CC}{V_{air}}$$

$$\phi_{0.1} = \frac{t_{air,0.1} - t_{air,s}}{t_{air,r} - t_{air,s}} \tag{5}$$

$$\phi_{0.1} = 0.0137 \frac{CC}{V_{air}} + 0.4748 \tag{6}$$

All the variables used in the equations 5 and 6 are described in the nomenclature. . The model is valid within the experimental conditions tested: 9.1 kW/(m³/sec)<CC/V_{air} <22.9 kW/(m³/sec).

The ANOVA analysis of the regression model indicated that the model is significant (p<0.001) and the Adjusted R-squared is equal to 0.73. Visual evaluation of the plot of residuals indicated that the hypotheses of the linear regression model were met, and thus, the model is valid.

Air change effectiveness

When ACE > 1 the designer, according to ASHRAE 62.1-2010 [27], has the opportunity to reduce the outdoor airflow rate or increase the indoor air quality with the same outdoor airflow. Table 4 shows that all the ACE median values are greater than one. This implies that displacement ventilation with a chilled ceiling is able to provide a better indoor air quality than mixing ventilation system even for extremely high cooling load (91 W/m²). In this research ACE was measured at a location far from thermal plumes in order to have a fair representation of undisturbed contaminant concentration. For a seated 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). A moving occupant would most likely be exposed to the air at 1.1m.

We found that ACE0.6 is strongly correlated with $\phi_{0.1}$ (r = -0.74), stratification, s, (r = 0.75) and radiant panel surface temperature, t_p, (r = 0.43). This means that if the stratification or panel surface temperature increase, or $\phi_{0.1}$ decreases, then ACE0.6 increases. This is an expected result, the higher the stratification the better the air quality, but this is the first time, to our knowledge, that these results have been obtained for high cooling load.

Figure 9 shows the boxplot of the ACE0.6 for the two CPU locations (floor level and at 1.52 m). Moving the CPUs from the floor to the higher part of the room increased markedly the air change effectiveness. Moving at least 50% of the heat gains from the floor to 1.52 m caused a median increase of the ACE measured at 0.6 m of 1.75 (from 1.15 to 2.90). Therefore, raising the height of the heat sources not only increases stratification, but also improves indoor air quality (p<0.001). The spread (or inter quintile range) in Figure 9 for the tests with the heat sources located in the higher part of the room is very small. This implies that it is possible to summarize the data with the median and affirm that when the CPUs are located in the higher part of the room ACE0.6 is equal to 2.9. Figure 10 shows the regression models of radiant panel surface temperature versus air change effectiveness measured at 0.6 m with 95% confidence intervals for the cases when the heat sources are located at the floor level and at 1.52 m (high). The regression equation to predict ACE0.6 as a function of t_p is expressed in equation 7. In this case only six values from the dataset have been used because either we did not have the ACE values or they were obtained for the CPUs located above the desks.

$$t_p -$$
$$\text{ACE0.6} = 0.13 t_p - 0.9 \tag{7}$$

The model is valid within the experimental conditions values used for its development: 14.1°C (57.4°F)< t_p <22.8°C (73°F). The ANOVA analysis of the regression model indicated that the model is significant (p<0.028) and the Adjusted R-squared is equal to 0.67. Visual evaluation of the plot of residuals indicated that the hypotheses of the linear regression model were met, and thus, the model is valid.
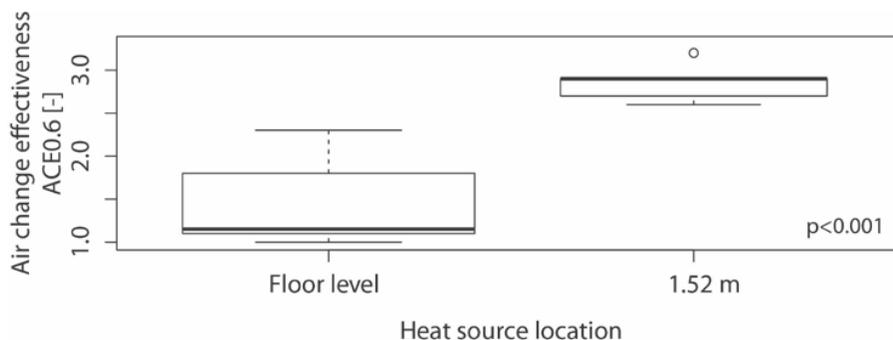

Figure 9. Boxplot of the air change effectiveness measured at 0.6 m (ACE0.6) for the CPUs located at the floor level and at 1.52 m (high)
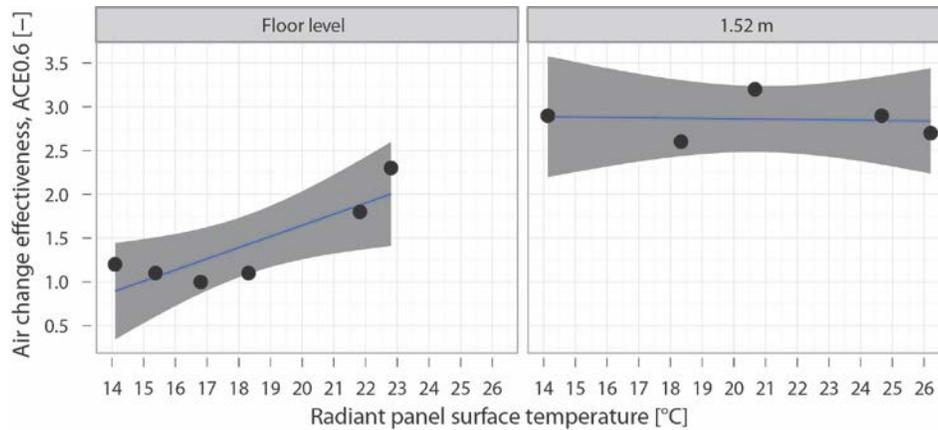
Figure 10. Regression model of radiant panel surface temperature versus air change effectiveness measured at 0.6 m with 95% confidence intervals for the cases when the heat sources are located at the floor level (equation 7) and at 1.52 m (high).

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 73% of the heat load is removed by a chilled radiant ceiling and the radiant panel surface temperature is higher than 14.1°C.

Stratification and air change effectiveness both increase with the increase of the air flow rate, the decrease of η and the increase of the panel surface temperature. We can conclude that the higher the stratification the better the air change effectiveness. An explicit regression model between the two variables is shown in Figure 11.
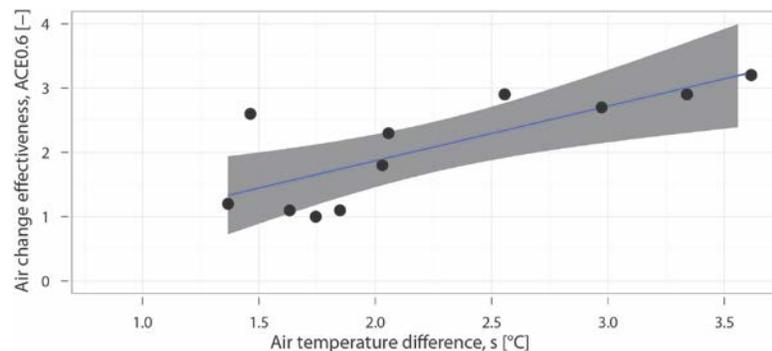

Figure 11. Regression model of stratification (air temperature stratification) versus air change effectiveness measured at 0.6 m with 95% confidence intervals for the data reported in Table 4.

LIMITATIONS

In these experiments we did not directly calculate the uncertainty associated with ACE. Compliance with the standard was considered sufficient. We did not investigate the influence of exterior windows on air distribution. The experiments were performed in a test room representative of an interior zone with (almost) adiabatic walls. Under cooling conditions, it is possible that a rising thermal plume may develop close to warm exterior windows. We do not have evidences of how this may affect the temperature stratification and the pollutant concentration. The proposed models are valid only within the boundary conditions reported in this paper. Caution should be used if applied in perimeter zones. In this study, the influence of variations, supply air temperature, thermal comfort set points, and heat source radiant/convective ratio has not been investigated.

CONCLUSIONS

A laboratory experiment was conducted to investigate room air stratification in a typical office space with a radiant chilled ceiling

(CC) and displacement ventilation (DV). The main conclusions of this study are:

- Displacement ventilation and chilled ceiling are able to maintain thermal stratification and improved ventilation efficiency compared to mixing ventilation system for a wide range of configurations and system design even for extremely high cooling load (91 W/m²).

- Stratification and air change effectiveness both decrease when the surface temperature of the panel also decreases (larger percentage of cooling load removed by chilled ceiling). For every degree decrement of the panel temperature, stratification decreases by 0.13°C and ACE by 0.13. Combining a larger active area (TABS) radiant slab with a DV system (instead of a typically smaller-area radiant panel design) would allow higher radiant surface temperatures to be used, thus increasing stratification and improving ventilation performance.

- Employing a simple strategy of raising the height of the CPUs (representing 51% of total heat gain, or 71% of office equipment heat gains) from the floor level to 1.5 m (5 feet) increased markedly stratification (0.8°C) and the air change effectiveness measured at 0.6 m (1.75). Therefore, moving the heat sources to the higher part of the room reduces energy use and increases indoor air quality. When the CPUs where located in the higher part of the room the median stratification in the occupied zone was 2.95°C and the ACE at 0.6 m was 2.9.

- For the same heat source location the ACEs at 0.6 m and 1.1 m increase with increasing airflow rate, decreasing η, and with increasing panel surface temperature. Similar trends are obtained for stratification in the lower part of the room. The higher the stratification, the better the air change effectiveness.

NOMENCLATURE

ACEX	Air Change Effectiveness measured at X=0.6, 1.1. and 1.7 m.
CC	Chilled ceiling
CL_{CC}	Cooling load removed by the chilled ceiling, W
CL_{DV}	Cooling load removed by the DV system, W
$c_{p,w}$	Specific heat capacity of the water, J/(Kg K)
DV	Displacement ventilation
m_w	Water mass flow rate, kg/h
p	Number of radiant ceiling panels
s	Air temperature stratification between 0.1 and 1.1 m, °C
$t_{air,r}$	Return air temperature from the DV system, °C
$t_{air,s}$	Supply air temperature to the DV system, °C
t_p	Surface temperature of the panel, here supposed equal to $t_{w,m}$, °C
t_{op}	Operative temperature, °C
$t_{w,m}$	Mean water temperature, it is the average of $t_{w,s}$ and $t_{w,r}$, °C
$t_{w,r}$	Water temperature returned from the chilled ceiling, °C
$t_{w,s}$	Water temperature supplied to the chilled ceiling, °C
V_{air}	Air flow rate of the DV system, L/s
η	Ratio of the cooling load removed by chilled ceiling, CL_{CC}, over the total cooling load
$\phi_{0.1}$	Dimensionless air temperature measured at 0.1 m

REFERENCES

[1] Q. Chen, L.R. Glicksman, System Performance Evaluation and Design Guidelines for Displacement Ventilation, ASHRAE, Atlanta, US, 2003.

[2] H. Skistad, E. Mundt, P.V. Nielsen, K. Hagstrom, J. Railo, Displacement ventilation in non-industrial premises, Guidebook n. 1, REHVA, 2002.

[3] ANSI/ASHRAE, ANSI/ASHRAE 55-2010: Thermal environmental conditions for human occupancy, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta (2010).

[4] C. Stetiu, Energy and peak power savings potential of radiant cooling systems in US commercial buildings, Energy Build. 30 (1999) 127-138.

[5] H.E. Feustel, C. Stetiu, Hydronic radiant cooling—preliminary assessment, Energy Build. 22 (1995) 193-205.

[6] Z. Tian, J.A. Love, An integrated study of radiant slab cooling systems through experiment and building simulation, Proceedings of 9th International IBPSA Conference

(2005) 1229-1236.

[7] Z. Tian, J.A. Love, Energy performance optimization of radiant slab cooling using building simulation and field measurements, Energy Build. 41 (2009) 320-330.

[8] S. Schiavon, F. Bauman, B. Tully, J. Rimmer, Room air stratification in combined chilled ceiling and displacement ventilation systems, HVAC&R Research. 18 (2012) 147-159.

[9] F. Causone, F. Baldin, B.W. Olesen, S.P. Corgnati, Floor heating and cooling combined with displacement ventilation: Possibilities and limitations, Energy Build. 42 (2010) 2338-2352.

[10] B. Lehmann, V. Dorer, M. Koschenz, Application range of thermally activated building systems tabs, Energy Build. 39 (2007) 593-598.

[11] F. Alamdari, D. Butler, P. Grigg, M. Shaw, Chilled ceilings and displacement ventilation, Renewable Energy. 15 (1998) 300-305.

[12] S.J. Rees, P. Haves, A nodal model for displacement ventilation and chilled ceiling systems in office spaces, Build. Environ. 36 (2001) 753-762.

[13] A. Novoselac, J. Srebric, A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems, Energy Build. 34 (2002) 497-509.

[14] H. Tan, T. Murata, K. Aoki, T. Kurabuchi, Cooled ceiling/Displacement ventilation hybrid air conditioning system—Design criteria, Proceedings of Roomvent. 1 (1998) 77-84.

[15] M. Behne, Indoor air quality in rooms with cooled ceilings.: Mixing ventilation or rather displacement ventilation?, Energy Build. 30 (1999) 155-166.

[16] N. Ghaddar, K. Ghali, R. Saadeh, A. Keblawi, Design charts for combined chilled ceiling displacement ventilation system, ASHRAE Transactions, SL-08-059 (RP-1438). 143 (2008) 574-587.

[17] M. Ayoub, N. Ghaddar, K. Ghali, Simplified Thermal Model of Spaces Cooled with Combined Positive Displacement Ventilation and Chilled Ceiling System, HVAC&R Research. 12 (2006) 1005-1030.

[18] A. Keblawi, N. Ghaddar, K. Ghali, L. Jensen, Chilled ceiling displacement ventilation design charts correlations to employ in optimized system operation for feasible load ranges, Energy Build. 41 (2009) 1155-1164.

[19] M. Kanaan, N. Ghaddar, K. Ghali, Simplified Model of Contaminant Dispersion in Rooms Conditioned by Chilled-Ceiling Displacement Ventilation System, HVAC&R Research. 16 (2010) 765-783.

[20] W. Chakroun, K. Ghali, N. Ghaddar, Air quality in rooms conditioned by chilled ceiling and mixed displacement ventilation for energy saving, Energy Build. 43 (2011) 2684-2695.

[21] D. Rim, A. Novoselac, Ventilation effectiveness as an indicator of occupant exposure to particles from indoor sources, Build. Environ. 45 (2010) 1214-1224.

[22] ANSI/ASHRAE, ANSI/ASHRAE 129-2002: Measuring Air-Change Effectiveness, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta (2002).

[23] S. Schiavon, F. Bauman, B. Tully, J. Rimmer, Ventilation effectiveness in combined chilled ceiling and displacement ventilation systems, Proceedings of International Conference Indoor Air Quality and Climate (2011).

[24] CEN, EN 14240-2004, Ventilation for Buildings—Chilled Ceilings—Testing and Rating., European Committee for Standardization, Brussels, Belgium (2004).

[25] ISO, ISO 7726, International Standard: Ergonomics of the thermal environment - Instruments for measuring physical quantities, International Organization for Standardization (1998).

[26] ISO, Guide to the expression of uncertainty in measurement, International Organization for Standardization (1993).

[27] ANSI/ASHRAE, ANSI/ASHRAE 62.1-2010: Ventilation for acceptable indoor air quality, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta (2010).