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Room air stratification in combined chilled ceiling and displacement ventilation systems

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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. The purpose of this study was to conduct laboratory experiments for a typical U.S. interior zone office to investigate how room air stratification is affected by the ratio of cooling load removed by a chilled ceiling to the total cooling load, η, for two different chilled ceiling configurations. The experiments were carried out in a climatic chamber equipped with radiant panels installed in the suspended ceiling. In the first test configuration representative of thermally activated slab applications, 12 panels covering 73.5% of the ceiling were used. During the second series of tests, 6 panels covering 36.7% of the ceiling were used, representing a typical installation of metal radiant panels. The cooling load removed by the panels varied between 0 and 73 W/m^2 [0-23.1 Btu/(h ft^2)](based on radiant panel area) or between 0 and 28 W/m^2 [0-8.9 Btu/(h ft^2)] (based on room area). The average mean water temperature of the panels varied over a more moderate range of 20-24°C [60-75.2°F] for the 12-panel tests and over a colder range of 16.5-22.6°C [61.7-72.7°F] for the 6-panel tests. The displacement ventilation airflow rate varied between 1.65 and 4.03 l/(s m^2) [0.32-0.79 cfm/ft^2], and the supply air temperature was kept constant at 18°C [64.4°F]. The results showed that increasing η, the relative amount of the cooling load removed by the chilled ceiling, reduced the total room stratification. However, a comparison between the colder 6-panel tests and the warmer 12-panel tests indicated that average radiant surface temperature (mean chilled water temperature in panels) was a stronger predictor of stratification performance. 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. Despite the impact that the chilled ceiling has on stratification, the results indicate that a minimum head-ankle temperature difference of 1.5°C (2.7°F) in the occupied zone (seated or standing) will be maintained for all radiant ceiling surface temperatures of 18°C (64.4°F) or higher.

KEYWORDS

Displacement ventilation, Chilled ceiling, Air vertical temperature stratification, Radiant panel, thermally activated slab system (TABS).

INTRODUCTION

Displacement ventilation (DV) is a method of room air distribution that could provide improved indoor air quality for contaminants emitted by heat sources (ventilation performance) compared to the dilution ventilation provided by overhead mixing systems. In the classic definition of 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. 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.

The combination of chilled floor and DV may increase the draft risk at ankle level as well as discomfort due to temperature vertical stratification compared to a "DV-only" system. Causone et al. (2010) showed by laboratory experiments 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 (2010). The temperature difference between 1.1 and 0.1 m (43 and 4 in.) varied between 3.2 to 6.6 K (5.8 and 11.9°F) for a heat load varying between 31 and 76 W/m^2 (97.8 and 239.7 Btu/(h ft^2)), DV airflow rate varying between 35 to 80 L/s (74 and 170 cfm) and a supply air temperature varying between 16 and 22°C (60.8 and 71.6°F). They 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.

Mundt (1996) found that a relevant mechanism affecting the temperature stratification level in a displacement ventilation system was of the heat flow from the warm ceiling to the floor through radiation and the consequent convective heat flow from the floor to the adjacent air layer. Causone et al. (2010) affirmed that the large vertical gradients in the case of cooled floor may be caused by the reduction of the convective heat transfer from the floor surface and the air near the floor due to the ability of the cool floor to remove part of the heat coming from the ceiling. Causone et al. (2010) showed that the presence of the radiant cooling floor does not affect the contaminant removal effectiveness (a.k.a. ventilation effectiveness in Europe) of the DV system. During the experiments mentioned above, an average value of contaminant removal effectiveness measured at 1.1 m (43 in.) of 2.20 was found when the airflow rate was 50 L/s (106 cfm), and an average value of 5.70 was found when the airflow rate was 80 L/s (170 cfm). Those values are similar to the one obtained with DV alone.

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 built 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.

Alamadari et al. (1998) concluded, based on computational fluid dynamic (CFD) simulations, that adding CC to a DV system influences the air distribution characteristics of DV. Chilled ceiling panels change the air temperature near the ceiling, which creates downward convection, and hence increases the depth of the upper mixed warm and contaminated region. Furthermore, radiation heat transfer between the chilled panels and walls reduces the room surface temperatures below the room air temperature, causing downward convection near the walls. This may transport pollutants from the upper mixed region down into the supply air and occupied zone (Alamadari et al. 1998). They emphasized that in the CFD simulations, high thermal comfort was achieved.

Rees and Haves (2001) developed a nodal model to represent room heat transfer in DV and CC systems. According to the authors, the model, by separately representing the air movement in the plume and the rest of the room, is able to represent correctly the relationship between the internal load and the air and surface temperatures found in the occupied part of the room. The model is suitable for implementation in an annual energy simulation program (Rees and Haves, 2001) but it cannot be applied as a stand-alone design tool.

Novoselac and Srebic (2002) did an extensive critical literature review of the performance and design of a combined chilled ceiling and displacement ventilation system. They stated that the design of the combined system is more difficult than the design of the CC and DV system working independently and, due to the strong interaction of the two systems, the use of the design guidelines for CC and DV as independent systems is not appropriate for the design of the combined CC/DV 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 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. 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. Tan et al. (1998) suggested that, to maintain a temperature gradient of at least 2°C/m [1.1°F/ft], the DV system should remove a minimum of 33% of the cooling load (i.e., η = 0.67). Behne (1999) 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. (2008) 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. (2006) 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.

Knowing the minimum fraction of the cooling load removed by DV without destroying the stratification is important to maximize energy savings and indoor air quality. The ideal condition is to supply only the minimum outdoor airflow rate needed to maintain indoor air quality and handle latent loads through the DV system while using the cooled ceiling to manage all sensible internal heat loads. In the tests reported in the literature the active chilled ceiling area has been kept constant at a high coverage factor. A large chilled surface area is typical of TABS. Research is needed to investigate what happens if the chilled area is reduced to a smaller percentage, typical of radiant panel installations.

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 (1) the ratio of cooling load removed by CC over the total cooling load and (2) the percentage of active ceiling area affects the room air stratification.

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 [168 in. x 168 in. x 118 in.]) equipped with radiant panels located in a suspended ceiling placed at a height of 2.5 m (98.4 in.) 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 (196 ft^2) and the volume is 54.7 m^3 (1931 ft^3). 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 m^2K/W (20.01 ($ft^2h°F$)/Btu) insulation, a stagnant 0.102 m (4 in.) air gap (0.352 m^2K/W [2.00 ($ft^2h°F$)/Btu]), aluminum extruded walls, and another layer of 0.102 m (4 in.) of polyurethane board (3.522 m^2K/W (20.01 [$ft^2h°F$)/Btu]). By adding up this assembly the overall transmittance is 0.135 W/m^2K (0.0238 Btu/($ft^2h°F$)).

The aluminum radiant panels installed in the suspended ceiling are 1.83 m (72 in) long and 0.61 m (24 in) wide (area equal to 1.11 m^2 [12 ft^2]). Copper pipes are thermally connected to aluminum channels in panels with a spacing of 0.15 m (6 in.). The suspended ceiling is composed of radiant ceiling panels connected in series. Cotton fiber insulation was present over the panels (2.288 m^2K/W [13 ($ft^2h°F$)/Btu]). Two radiant panel configurations were used. In tests 12-1, 12-2 and 12-3 twelve panels were used (13.4 m^2 [144 ft^2] of the ceiling equals 73.5 % of the ceiling area); From test 6-1 to 6-5 six panels were used (6.7 m^2 [72 ft^2] of the ceiling equals 36.7 % of the ceiling area), this second configuration is 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 (24 in.) tall corner mounted displacement diffuser (Figure 3). During the experiments the radius of the diffuser was changed from 0.31 m (12 in.) to 0.46 m (18 in.), different radial dimensions of the diffuser does not affect thermal stratification in the room but they influence the fluid dynamic field in the proximity of the diffuser (adjacent zone). In this study the dimensions and properties of the adjacent zone were not relevant. Heat sources are summarized in Table 2. 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.

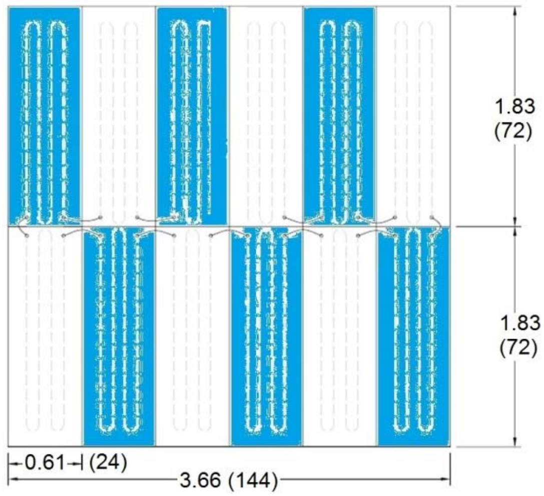


Figure 1. Radiant panels configuration for tests from 6-1 to 6-5. For tests 12-1, 12-2 and 12-3 all the twelve radiant panels were connected. For test 0-1 the panels were disconnected. Dimensions are given in meters with inches in parentheses.

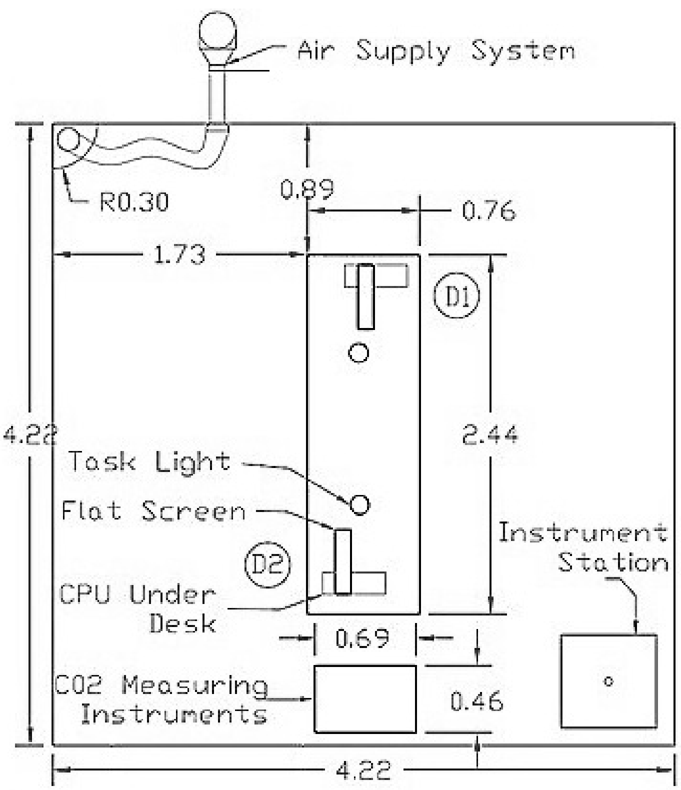


Figure 2. Layout of test chamber. All dimensions are in meters.

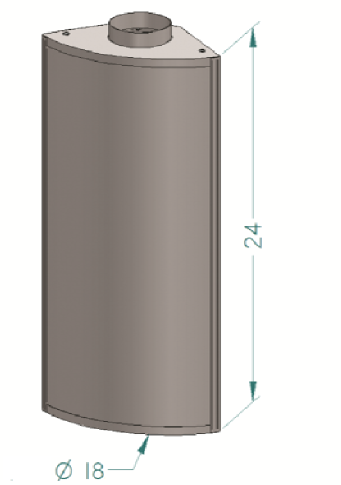


Figure 3. Displacement diffuser used in the room. Dimensions in inches.

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, $t_{w,s}$ ann $t_{w,r}$, 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 suspending 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 [4, 10, 24, 43, 67, 75, 94 in.)] 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 (24 in.) height with a black-globe thermometer. The black-globe thermometer fulfills the requirements of ISO 7726 (1998). The displacement ventilation airflow rate, V_{air}, was measured with a calibrated plate orifice having an accuracy 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 (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 (±0.74°F), the water temperature difference is ±0.125°C (±0.225°F), the cooling load removed by the chilled ceiling is ±25.5 W (±87.1 Btu/h), the electrical total power is ±14.7 W (±50.2 Btu/h), 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).

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:

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

where the $c_{p,w}$ is the specific heat capacity of the 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.

Table 1. Experimental summary.

Test	Airflow rate [L/s] (cfm)	Number of radiant panels, p	Predicted η
Test 0-1	73.5 (156)	0	0
Test 6-1	60.4 (128)	6	25
Test 6-2	54.2 (115)	6	35
Test 6-3	43.1 (91)	6	50
Test 6-4	38.7 (82)	6	65
Test 6-5	30.0 (64)	6	80
Test 12-1	54.2 (115)	12	35

Test 12-2	38.7 (82)	12	65
Test 12-3	30.0 (64)	12	80

The experiments are summarized in Table 1. The heat load in the room was kept constant and equal to 631 W (34.7 W/m^2 [109.5 Btu/(h ft^2)]). The heat loads are described in Table 2. The operative temperature, t_{op}, was kept constant and almost equal to 24°C (75.2°F). 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 (4, 24, and 43 in.). 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 (2010) heights was used. The mean radiant temperature was measured at 0.6 m (24 in.). The DV supply air temperature, $t_{air,s}$, was kept constant and equal to 18°C (64.4°F). In order to keep the operative temperature setpoint equal to 24°C (75.2°F) for all the tests the water mass flow rate and the cold water supply temperature were manually adjusted. 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 2. Heat load summary.

Heat source		Power [W]	Power per floor area [W/m^2] (Btu/(h ft^2))
Workstation 1	Personal computer	67	11.8 (37.2)
	Desk lamp	45	
	Screen	28	
	Thermal manikin	75	
Workstation 2	Personal computer	82	12.7 (40.1)
	Desk lamp	45	
	Screen	29	
	Thermal manikin	75	
Overhead lights		145	8.0 (25.2)
Measuring instruments and data acquisition		40	2.2 (6.9)
Total		**631**	**34.7 (109.5)**

The tests summarized in Table 1 were performed in two different periods. All the tests with 12 panels were performed in November 2009, all the tests with the 6 panels were performed in August 2010. The location and magnitude of internal loads were the same for all tests. The overhead light configuration was changed. In the first series of tests, the overhead lights were in one ballast in the center of the room. In the second series the overhead lights were located in three ballasts aligned parallel to the table at the ceiling level. In the first visit all measuring instruments and data-logger were installed inside the room, in the second visit a data-logger (8 W) was located outside the room. To compensate for the movement of the data-logger, the internal load was increased by 8 W. To verify that these small changes did not affect the temperature stratification, the experiment without the radiant panels (only displacement ventilation) was repeated for both visits. The obtained temperature profiles were very similar. The average of air temperature differences between the two cases calculated at each height was 0.2°C (0.36°F).

RESULTS

The main performance parameters of the displacement ventilation and chilled ceiling systems obtained in the experiments are summarized in Table 3. The experiments are identified based on the calculated η value and the number of radiant panels. The operative temperature was controlled within the range of 23.9-24.2°C (75.0-75.6°F), therefore we may conclude that the comparison was done with almost thermal equal comfort conditions (air velocity and relative humidity were constant as well). The displacement ventilation supply air temperature varied between 17.9 and 18.1°C (64.1 and 64.6°F). The airflow rate varied between

30 to 73.5 L/s (64 to 128 cfm) [1.9 - 4.8 air changes per hour]. The vertical air temperature profiles are shown in Figure 4. $\eta = 0$ means that a full displacement ventilation system was used.

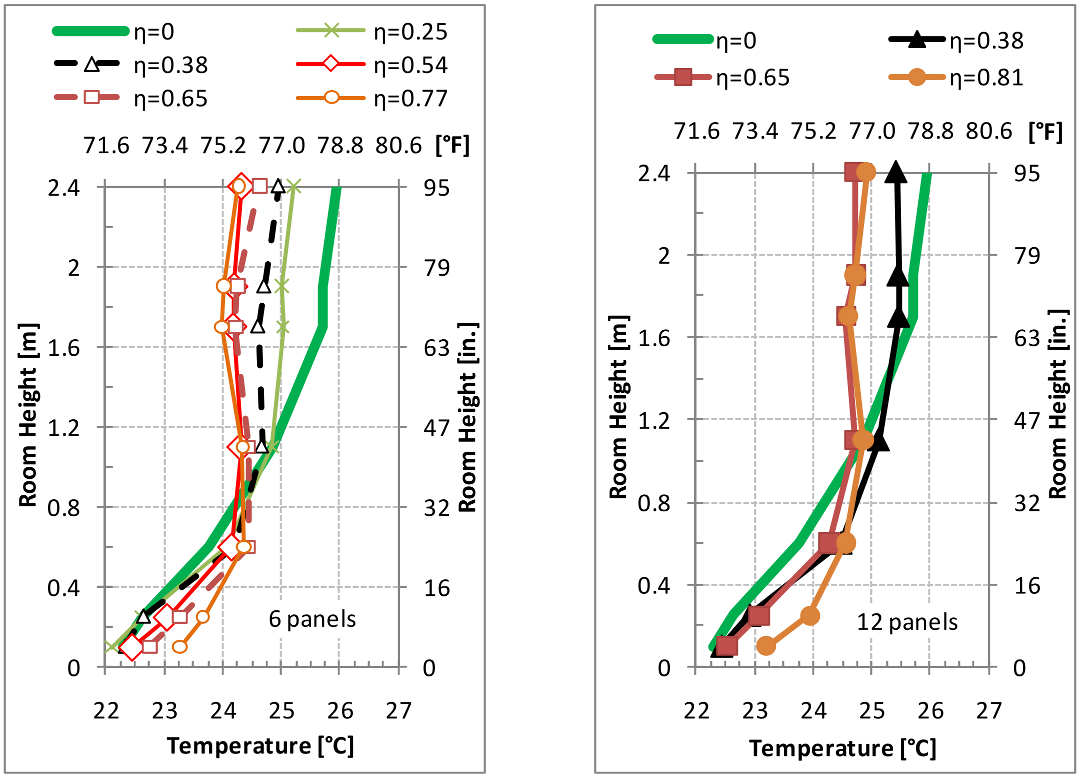
$\eta = 0.25, 0.38, 0.54, 0.65, 0.77$, and 0.81 indicate that the chilled ceiling removed 25, 38, 54, 65, 77 and 81% of the cooling load, respectively. From Figure 4 it can be deduced that the temperature stratification in the occupied zone for a seated person (up to 1.1 m [43 in.] height) is not strongly affected by the change in the cooling load split between displacement ventilation and chilled ceiling for the range of conditions tested. 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 (98.4 in.) from the floor. Figure 4 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 (94.5 in.) by the panels.

Figure 4 shows that most of the temperature stratification is occurring in the occupied zone except for the pure displacement ventilation test (η=0). 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. When η=0, the temperature profile suggests that the stratification height is between 1.1 and 1.7 m (43 and 67 in.). When the chilled ceiling was turned on (η>0), the stratification height appears to be reduced to a height close to 0.6 m (23 in.).

Table 3. Experimental performance parameters.

					Displacement ventilation				Chilled ceiling				
T[a]	η	p	t_{op}		V_{air}		$t_{air,r}$		m_w		$t_{w,m}$		CL_{CC}
-	-	-	°C	°F	L/s	cfm	°C	°F	kg/h	lb/min	°C	°F	W
0-1	0	0	23.9	75.1	73.5	156	25.4	77.8	0	0.0	24.9	76.9	0
6-1	0.25	6	23.9	75.0	60.4	128	24.7	76.5	200	7.3	22.6	72.8	160
6-2	0.38	6	23.9	75.0	54.2	115	24.1	75.4	500	18.4	21.2	70.2	237
6-3	0.54	6	23.9	75.0	43.1	91	23.6	74.6	200	7.3	18.7	65.7	341
6-4	0.65	6	24.1	75.3	38.7	82	23.8	74.8	300	11.0	18.0	64.4	408
6-5	0.77	6	24.1	75.3	30.0	64	23.4	74.2	300	11.0	16.5	61.7	489
12-1	0.38	12	24.2	75.5	54.2	115	25.1	77.2	150	5.5	23.2	73.7	241
12-2	0.65	12	23.9	75.0	38.7	82	24.2	75.5	200	7.3	20.9	69.6	411
12-3	0.81	12	24.2	75.6	30.0	64	24.2	75.6	200	7.3	20.1	68.2	514

[a] Test number



(a) **(b)**

Figure 4. Air temperature profiles for the nine tests described in Table 3. (a) tests with six radiant panels installed in the ceiling (tests from 6-1 to 6-5 and 0-1); and (b) tests with twelve radiant panels installed in the ceiling (test from 12-1 to 12-3 and 0-1).

Figure 5 shows the temperature profiles for four tests (6-2, 12-1, 6-4 and 12-2) where the cooling load split is the same (η=0.38 or η=0.65) but the number of panels is different (p=6 or 12). For η=0.38 reducing the number of panels strongly reduced the temperature stratification, and a similar trend is also observed for the test with η=0.65.

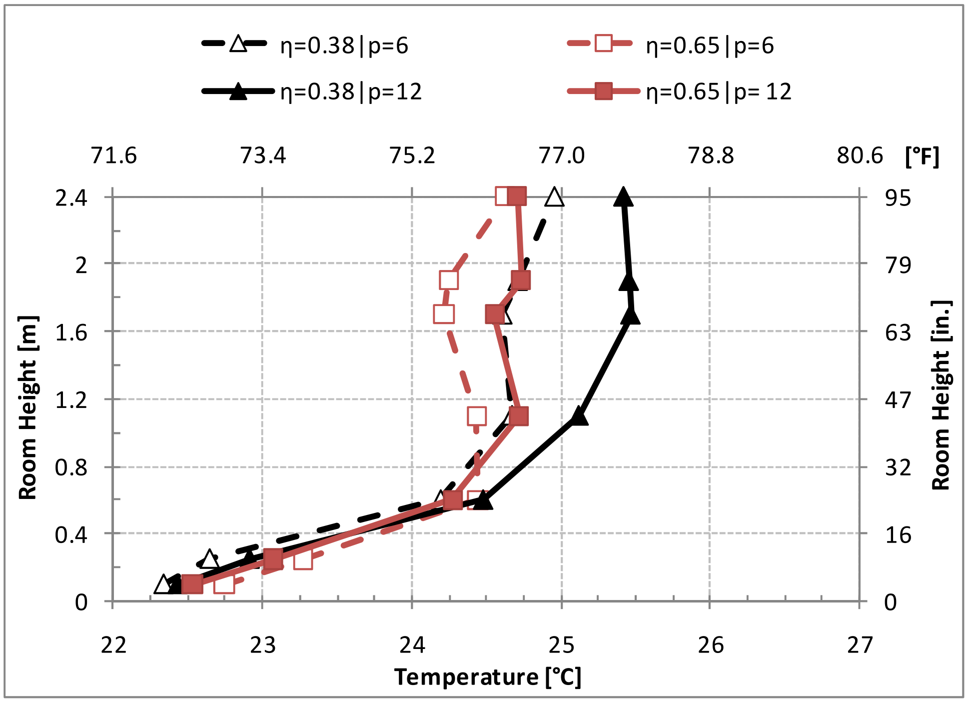


Figure 5. Temperature profiles for test 6-2 (η=0.38 and p=6) , 12-1 (η=0.38 and p=12), 6-4 (η=0.65 and p=6), and 12-2 (η=0.65 and p=12). η is defined in equation 1 and p is the number of panels installed in the suspended ceiling.

DISCUSSION

Figure 5 demonstrates 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 chilled ceiling coverage factor is important too. Figure 6 presents air temperature differences between head and ankle of a seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as a function of the chilled ceiling cooling load ratio. The temperature differences decrease almost linearly with η, this means that increasing the portion of the load managed by the chilled ceiling tends to reduce the temperature stratification. For test 12-1 the vertical temperature difference is equal to the maximum thermal comfort limit (3°C [5.4°F]) specified by ASHRAE Standard 55 (2010). This should not be a problem because research results from an advanced thermal comfort model suggest that greater stratification may be acceptable in the middle of the comfort zone (Zhang et al. 2005). For a seated occupant the temperature gradient varied between 1.1 to 2.7°C [2 to 4.9°F], and for a standing occupant between 0.7 to 3 °C [1.3 to 5.4°F]. These results confirm the statement that temperature stratification is reduced as η increases. When the chilled panels are activated (η>0) the stratification height is reduced and the difference between the temperature stratification of a seated and standing occupant is negligible. In Figure 6, for η=0.38 (tests 6-2 and 12-1) and η=0.65 (tests 6-4 and 12-2), the results indicate that for the same chilled ceiling cooling load ratio , η, it is possible to get different stratification values. This supports the same conclusion shown in Figure 5. The "best-fit" linear regression line for seated occupants in Figure 6 has a coefficient of determination of 0.76, meaning that η is able to explain 76% of the variation seen in the stratification. In the following discussion of Figure 6, Figure 7 and Figure 8, the results from test 0-1 (pure DV) are not included because the focus was on the influence of key parameters on the combined DV/CC system.

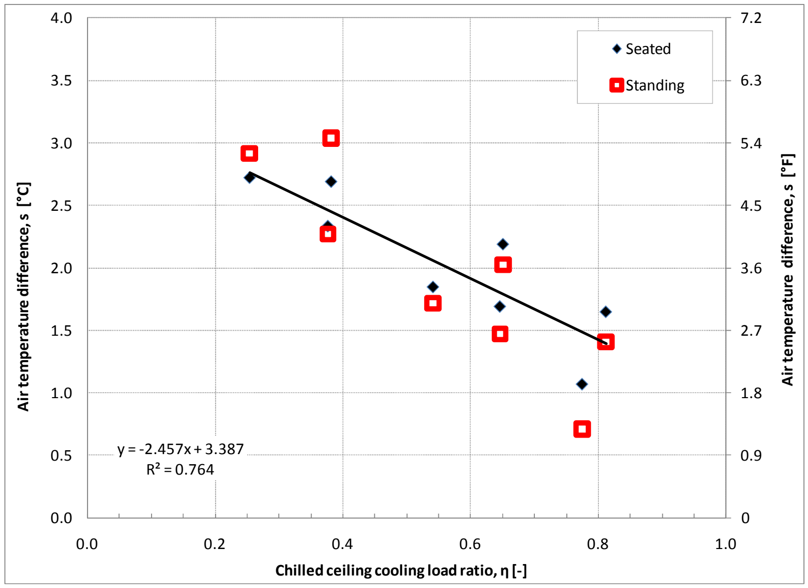


Figure 6. Air temperature difference calculated between head and ankle for seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as function of the chilled ceiling cooling load ratio.

According to Tan et al. (1998) and Ghaddar el al. (2008), 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. Figure 7 shows the air temperature differences between head and ankle of a seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as a function of the ratio between the total cooling load and the displacement air flow rate. The results indicate that stratification is reduced as this ratio is increased (i.e., by increasing the cooling load or by reducing the airflow rate). The linear regression line for seated subjects shown in Figure 7 has a coefficient of determination equal to 0.80.

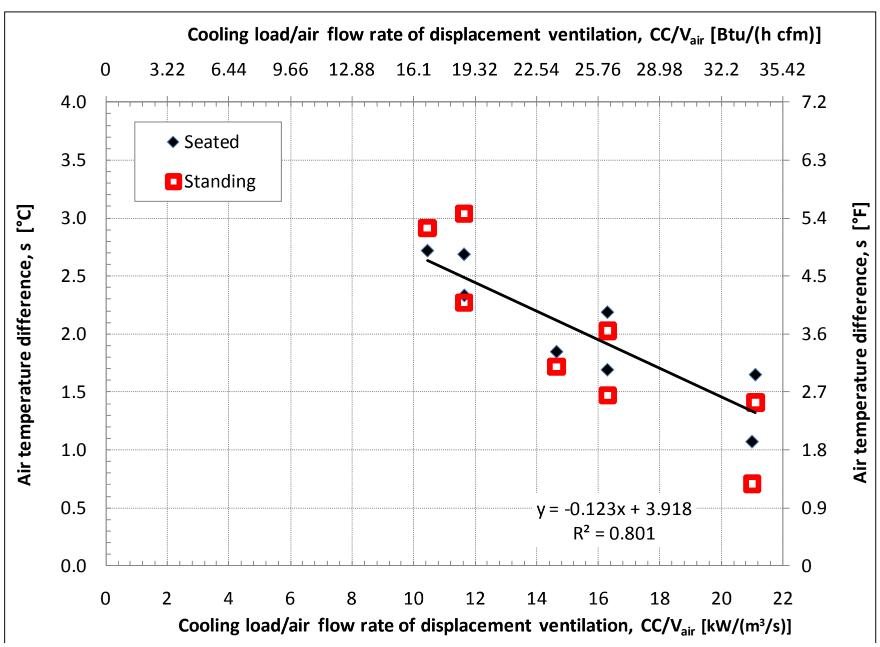


Figure 7. Air temperature difference between head and ankle for seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as function of ratio between the total cooling load and the displacement air flow rate.

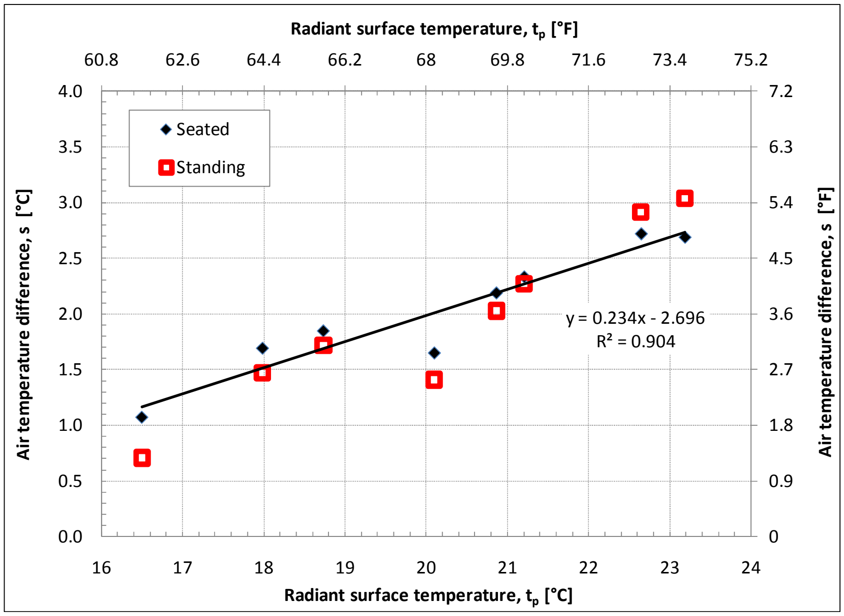


Figure 8. Air temperature difference between head and ankle for seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as function of the radiant panel average surface temperature.

Figure 8 presents air temperature differences between head and ankle of a seated (1.1 - 0.1 m [43-4 in.]) and standing occupant (1.7 - 0.1 m [67-4 in.]) as function of the mean surface radiant panel temperature. In this section the terms "mean water temperature" and "radiant surface temperature" are synonymous because in these tests the two values were almost the same. That will not be true for TABS systems, but here we would like to develop an analysis for the radiant surface temperature. The stratification linearly increases with the increase of the mean water temperature. The “best-fit” linear regression line for seated occupants in Figure 8 has a coefficient of determination of 0.90, meaning that the mean panel water temperature is able to explain 90% of the variation seen in the stratification. In comparison to Figure 6 and Figure 7, this result indicates that the radiant surface temperature is a better predictor of the temperature difference between the head and ankle than the chilled ceiling cooling load ratio or the ratio between the cooling load and the displacement air flow rate.

These three variables (average surface temperature of the panel, chilled ceiling cooling load ratio and ratio between the cooling load and the displacement air flow rate) 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 2.10.1. The model including all three variables had the highest R-squared adjusted but it presented strong multicollinearity problems, therefore a two-variable model was studied. The best regression model, in SI and IP units, is reported below.

$$s = 0.157t_p - 0.059\frac{CC}{V_{air}} - 0.235 \qquad \text{(SI)} \quad (3)$$

$$s = 0.157t_p - 0.066\frac{CC}{V_{air}} - 5.448 \qquad \text{(I-P)} \quad (4)$$

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]), *CC* is the total room design cooling load (kW [Btu/h]), and V_{air}

is the displacement air flow rate (m^3/s [cfm]). The model is valid within the experimental conditions tested: 16.5°C (61.7°F)< t_p <24.9°C (76.9°F); 1.64 L/(sm^2) [0.32 cfm/ft^2]< V_{air} < 3.31 L/(sm^2) [0.65 cfm/ft^2]; and CC < 34.7 W/m^2 (109.5 Btu/(h ft^2)). When s is less than 0 it should be considered as well mixed condition with no stratification.

The ANOVA analysis of the multi-variable regression model indicated that all variables are significant (p<0.002) and the Adjusted R-squared is equal to 0.98. Visual evaluation of the plot of residuals indicated that the hypotheses of the linear regression model were met, and thus, the model is valid.

Tan et al. (1998) proposed a design diagram where the x-axis represents the ratio (P/Q) of the total cooling load (kW) to the displacement ventilation airflow rate (m^3/s). The design diagram was obtained from a set of laboratory experiments where the supply air temperature of the DV system was kept constant at 22°C (71.6°F) and the surface of the radiant panels varied between 16°C and 24°C (60.8-75.2°F); the radiant panels covered 70% of the ceiling. The diagram, reproduced in Figure 9, shows the relationship between the vertical temperature gradient, the ratio P/Q and η. In the figure, R is what we call η. Tan et al. (1998) proposed a maximum ratio of P/Q equal to 18 kW/(m^3/s) [28.98 Btu/(h cfm)] because higher values would probably cause mixing and lower air quality in the occupied zone. Seven of the nine points obtained in our tests are plotted for comparison in Figure 9.

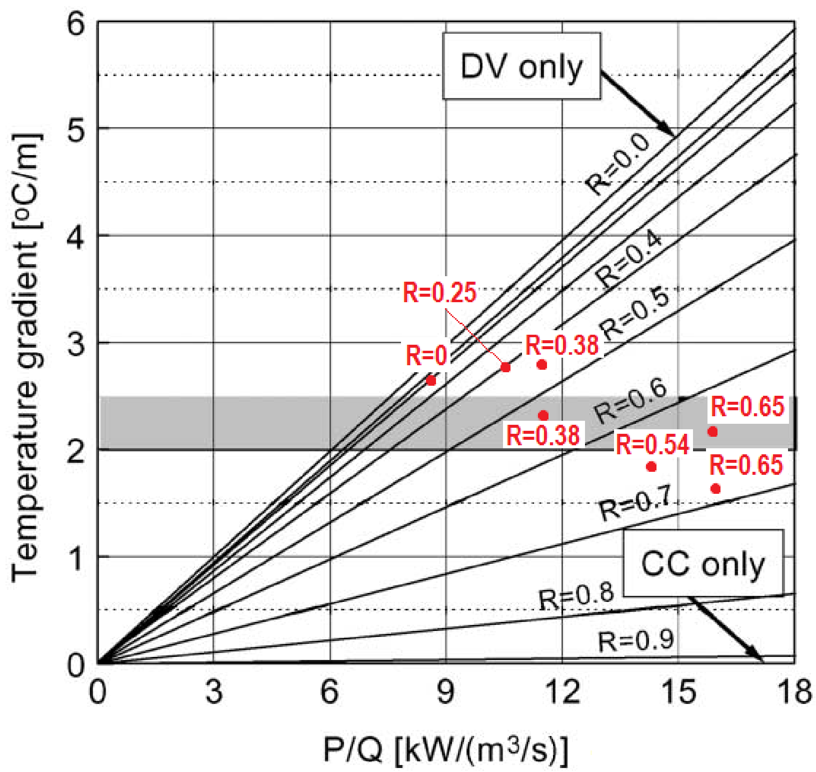


Figure 9. Design diagram developed by Tan et al. (1998). In this graph R is equal to *η* and P/Q is CC/V_{air}.

The temperature gradient has been calculated for the occupied zone of a seated person. The results showed that in our experiments the stratification is generally lower (roughly 1°C [1.8°F]) than the one predicted by the method. Figure 9 shows that for the same η (referred to as R in the figure) the obtained stratification is different, and for the same η the stratification is lower when the number of panels is equal to six. This method tends to overestimate the stratification, therefore it is possible to destroy the stratification without being able to predict it. In Figure 9, only seven points are shown because for η=0.81, the P/Q ratio was 21, thus outside of the range for the figure; for that point the temperature gradient was 1.6°C (2.9°F) for test 12-3 and 1.1°C (2°F) for test 6-5.

The results presented in this paper have shown that the chilled ceiling cooling load ratio is not a sufficient predictor of the stratification (see Figure 6). Therefore, it is not sufficient to report the stratification for a given η. For example, for η=0.77 the stratification was 1.1°C (2°F) and for η=0.81 the stratification was 1.6°C (2.9°F). The findings from this study demonstrate that a better predictor of stratification performance of a CC/DV system will be based on the average radiant surface temperature and the ratio of the total room cooling load to the DV airflow rate..

In the experiments with 12 panels 73.5% of the ceiling area was covered with radiant panels. The cooling load removed by the panels varied between 18 and 38 W/m^2 (based on radiant panel area) [56.8-119.9 Btu/(h ft^2)] or between 13 and 28 W/m^2 (based on room area) [41.0-88.3 Btu/(h ft^2)]. The average mean surface temperature of the panels varied between 20.1°C and 23.2°C (68.2-73.7°F). These values are representative of thermally activated slab applications. In the experiments with 6 panels 36.7% of the ceiling area was covered with radiant panels. The cooling load removed by the panels varied between 24 and 73 W/m^2 (based on radiant panel area) [75.7-230.3 Btu/(h ft^2)] or between 9 and 27 W/m^2 (based on room area) [28.4-85.2 Btu/(h ft^2)]. The average mean surface temperature of the panels varied between 16.5°C and 22.6°C (61.7-72.8°F). These values are typical of metal radiant panel applications.

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:

- Average radiant ceiling surface temperature is a better predictor of the temperature difference between the head (1.1 m [43 in.]) and ankle (0.1 m [4 in.]) 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.
- For the range of test conditions covered, 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.
- Despite the impact that the chilled ceiling has on stratification, the results indicate that a minimum head-ankle temperature difference of 1.5°C (2.7°F) in the occupied zone (seated or standing) will be maintained for all radiant ceiling surface temperatures of 18°C (64.4°F) or higher.
- A model to predict the temperature difference between the head and ankle as a function of the average radiant surface temperature and the ratio between the cooling load and the displacement air flow rate has been developed for the displacement ventilation and chilled ceiling combined system.

NOMENCLATURE

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_c	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

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