IN-SITU PERFORMANCE OF CHILLED CEILINGS IN STRATIFIED ENVIRONMENTS

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ABSTRACT

In this study, the thermal comfort, air change effectiveness, and stratification profiles are characterized for an in-situ test. A private office was built in order to analyze the effects of the load split between the chilled ceiling and displacement ventilation systems on the stratification and thermal comfort. Tests included varying the ceiling temperature, room load, and load placement. Results from six load configurations are compared to the literature and offer field observations of the trends shown in lab studies. The study indicates that the use of Chilled Ceilings with stratified air supply systems can lead to comfortable room conditions with good air quality as described in the literature.

KEYWORDS

Thermal comfort, displacement ventilation, underfloor air distribution (UFAD), temperature stratification

INTRODUCTION

Chilled ceilings and displacement ventilation systems are independently well developed and understood building cooling technologies. Gaining momentum is their application as a hybrid solution in high performance buildings. A proposed development in North America sought to use the advantages of these technologies in cellular office configurations with higher than typical loads and with custom made displacement air outlets under the desks. A typical cellular room was constructed to contain systems, desk layout, and office equipment representative of those which are proposed in order to vet the analytical models used in the design, test operational hypotheses, as well as to expose the owner to the general comfort of the proposed system configuration. It was also developed to investigate and optimize the indoor environment, including thermal comfort and air quality, as well as the larger HVAC system design, which was conducive to an analytical approach. The tests undertaken were generally intended to demonstrate the system's performance when environmental conditions are constant and a steady state of operation has been achieved. The thermal conditions of the office were measured by the operative temperature, which is the customary reference temperature for comfort. This was targeted to be 23°C with gradual swings between 21°C and 25.5°C as the system responds and loads change.

The proposed design responded to variations in internal load, varying the supply air temperature by maintaining a stable internal comfort temperature. Two load cases were tested. These were a medium load office (seen as the base case for the majority of enclosed spaces), and a heavy load case (seen as towards the upper load level that might be required in such offices).

Particular interest was in understanding the following configurations:

- Cooling radiant panels operating with floor level air supply compared to a floor level air supply only. For the floor level air supply, airflow volume was increased to give broadly similar comfort conditions as seen with the radiant panels in operation.
- Varying the supply air temperature to assess in detail the envelope of potential operating conditions. Air was supplied at both 25.0°C (the design case) and also at 18°C for each load case. This demonstrated the potential to operate the building at either lower internal temperatures or to extract more load.

The challenge with realizing hybrid systems is reassuring clients that they are responsive and appropriate for managing the high internal loads which are perceived and real with proposed buildings. As such, the main purpose of this investigation was to prove the design under typical and extreme scenarios to see how far the system could be pushed while providing a comfortable environment.

Tan et al. [1] 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, with 0 representing a pure displacement ventilation (DV) and 1, a pure chilled ceiling system. Schiavon et al. [2,3] experimentally investigated the influence of percentage of ceiling active area and of the split of cooling load between displacement and chilled ceiling on stratification. They found that the average radiant ceiling surface temperature is a better predictor of the room temperature stratification, measured as the temperature difference between the head (1.1 m) and ankle (0.1 m) of a seated occupant.

METHOD

Analytical Model

The load split, η, can be adjusted by selecting the appropriate supply temperatures and flow rates of air to the room and of water to the chilled ceiling. In order to select the desired operating point, an analytical model was developed, characterizing the steady state performance of the chilled ceiling and accounting for the cooling to be provided by the air system. A nomogram was developed, allowing the rapid selection of supply conditions for the achievement of desired room operating conditions.

The analytical model is based on steady state heat balances at two points – the ceiling and the room. The ceiling energy balance equates the radiative and convective heat transfer from the room with the conductive heat transfer through the slab to the pipes:

Radiant heat transfer is calculated using the Stefan-Boltzmann law for a grey body, and assuming a soffit surface emissivity of 0.9 for concrete and the mean radiant temperature of the room equal to the air temperature. The convection heat transfer coefficient is found from the correlation given by Novoselac et al. [4], based on measurements taken in a controlled room with a chilled ceiling:

1) $$q_{conduction} = \frac{4\pi k\left(T_2 - T_1\right)}{\frac{1}{Bi_1} + \ln\left[\frac{d}{\pi r_1 D}\sinh\left(2\pi\left[D + \frac{D}{Bi_2}\right]\right)\right]}$$

2) $$h_c = 2.12\left(T_{air} - T_{extract}\right)^{1/3}$$

3) $$D = \frac{d}{s} \qquad Bi_2 = \frac{h_2 d}{k} \qquad Bi_1 = \frac{h_1 r_1}{k}$$

Steady state conduction from the soffit to the cold water pipes is calculated using the relation given by Equation 1 to account for the dimensions and spacing of the pipes. This was taken from an extensive list of analytical conduction solutions [5]. Since this conduction factor accounts for conduction to both the upper and lower horizontal surfaces, it was halved to reflect that only the lower surface is active.

In order to calculate the Biot numbers shown, the heat transfer coefficients for the inner and outer surfaces, h_1 and h_2, are required. The parameter h_2 is the combined convective and radiative heat transfer coefficient ($h_c + h_r$) at the soffit. T_2 is the operative temperature in the room. The parameter h_1 was found by calculating the Nusselt number for pipe flow with a given mass flow rate and water temperature, T_1, according to a correlation given by Nellis and Klein [6]. This was found not to vary significantly over the range of mass flow rates recommended by manufacturers of chilled water ceiling systems.

The room energy balance equates the sensible heat of the internal loads with the cooling achieved by the ceiling and the remaining cooling required by the air system – parameterized by air change rate and temperature difference between supply and return. The nomogram, which is shown in Figure 1, facilitates the selection of supply conditions to achieve a room operating condition, using geometrical constructs. By placing graphs for achieved and required ceiling cooling side by side with a common y-axis, the user can read off the pipe water temperature and air change rate required given a desired room operative temperature and supply air temperature.

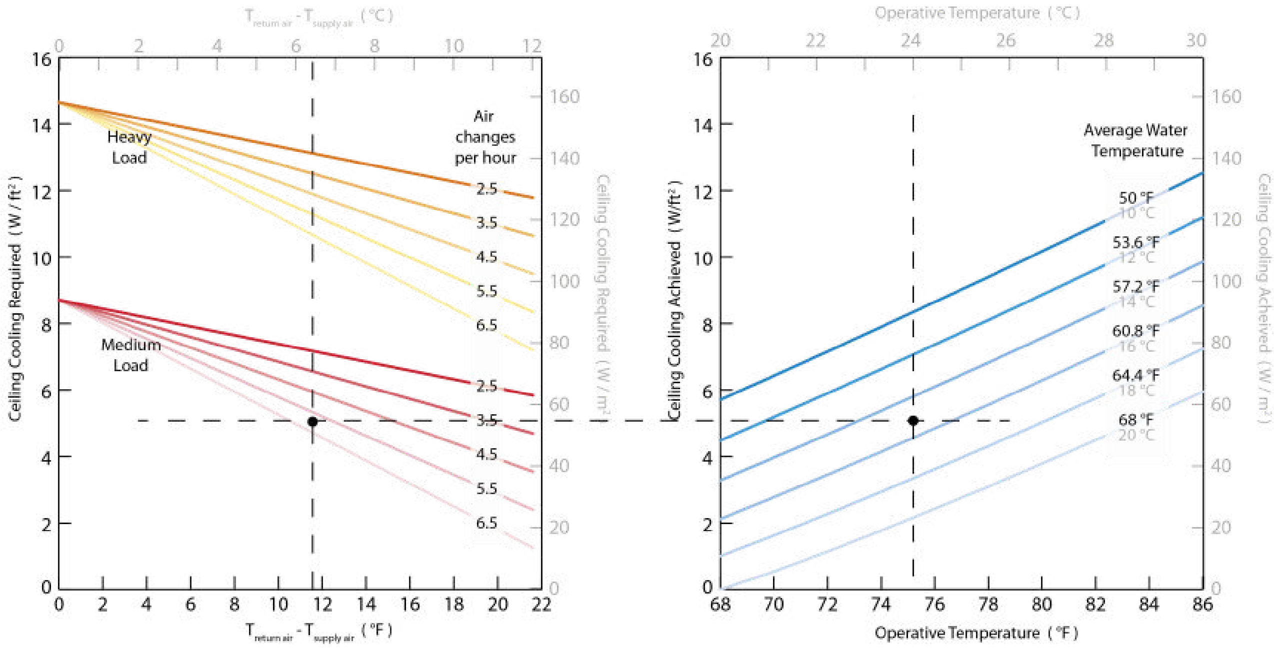

Figure 1 Cooling Nomogram

To use the nomogram, a vertical line should be drawn on the right-hand graph at the desired room operating temperature, and another on the left-hand graph at a specified air temperature difference (return - supply). Finally, any horizontal line between the two verticals shows pairs of air change rate and average pipe water temperature which satisfy the cooling requirement.

A significant proportion of the cooling achieved by a chilled ceiling is radiative. The analytical calculation of this mode of cooling requires that the mean radiant temperature is estimated and held constant. Typically, a complete picture of the radiative environment of a room is difficult to ascertain, so the MRT is held equal to the air temperature. In our analysis, however, numerical simulations were also developed, using a finite element software package [7] designed to solve thermal radiation heat transfer. It was used to calculate all surface temperatures in the room and hence give a more accurate picture of the radiant cooling achieved. This model included geometrically accurate computer equipment with appropriate heat fluxes applied, and seated human manikins controlled by a full physiological model developed by Fiala [8] to produce an appropriate thermal response to their surroundings. Whilst the results are not presented in this paper, the numerical simulations gave a good degree of agreement with the analytical model presented above, for a number of different operating points. In particular, the assumption that the mean radiant temperature was equal to the air temperature was shown to be acceptable.

Laboratory Testing Procedure

The laboratory experiments were carried out in a simulated two-occupant office space (3.65 m x 3.05 m x 3.03 m) equipped with radiant panels located in a suspended ceiling placed at ceiling level, located within a conditioned laboratory. The office was constructed of plywood on three walls, the fourth being a glass wall with a sliding door. The floor was a lightweight concrete tile system above a void to accommodate an underfloor air distribution system. The office was set up to accommodate two to four occupants and their equipment.

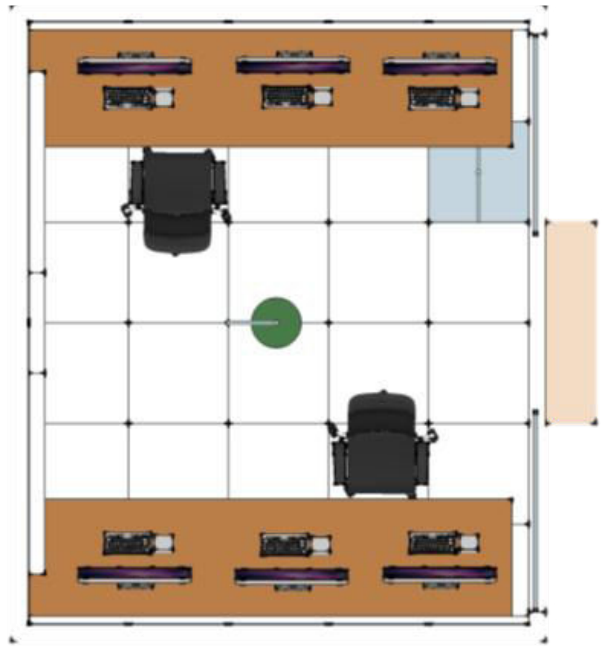

Figure 2 Configuration of the testing chamber

The ceiling system was constructed of ten aluminum radiant panels installed in the suspended ceiling, covering the entire floor area, except a 25mm gap along the back wall which served as the air extract. The aluminum panels were fitted with copper pipes embedded in a heat transfer paste and mechanically fixed to aluminum plates to convey heat efficiently from the pipes to the panel surface. There were two parallel circuits used, split along the room centerline with a set of five panels on either side connected in series. When the radiant panels were active, the average surface temperature was set to be 18°C, as measured at the midpoint of the left loop which was indicative of the overall ceiling temperature as verified through the commissioning process. To achieve this temperature, each loop was supplied with water at a rate of 227kg/h at 17°C entering water temperature. The panel system was designed to replicate a chilled slab having reached a steady state. For practical reasons, simulation of the thermal conductivity or response of the slab was not explored.

Figure 3 shows the locations of the workstations, heat loads, as well as the measurement location for both thermal comfort and gas label concentration. The supply air was delivered through continuous displacement ventilation diffusers located along the bottom of each of the side walls, beneath the desks. The supply air volume varied from test to test depending on the amount of load in the room. The supply air temperature set point was constant at 18 or 20°C. The exhaust air volume was adjusted to match the supply airflow and maintain neutral pressure within the space. Heat sources consisted of tower CPUs, and flat screen monitors.

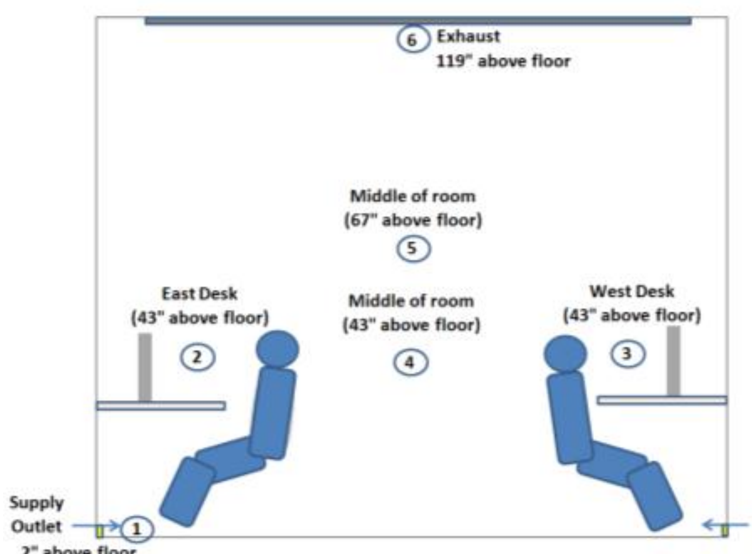

Figure 3 CO2 Room Configuration and Sensor Location

The primary lighting for the space was provided by 3m of linear 150mm wide fluorescent lighting centered on the back wall, 25mm below the ceiling. Secondary lighting was provided along a light ledge on both sides of the room with strings of LED rope lighting. The light ledge was at a height of 1.5m above the under floor and extended the 3m depth of the room. The total lighting load provided to the space was 120 Watts (410 Btu/H). The heat loads are summarized in Table 1.

Table 1. Baseline heat load summary

Heat Source	Number	q_{total} [W]	Power per floor area [W/m^2]
CPUs	4	400	35.9
Screens and lamps	4	280	25.1
People	2	200	17.9
Instrumentation	1	50	4.5
Overhead Lighting	2	120	10.8
	Total	**1050**	**94**

Occupants were simulated with heated manikins seated in office chairs next to each desk. Each occupant simulator was set to provide 100 Watts of load to the office space. When fully installed, the test chamber represented a two-person office with multiple computers at each workstation located on desks mounted 0.76m above the floor. Two test scenarios were run with a higher internal thermal gain, with an additional 700W of equipment load in the room.

The tests were conducted to evaluate the thermal comfort and air change effectiveness. Each test recorded the temperature and velocity sensor readings every 30 seconds with a Keithly 2600 data logger. The temperature and velocity readings were taken with RTD type temperature probes (± 0.4°C) and TSI model 8475 sensors (±3% of reading + ±1% of full scale), respectively.

The load split, η, was calculated from:

$$\eta = \frac{q_{CC}}{q_{total}} = \frac{q_{CC}}{q_{DV} + q_{CC}} . \quad (4)$$

with q being the cooling load. The amount of cooling provided by the chilled ceiling is determined from the heat equation, using the difference of the entry and exit water temperature from the ceiling array:

$$q_{CC} = \dot{m} c_p \left(t_{w,out} - t_{w,in} \right). \quad (5)$$

The primary metrics for the evaluation of thermal comfort are the operative temperature, t_{op}, and the draft rate, DR, as defined by ASHRAE [9]:

$$t_{op} = \frac{t_{air} + \bar{t}_r}{2} . \quad (6)$$

$$DR = \left[(34 - t_{air})(v - 0.05)^{0.62} \right] (0.37 v Tu + 3.14) . \quad (7)$$

The tests conducted to evaluate the ventilation effectiveness saw CO_2 sensors added to the room in the locations indicated in Figure 3. The air change effectiveness testing was conducted in accordance with ASHRAE Standard 129 [10]. Once the room had achieved a thermal equilibrium for a minimum of 30 minutes, the gas label, CO_2, was introduced to the air supply stream at a concentration of 4000ppm. The concentration data collected during the step-up and decay procedure was analyzed to evaluate the age of air at all measurement locations, given by equations 8 and 9, respectively.

8) $$A_i = \left(t_{stop} - t_{start}\right)\left(1 - \frac{C_{i,avg}}{C_{i,t_{stop}}}\right).$$

9) $$A_i = \left(t_{stop} - t_{start}\right)\left(\frac{C_{i,avg}}{C_{i,t_{start}}}\right).$$

10) $$\tau_n = \frac{Q_{ex} A_{ex}}{Q_{ex}} = A_{ex}.$$

Each of the tests are summarized in Table 2. Note that the loads in tests 1, 4-7 were as decribed in Table 1, though with measurements taken at the start of each test. Some variability is noticed in the amount of power drawn by the computer equipment from test to test, but is limited to 1-2% of the total load.

Table 2. Test Summary

Test	η	q_{total} [W]	Supply air temp. (°C)	Return air temp. (°C)	t_{CC} (°C)	Q (L/s)
1	0	1040	20.1	25.2	24.7	122.7
2	0.56	1040	17.7	24.1	17.3	56.6
3	0.65	1040	20.1	24.8	17.6	56.6
4	0.36	1750	20.1	23.9	18.2	212.4
5	0.26	1750	17.8	23.2	17.3	212.4
6	0	1068	20	25	24.0	113
7	0.56	1068	20	25	19.4	56
8	0.79	1068	20	24.5	18.3	28.3

The conditions for Test 2 are overlaid on the nomogram in Figure 1. The airflow rate of 56.6 l/s is equal to 6 ACHs, and Treturn air - Tsupply air is equal to 6.4°C. With an operative temperature of 24°C, an average water temperature of 15°C is specified by the nomogram. However, in the test described, water was held at an average temperature of 18°C. This disparity may exist because the analytical model described above is for cold water pipes set in concrete, whereas the equipment used for the laboratory tests was an aluminum radiant panel.

In order to better understand the thermal performance of the system under varying load, a case was tested where the room system remained on (though at a reduced airflow rate) while the occupant and equipment load was switched off. This was done in order to determine if there would be unacceptable thermal conditions when occupants were to return from a lunch break or extended meeting.

RESULTS

Table 3 presents the results from the air change effectiveness testing. In general, it is noted that the ACE at 1.1m inversely correlates with the value of η, with the largest ACE observed when the system is in 100% DV mode (η=0), to 0.99 when the percentage of zone cooling accomplished by the chilled ceiling increases.

Table 3. Air change effectiveness results

η	t_{CC} (°C)	Test	Air Change Effectiveness	
			Average 1.1m	Average 1.8m
0	24	6	20.1	25.2
0.26	17.3	5	17.7	24.1
0.56	17.3	2	20.1	24.8
0.56	19.4	7	20.1	23.9
0.79	18.3	8	0.99	0.98

This is not unexpected as systems with an increase in chilled ceiling capacity typically show an increase in stirring in the space due to convection currents developing in the region of the ceiling that would cause more mixing in the zone, and thereby a more even distribution of the gas label concentration. A review of the average ACE values at 1.1 and 1.8m seem to support this by demonstrating a potential lowering of the stratification height, above which we would expect the room to be fully mixed.

The results from the thermal comfort testing are shown in Table 4. From the table, it is noted that all but two tests satisfy the ASHRAE limit of DR=20 and a temperature difference between the head and ankle of Δ_{hf} >3K, indicating that design provides a thermally comfortable environment.

Table 4. Operative temperature, stratification and draft rate results

Test	η	Δt_{hf} (K)	t_{op} 1.1m (°C)	DR 0.1m
1	0	2.7	23.8	7
2	0.56	4.0	24.2	14
3	0.65	2.7	23.1	10
4	0.36	0.9	23.3	23
5	0.26	1.2	20.7	24
6	0	2.7	23.4	11.5
7	0.56	2.0	23.6	10.3
8	0.79	3.7	23.3	10.1

Figure 4 shows the decay in room temperature during the variable loading test.

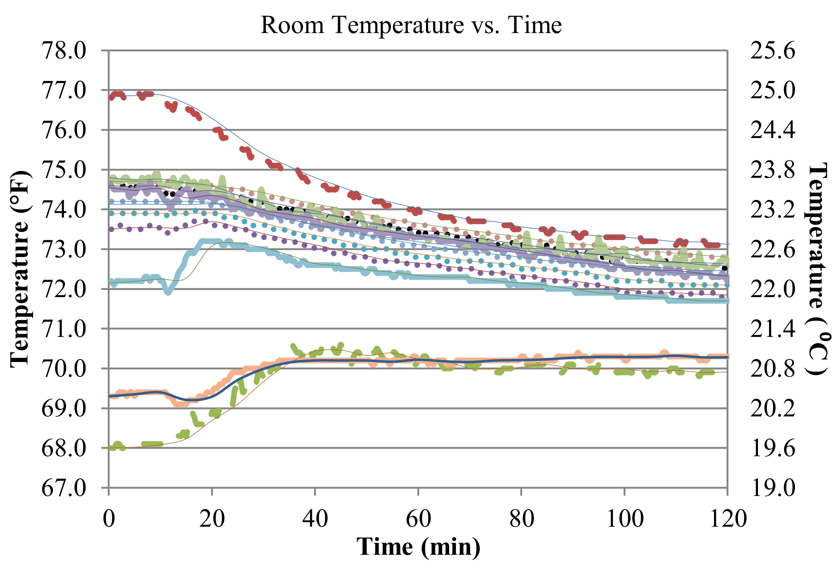

Figure 4 Decay in room temperature after the removal of the occupant and equipment loads

DISCUSSION AND CONCLUSION

For the medium load office with the design supply air temperature of 20°C the measured room operative temperature was 23.1°C. This shows very close alignment between the design target, 23.3°C, the earlier computer modeling undertaken, and the measured test data. At a supply air temperature of 18°C the measured room operative temperature was 22.4°C. Some ankle level coolness was sensed, although this is considered to be within acceptance criteria. To evaluate the case where there is no cooling contribution from the radiant system, the ceiling panels were turned off. In order to maintain the operative temperature of 23.2°C which was realized with the CC/DV configuration, an increase in the supply air volume flow rate of 267% of the design value was observed.

For the high load case 4 with a design supply air temperature of 20°C, the modeled prediction under steady state conditions was modeled to be close to ~24.7°C throughout the day. The measured room operative temperature was 23.3°C. This shows good agreement between the design temperature and the laboratory test, though somewhat lower than that predicted in the thermal model. Case 5, with a supply air temperature of 18°C had a measured room operative temperature of 20.7°C. For these tests, more ankle level coolness was observed with a higher DR value, although this is considered to be within acceptance criteria. This suggests that the opportunity exists to either extract more load than currently intended, or to reduce the supply air volume. The system is also robust in its performance at higher internal loads, or by providing lower indoor operative temperatures should this become preferred.

The CC did not demonstrate over-cooling during a lunchtime scenario when internal loads were reduced and the radiant cooling maintained. The radiant panel temperature was held constant to simulate a concrete ceiling, which would respond slowly. The temperature was seen to glide gradually down to 22.2°C over a two hour period where it stabilized, well within the target temperature range.

In all, this study demonstrated the effective use of a hybrid system and presented useful analytical tools that can be used with confidence early in schematic design. The laboratory testing aided in the validation of design assumptions and modeling, while offering an opportunity to help the client better understand the system and the expected high level of indoor environmental quality.

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