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Effect of acoustical clouds coverage and air movement on radiant chilled ceiling cooling capacity

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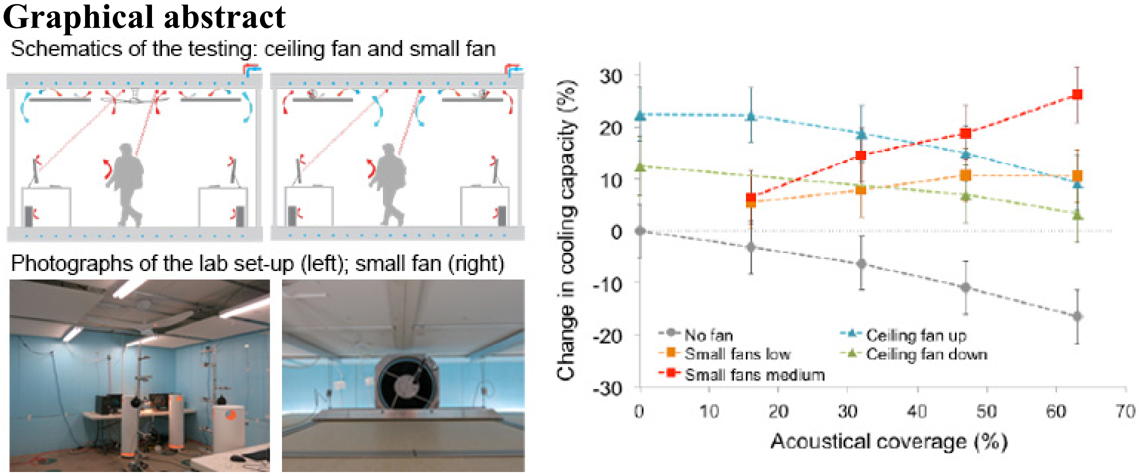
Abstract

Thermally activated building systems have the potential to achieve significant energy savings, yet, the exposed concrete may also create acoustical challenges due to the high reflectivity of the hard surface. Free-hanging acoustical clouds reduce the acoustical issues, but also the cooling capacity of a radiant chilled ceiling system. Fan-induced air movement can be used to compensate for the cooling capacity reduction. We experimentally assess the combined effect of acoustical clouds and fans on the cooling capacity for an office room. We installed a ceiling fan between the clouds (blowing in the upward or downward direction) and small fans above the clouds (blowing horizontally) at the ceiling level to increase the convective heat transfer along the cooled ceiling. We tested the different fan configurations against a reference case with no elevated air movement. The tests conducted without fans showed that cooling capacity decreased, but only by 11%, when acoustical cloud coverage was increased to 47%, representing acceptable sound absorption. The ceiling fan increased cooling capacity by up to 22% when blowing upward and up to 12% when blowing downward compared to the reference case over the different cloud coverage ratios. For the variants with small fans, cooling capacity increases with coverage, up to a maximum increase of 26%. This experiment proves that combining fans with acoustical absorbents close to the radiant surface increases cooling capacity while simultaneously providing improved acoustical quality, and quantifies the impact.

Graphical abstract



1 Introduction

Radiant hydronic cooling systems operate at relatively high temperature cooling and with low temperature differences between the space and the cooling surfaces. These characteristics allow these systems to be regarded as energy efficient [1]–[6]. Yet, the low temperature difference also requires them to cover large surface areas to ensure sufficient heat exchange with the space. Radiant systems are typically installed on the largest room surfaces (e.g., ceilings or floors) that are kept uncovered to preserve the prevailing radiant heat exchange between the conditioning surface and the heat sources, and other surfaces in the space. In the case of a commercial building, this is in direct conflict with the need to add acoustic absorption in the ceiling plane to improve the acoustic quality of the space. To address this conflict, researchers have studied the effect of free-hanging acoustical panels on cooling capacity and sound absorption of a room with a radiant ceiling [7]–[13]. Partial coverage of a radiant chilled ceiling with free-hanging clouds did not cause a proportionally equivalent reduction of the cooling capacity; instead, the reduction in cooling capacity is 3-4 times lower than the percentage cloud coverage (defined as the area of the clouds, perpendicularly projected on the ceiling, divided by the total ceiling area) depending on the cloud configuration. Our recent testing confirmed a similar trend and showed that there is only an 11% reduction in cooling capacity coefficient for 47% cloud coverage. The acoustical testing showed that if acoustical clouds covered 30% of the ceiling in a private office and 50% in an open plan space, acceptable sound absorption was achieved [8].

The present study focuses on the change in cooling capacity of an office room with a radiant cooled ceiling and a combination of free-hanging acoustical clouds and fans. We wanted to investigate the effect of air movement on radiant ceiling performance as it potentially provides two key advantages: increasing convective heat exchange at the chilled ceiling surface, and providing the option of having elevated air movement in the occupied zone for improved thermal comfort.

We conducted a review on studies that quantify the effect of air movement on cooling capacity. We did not find any studies that investigated the combination of fans below a radiant ceiling. Yet, we found studies on the effect of ventilation (type of air diffusers) on radiant ceiling capacity. Awbi and Hatton [14] tested the effect of convection for a heated floor partially covered by an air jet (the convective performance of a cooled ceiling has often been compared to that of a heated floor and therefore we decided to report this study). They modelled the total convective heat transfer coefficient based on natural and forced components. For the natural component, the correlation was based on the temperature difference between the mean panel temperature and the air, and for the forced component, the correlation was based on the width of the nozzle and the diffuser discharge air speed. Jeong and Mumma [15] modelled the effect of mixed convection on radiant ceiling cooling panels due to a nozzle diffuser at the ceiling level. They developed equations for the total heat transfer and radiant heat transfer coefficient based on surfaces and air temperatures. They found that, under normal operating temperatures, the total capacity of the panels can be enhanced by 5%-35% depending on the air speed. Novoselac et al. [16] tested the effect of mixed convection due to the presence of a high aspiration diffuser. For natural convection, they found a convection correlation that was based on the temperature difference between the cooled ceiling surface and the air. For forced convection, a correlation between the total heat exchange coefficient and the air change was established. Their results showed an increase of the total convection coefficient at cooled ceiling surfaces by 4%-17% with high aspiration diffusers. Venko et al. [17] studied the effect of natural and mixed convection along a thermally activated cooled wall (vertical configuration). Forced convection was generated from an air jet flowing downward through a longitudinal ceiling slot parallel to the cooled wall. The forced convection coefficient was a function of the height on the wall (distance to diffusor). It was correlated with the temperature difference between the cooled wall surface and the air, the height and the air speed. The increase in cooling capacity was not quantified. Overall, these studies show promising opportunities for the use of increased air movement on radiant cooled ceiling heat transfer. We could not find studies directly involving fans, which shows a gap in the literature. Moreover, we did not find papers on the interaction among acoustical panels, fans and radiant systems.

Using fans to bring air movement in the occupied zone presents multiple advantages in terms of thermal comfort and associated energy use. Elevated air speed is accepted as an effective strategy to cool people in moderately warm environments [18]–[26]. It allows equivalent thermal comfort conditions to be achieved at higher indoor air temperatures [27], [28]. This offset of warm indoor temperature has been identified as a relevant potential for energy savings as it can be used to reduce the need for compressor-based conditioning systems [29]–[33]. Additionally, if the fan is controlled by occupants, it provides the possibility to resort to air movement when needed which allows for more flexibility in the approach to thermal comfort and personal preferences.

The combination of fans and radiant systems may provide numerous advantages. The objective of this study is to determine the cooling capacity one can reach when adding both free-hanging acoustical clouds and increased air movement at the ceiling level. This paper focuses on air movement as a strategy to offset and increase the cooling capacity. We investigated the effect of two different strategies to provided increased air movement close to the radiant panels: (1) using a ceiling fan located in the center of the chamber that can operate in either upward or downward directions and (2) using a series of small fans located above the clouds (see schematics in Figure 1). We conducted laboratory experiments in a controlled climatic chamber equipped with a radiant cooled ceiling, fans and free-hanging clouds. This testing follows a first study with no air movement and detailed acoustical analysis [8]. The earlier series of tests with no air movement served as the reference case for our testing.

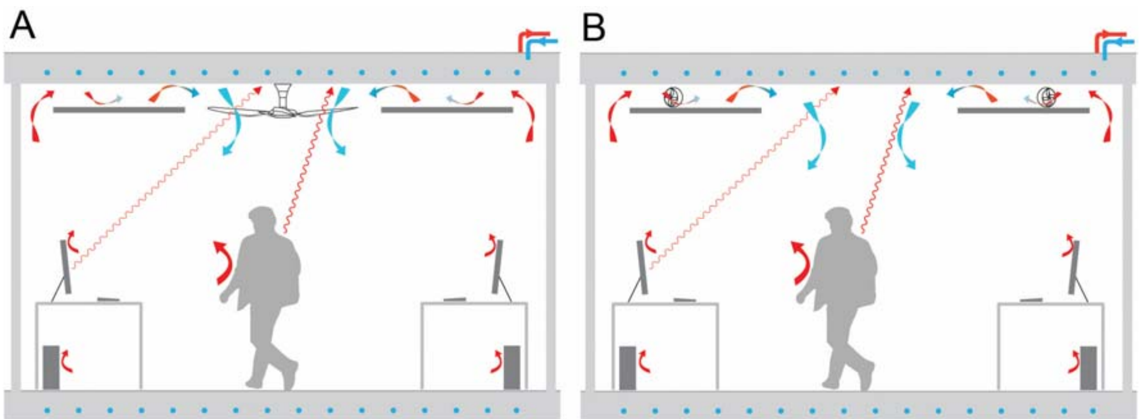


Figure 1: A: Schematic with ceiling fan blowing downwards from the ceiling – B: Schematic with small fans blowing along the ceiling (towards the center)

2 Method

2.1 Experimental facilities and room description

The chamber used for this experiment has a floor area of 18.2 m^2 and a volume of 54.7 m^3 (4.27 m × 4.27 m × 3.0 m). It is nearly adiabatic, has no windows and is located inside a large laboratory facility that was maintained at 21.6 ºC ± 0.5 ºC during our testing. The chamber's construction, thermal properties, radiant panels and acoustical clouds used for this testing are the same as described in our previous study [8]. We conducted our experiment in September 2015. The chamber is accredited by the EN 14240 [34] for cooled ceiling testing. Figure 2 shows a plan, sections and photographs of the chamber.

We used exactly the same radiant panels and the same acoustical clouds (type, size, location, thermal properties and set-up) for this testing as for the testing reported in [8]. Twelve radiant ceiling panels were centered on the ceiling to cover 73.5% (13.4 m^2) of the total ceiling area. This represented the maximum area that could be covered by the panels given the chamber ceiling design. The intention was to represent TABS, the system of interest in this study, as closely as possible by covering the largest possible ceiling area. For this study, a maximum of 8 clouds could be positioned on the ceiling (as represented in Figure 3). Acoustical coverage was one of the variables in this study and therefore the number of clouds is not constant across the tests. Figure 3 represents the various coverage ratios tested. The chamber did not include an air system because we wanted to specifically focus on the change in cooling capacity of the radiant cooled ceiling, and adding an air system would increase overall uncertainty in the results.

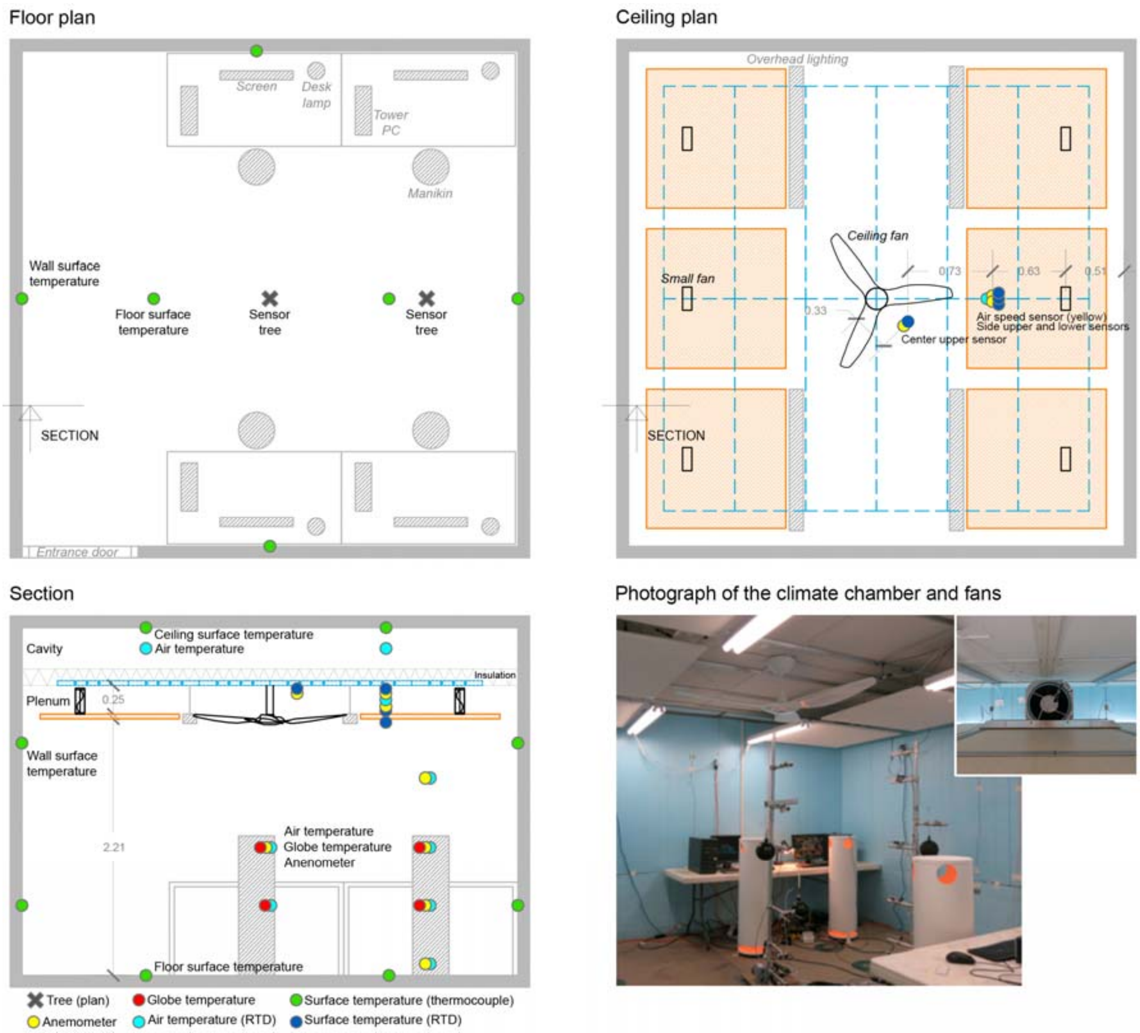


Figure 2: Plan, section and photographs of test chamber with sensors and cloud coverage. The acoustical clouds are in orange, the radiant ceiling in blue and the furniture/equipment in gray. The fans are represented in black: ceiling fan in the center and small fans above the clouds. This figure represents the variants with 47% ceiling coverage (6 clouds). The two photographs of the testing show the test chamber and one small fan mounted above a cloud.

In this study we will often refer to the 'occupied zone' typically defined as the volume of air confined by horizontal planes and by vertical planes [35]. We will use as horizontal planes the floor and an upper horizontal plane at 1.8 m above the ground (typical height used for a standing occupant). We will use as vertical planes an imaginary inner room 0.5 m away from the walls of the chamber.

Table 1 summarizes the heat sources used in the experiment. We modeled office heat sources using computers (tower CPUs), flat screens and desk lamps on the desks, and suspended overhead lighting (0.25 m below the radiant ceiling). We simulated occupants with power-adjustable dummies according to EN 14240 [34]. When fully installed, the test chamber represented a four-person office with a computer at each workstation; a relatively high occupant density of 4.55 m^2 per person. The data acquisition system was outside the chamber, and therefore not listed here as a heat load.

Table 1: Heat load summary

Heat source	Number	Total power (W)	Power per floor area (W/m^2)
People	4	300	16.5
Computers	4	300	16.5
Desk lamps and screens[(1)]	4	283	15.5
Overhead lighting	4	214	11.8
Total		1097	60.3

[(1)] For some of the small fan variants, the power of the desk lamps and screens was sometimes used to compensate for the added power generated by the small fans. This power adjustment was done by switching off devices.

Table 2: Fan test conditions and power

Fan	Configuration	Fan rotational speed (rpm)[(6)]	Fan setting	Number of fans	Total fan power measured[(1)] (W)	Added load in the room (%)	Adjustment[(2)] (W)
Ceiling	Blowing up	146	speed 4	1	6	0.6%	0
	Blowing down	146	speed 4	1	6	0.6%	0
Small	Low speed	648	35%[(3)]	2[(4)]	22[(5)]	2%	0
	Medium speed	832	40%[(3)]	2[(4)]	26[(5)]	2.4%	-21
	Low speed	648	35%[(3)]	4[(4)]	43[(5)]	4%	-30
	Medium speed	832	40%[(3)]	4[(4)]	57[(5)]	5.2%	-30
	Low speed	648	35%[(3)]	6[(4)]	65[(5)]	6%	-30
	Medium speed	832	40%[(3)]	6[(4)]	87[(5)]	7.7%	-58
	Low speed	648	35%[(3)]	8[(4)]	87[(5)]	7.7%	-58
	Medium speed	832	40%[(3)]	8[(4)]	115	9.3%	-58

[(1)] Power measurements reported for the small fan testing were corrected based on a calibration conducted after the testing; [(2)] Adjustments were made to compensate the fan output power; they were based on the power measurements conducted during the testing (prior to more accurate calibration of the power meter); [(3)] Based on the variable transformer input; [(4)] Based on the number of clouds; [(5)] The uncertainty of the power meter increases at low values; [(6)] Rotational speed for the ceiling fan was given by the manufacturer; rotational speed for the small fan was measured using a strobe (Nova-Strobe Basic Battery LED Stroboscope from Monarch Instruments) and averaged.

We equipped the climatic chamber with fans. Figure 2 shows a ceiling plan of the chamber (with 6 clouds) and photographs of the two types of fan. The type and configuration of fans was the second variable of this study. We used the two following models in our experiment:

(1) DC motor ceiling fan (Haiku 52” - 1.32 m diameter from Big Ass Fans) located in the center of the room in between the clouds. This fan can blow air either towards the room (downward direction) or towards the ceiling (upward direction). We conducted preliminary testing and decided to conduct all our testing at 146 rpm that corresponded to speed 4 (for both directions) because: (a) it provided acceptable average air velocities for comfort in the occupied zone (< 0.8 m/s as prescribed by ASHRAE Standard 55 [27]); (b) it represented the mid-speed between 1 and 7; (c) it was the maximum upward speed and we wanted to be able to directly compare it with the same downward speed; and (d) we ran the experiment at fixed speed as we had limited experimental time, which was insufficient to investigate a fourth factor thoroughly.

(2) Small/compact AC axial fans (W2E200-HK86-01 - 0.20 m diameter from Ebm-Papst) mounted above the clouds and blowing air parallel to the ceiling. We controlled the fan speed with a variable transformer (1032, Control Concepts, USA). During preliminary testing, we discovered that the maximum speed of this fan exceeded our needs and we decided to run our experiment at 648 and 832 rpm (35% and 40% of the maximum setting, respectively), which we defined as ‘low’ and ‘medium’ speed, respectively.

Table 2 reports the fan test conditions. While the ceiling fan at the selected speed was extremely efficient in terms of energy, the small fans were less efficient, and brought an additional source of power within the chamber. We chose the small fan model based on their compatibility with the variable transformer used for this chamber and we realized much later that they were not energy efficient. There are far more efficient small (DC) fans available in the market that would be used in a final product, once the desired airflow is known, and the constraint to operate at a variable AC voltage is no longer necessary. Thus, to maintain a similar total heat load level during all experiments, we turned off selected office equipment (e.g., desk lamps, computer screens) that was approximately equal to the amount of extra heat added by the additional inefficient small fans used in this experiment. This adjustment was based on original power measurements conducted during the testing. Yet, these original measurements have proved to be inconsistent and the power meter was recalibrated after the testing. The value reported in Table 2 are the corrected values. We do not have the uncertainty of the power meter used to control the small fan.

2.2 Variant description

Figure 3 represents the location of the clouds and fans for different levels of acoustical cloud coverage: 0% (0 clouds), 16% (2 clouds), 32% (4 clouds), 47% (6 clouds) and 63% (8 clouds). We tested four different fan configurations: (1) ceiling fan blowing upward, (2) ceiling fan blowing downward, (3) small fans at low speed and (4) small fans at medium speed. For the test with small fans and 8 clouds, we oriented the fans toward the center of the room to avoid the corner fans blowing perpendicular to other fans. Table 3 summarizes all the experiments.

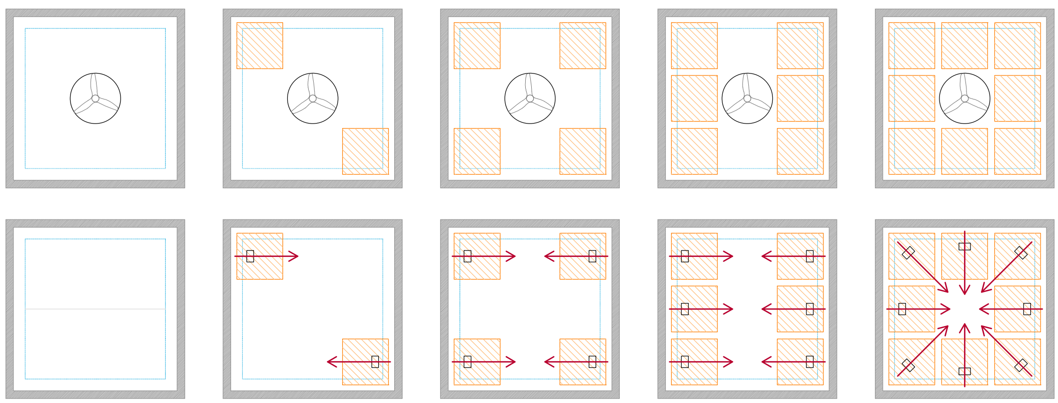


Figure 3: Acoustical coverage and fan configurations. The grey squares represent ceiling views. The orange hatched surfaces are the acoustical clouds (0%, 16%, 32%, 47%, 63% coverage). The upper row shows the variants with ceiling fan and the lower row the variants with small fans (the arrows on the small fan represent the blowing direction)

Table 3: Sequence of experiments and testing conditions

Test sequence	Coverage	Number of fans	Type of fan	Fan setting
1	63% coverage	8	Small fan	Medium speed (832 rpm/fan)
2	63% coverage	1	Ceiling fan	Upward blowing (146 rpm)
3	63% coverage	-	No fan	-
4	63% coverage	1	Ceiling fan	Downward blowing (146 rpm)
5	47% coverage	6	Small fan	Medium speed (832 rpm/fan)
6	47% coverage	6	Small fan	Low speed (648 rpm/fan)
7	47% coverage	1	Ceiling fan	Downward blowing (146 rpm)
8	47% coverage	1	Ceiling fan	Upward blowing (146 rpm)
9	47% coverage	6	No fan	-
10	0% coverage	1	Ceiling fan	Downward blowing (146 rpm)
11	0% coverage	-	No fan	-
12	0% coverage	1	Ceiling fan	Upward blowing (146 rpm)
13	32% coverage	4	Small	Medium speed (832 rpm/fan)
14	32% coverage	4	No fan	-
15	32% coverage	1	Ceiling fan[3]	Upward blowing (146 rpm)
16	32% coverage	4	Small fan	Low speed (648 rpm/fan)

17	16% coverage	2	Small fan	Low speed (648 rpm/fan)
18	16% coverage	-	No fan	-
19	16% coverage	1	Ceiling fan(3)	Upward blowing (146 rpm)
20	16% coverage	2	Small fan	Medium speed (832 rpm/fan)
21	47% coverage	6	Small fan	Low speed (648 rpm/fan)
22	47% coverage	6	No fan(1)	-
23	47% coverage	6	Small fan	Medium speed (832 rpm/fan)
24	0% coverage	-	No fan(1)	-
25*	0% coverage	-	No fan(2)	-
26*	0% coverage	1	Ceiling fan(2)	Upward blowing (146 rpm)

(1) Duplicated experiment; (2) EN14240 test; (3) Ceiling fan downward blowing for 16% and 32% coverage tests were not conducted

2.3 Experimental conditions and procedures

The experimental conditions and procedures were similar to the ones used for our previous study [8]. This section therefore mainly focuses on the differences in the testing conditions and equations used.

2.3.1 Procedure

As in our first study, we used the operative temperature measured in the center of the room at 1.1 m as our reference temperature to determine the cooling capacity between the ceiling and the room. In order to account for the effect of elevated air movement, we calculated the operative temperature based on the air and mean radiant temperatures and local air speed [36]:

$$t_{op} = \frac{t_{mr}+\left(t_a \cdot \sqrt{10\, v_a}\right)}{1+\sqrt{10\, v_a}} \qquad (1)$$

The mean radiant temperature was derived from the globe temperature with the following equation [36]:

$$T_{mr} = \left[\left(T_{globe}+273\right)^4 + 2.5 \cdot 10^8 \cdot {v_a}^{0.6} \cdot \left(T_{globe}-T_a\right)\right]^{1/4} - 273 \qquad (2)$$

The heat exchange between the radiant ceiling and the room is expressed in Eq. 3, where U_{cc} represents the cooling capacity coefficient in ($W \cdot m^{-2} \cdot K^{-1}$).

$$q_{cc} = U_{cc}\, A_{panels}\left(t_{op} - t_{w,m}\right) \qquad (3)$$

with $t_{w,m} = \frac{t_{w,s}+t_{w,r}}{2}$

On the water side, the radiant system cooling rate is expressed as:

$$q_w = \dot{m}_w\, {c_p}_w(t)\left(t_{w,r} - t_{w,s}\right) \qquad (4)$$

Under steady state conditions, the water in the radiant panels absorbs the electrical power of the heat sources, and thus:

$$P = q_{cc} = q_w \qquad (5)$$

By substituting (2) & (3) in (4), and rearranging:

$$U_{cc} = \frac{\dot{m}_w\, {c_p}_w\left(t_{w,r}-t_{w,s}\right)}{A_{panels}\left(t_{op}-t_{w,m}\right)} \qquad (6)$$

We conducted all tests with the same water mass flow rate (220 kg/h ± 0.02%) and water supply temperature (15 ºC ± 0.01 ºC). The return water temperature stayed between 19.1 ºC and 19.9 ºC. From the cooling capacity coefficient equation, the remaining variable that can influence the cooling capacity is the operative temperature, which is the equilibrium temperature at which the room settles for a given variant of acoustical cloud coverage under steady state conditions.

Steady state conditions were defined as a difference of less than 0.05 °C between the mean of the most recent 60 samples (most recent 30 min) against the mean of the 60 samples immediately prior (30 min immediately prior), for every temperature sensor used in this experiment. We recorded all monitored data once the test had reached steady state conditions, which typically took 300 min from the start of each experiment. After recording the data, we calculated the average temperatures in the occupied zone and the cooling capacity coefficient U_{cc}. This procedure was the same as in our previous study.

2.3.2 Comparison with EN 14240

We compared our approach to determine the cooling capacity of radiant panels to the methodology provided by the European Standard EN 14240 [34]. When maintained at the same temperature, we expect the surface heat transfer from the cooled surface to the room to be the same in the case of TABS as for radiant panels (we are not modelling conduction). Our comparison showed that for the range of temperatures tested, using a linear model was comparable to a power model in predicting the cooling capacity: R^2 of 0.980 for the power model and 0.974 for the linear model for the test without air speed, and R^2 of 0.986 for the power model and 0.985 for the linear model for the test with air speed. This comparison confirmed our approach of using a linear model.

2.3.3 Experimental sequence

The experimental sequence was partially randomized. Yet, to ease the laboratory set-up, we grouped the tests for each cloud coverage level. We started with the tests that included the largest amount of clouds (9, 8 and 6 clouds), then conducted the test without coverage and ended with the tests with lowest coverage (4 and then 2 clouds). Within each coverage, the tests were randomized. We included in the experiments two replications (no fans for 0 and 6 clouds) that we ran at the end of the testing. Table 3 reports the testing sequence.

2.4 Measuring instruments and uncertainty

The measuring instruments and equipment used in this experiment and their accuracy were the same as in our previous study. This testing included additional air velocity sensors that were the same models as the ones previously used. Figure 2 shows the location of all the sensors in plan and section view. We analysed the data in accordance with the ISO 13005 [37] and the JCGM 100 guidelines [38] for the expression of uncertainty. The equation used to calculate the operative temperature included the air speed. Thus, the expression of the uncertainty of the operative temperature is of the form:

$$u_{t_{op}} = \sqrt{\left(\frac{\delta t_{op}}{\delta t_a}\right)^2 {u_{t_a}}^2 + \left(\frac{\delta t_{op}}{\delta t_{mr}}\right)^2 {u_{t_{mr}}}^2 + \left(\frac{\delta t_{op}}{\delta v_a}\right)^2 {u_{v_a}}^2} \qquad (7)$$

with:

$$\frac{\delta t_{op}}{\delta t_a} = \frac{\sqrt{10 \cdot v_a}}{1+\sqrt{10 \cdot v_a}}$$

$$\frac{\delta t_{op}}{\delta t_{mr}} = \frac{1}{1+\sqrt{10 \cdot v_a}}$$

$$\frac{\delta t_{op}}{\delta v_a} = \frac{\sqrt{2.5} \cdot t_a}{(1+\sqrt{10 \cdot v_a}) \cdot \sqrt{v_a}} - \frac{\sqrt{2.5} \cdot (t_{mr} + t_a \cdot \sqrt{10 \cdot v_a})}{(1+\sqrt{10 \cdot v_a})^2 \cdot \sqrt{v_a}}$$

All other uncertainty equations stayed the same as those reported in our previous study. The uncertainty in the cooling capacity coefficient U_{cc} is of the form:

$$u_{U_{cc}} = \sqrt{\left(\frac{\delta U_{CC}}{\delta \dot{m}_w}\right)^2 {u_{\dot{m}_w}}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,s}}\right)^2 {u_{t_{w,s}}}^2 + \left(\frac{\delta U_{CC}}{\delta t_{w,r}}\right)^2 {u_{t_{w,r}}}^2 + \left(\frac{\delta U_{CC}}{\delta T_{op}}\right)^2 {u_{t_{op}}}^2} \qquad (8)$$

with:

$$\frac{\delta U_{CC}}{\delta \dot{m}_w} = \frac{c_{p_w}(t_{w,r} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{w,s}} = \frac{\dot{m}_w \, c_{p_w} \, (t_{op} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{w,r}} = \frac{\dot{m}_w \, c_{p_w} \, (t_{w,r} - t_{op})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

$$\frac{\delta U_{CC}}{\delta t_{op}} = -\frac{\dot{m}_w \, c_{p_w} \, (t_{w,r} - t_{w,s})}{A_{panels}\left(t_{op} - \frac{t_{w,r} + t_{w,s}}{2}\right)}$$

The derived uncertainties for the operative temperature, the total cooling load and U_{cc} were respectively 0.8%, 4.9% and 5.7% (maximum values).

3 Results

Table 4 summarizes the main results of the testing. The chamber reached steady-state equilibrium at an operative temperature between 25.2 ºC (no coverage and ceiling fan up) and 28.0 ºC (63% coverage and no fan).

Table 4: Experimental results

Cov	Fan(5)	t_a(1) (ºC)	t_{mr}(1) (ºC)	t_{op}(1) (ºC)	v(1) (m/s)	$t_{w,r}$ - $t_{w,s}$ (ºC)	$t_{w,m}$ (ºC)	q (W)	q''(2) (W/m²)	q''(3) (W/m²)	U_{cc}(3) (W/(m²·K))	Change(3) (%)
0%	No fan	27.1	26.0	26.5	0.10	4.2	17.1	1069	79.8	58.7	8.46	0.0%
	No fan(4)	26.7	25.7	26.2	0.11	4.1	17.0	1038	77.5	57.0	8.44	-0.3%
	Clg-Dn	25.4	26.1	25.6	1.21	4.2	17.1	1076	80.3	59.1	9.51	12.4%
	Clg-Up	25.3	25.1	25.2	0.40	4.4	17.2	1113	83.1	61.2	10.36	22.4%
16%	No fan	27.2	26.4	26.8	0.12	4.2	17.1	1067	79.7	58.6	8.19	-3.2%
	Clg-Up	25.3	25.3	25.3	0.37	4.4	17.2	1120	83.6	61.5	10.35	22.3%
	Small-L	26.3	25.9	26.1	0.05	4.2	17.1	1075	80.3	59.1	8.93	5.5%
	Small-M	26.3	25.9	26.1	0.06	4.2	17.1	1079	80.5	59.3	9.00	6.4%
32%	No fan	27.2	26.7	27.0	0.12	4.1	17.1	1054	78.7	57.9	7.93	-6.2%
	Clg-Up	25.5	25.7	25.6	0.41	4.4	17.2	1129	84.3	62.0	10.04	18.7%
	Small-L	26.0	25.9	25.9	0.09	4.2	17.1	1079	80.5	59.3	9.12	7.8%
	Small-M	25.5	25.7	25.6	0.15	4.3	17.1	1097	81.9	60.2	9.69	14.6%
47%	No fan	27.9	27.3	27.6	0.07	4.1	17.1	1061	79.2	58.3	7.54	-10.9%
	No fan(4)	27.5	27.2	27.3	0.07	4.1	17.0	1040	77.6	57.1	7.54	-10.9%
	Clg-Dn	25.9	26.7	26.1	0.90	4.2	17.1	1085	81.0	59.6	9.06	7.1%
	Clg-Up	25.7	26.1	25.8	0.36	4.4	17.2	1124	83.9	61.8	9.72	14.9%
	Small-L	25.7	26.0	25.8	0.24	4.3	17.1	1090	81.4	59.9	9.36	10.7%
	Small-M	25.4	25.9	25.6	0.33	4.4	17.2	1125	84.0	61.8	10.05	18.8%
63%	No fan	28.0	28.1	28.0	0.08	4.1	17.0	1041	77.7	57.2	7.06	-16.5%
	Clg-Dn	25.7	26.9	26.0	0.85	4.1	17.0	1048	78.2	57.6	8.75	3.4%
	Clg-Up	25.6	26.2	25.8	0.35	4.2	17.1	1077	80.4	59.2	9.25	9.3%
	Small-L(6)	26.3	26.3	26.3	0.39	4.4	17.2	1137	84.9	62.5	9.35	10.5%
	Small-M(6)	26.1	26.4	26.1	0.72	4.8	17.4	1239	92.5	68.1	10.67	25.5%

(1) Measured at the central tree at 1.1 m; (2) Per unit of panel area; (3) Per unit of floor area; (4) Replicated experiment; (5) Fan acronyms: 'Small-L' and 'Small-M' stands for small fan low and medium speeds, 'Clg-Up' and 'Clg-Dn' stands for ceiling fan blowing up and down; (6) Small fans blowing towards the center of the room (not set parallel to each other)

3.1 Cooling capacity

Figure 4 shows the change in cooling capacity coefficient, U_{cc}, for all tests as a function of the acoustical coverage. For the variants with no fan, adding coverage reduces the cooling capacity. Yet, the reduction in cooling capacity is on average a factor of 4-5 times lower than the percentage increase in cloud coverage. A detailed description of this result and a comparison with the literature is reported in [8]. The variants with ceiling fan have a similar slope as the reference case. However, the ceiling fan (for the speed and type tested) brings an increase in the cooling capacity of 22-26% when blowing up and 12-20% when blowing down compared to the reference case over the same range of coverage. When the ceiling fan is used, the increase in cooling capacity is substantial. For both blowing directions, even with the highest coverage of 63%, the change in overall cooling capacity is positive compared to the reference case (no fan and no coverage). The variants with small fans also show a significant increase in cooling capacity compared to the reference case (up to 26% increase). We note that the slope here goes in the opposite direction: the cooling capacity increases with increasing coverage. We can conclude that the combination of a small fan on top of a cloud brings an increase of cooling capacity compared to the bare ceiling. Increasing the air speed of the small fans brings an additional increase in cooling capacity. Yet, the two curves (for small fan low and medium speeds) do not show equal slopes: at 63% coverage, the slope for the medium speed increases while the slope for low speed flattens. For this last coverage, we oriented the fans toward the center of the room (see Figure 3) and a possible explanation for this may be a stronger increase in convection occurring at this speed. Overall the results of the testing (for both fans) demonstrate that we can increase the acoustical coverage to provide the needed sound absorption while simultaneously increasing the cooling capacity of a radiant ceiling. Section 3.4 compares the increased cooling capacity and increased fan power.

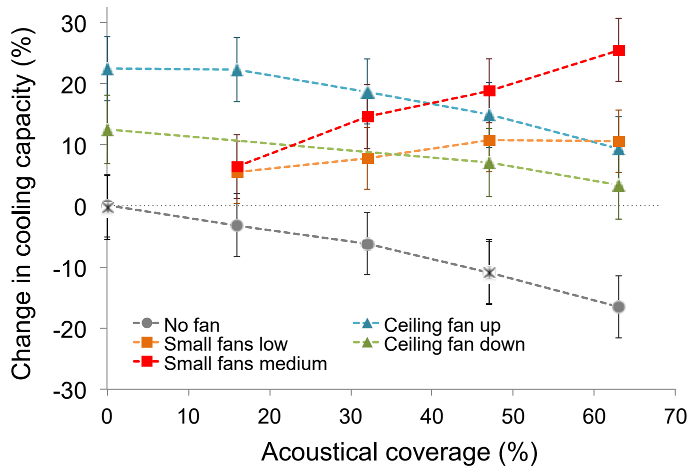


Figure 4: Change in cooling capacity coefficient as a function of acoustical cloud ceiling coverage. Three configurations are reported: (1) no fans, (2) small fans, and (3) ceiling fan. The black "x" represent the replications

Our testing conditions did not allow for a detailed analysis of the two components (convection and radiation) of the heat transfer coefficient. To do so, our set-up would have involved heat flux sensors and more surface temperature and air speed sensors close to the ceiling. We decided not to investigate this point due to budget and sensor constraints and due to its limited value on practical applications. Its relevance is mainly from the theoretical point of view, and this was not a priority in this work. This question is further addressed in section 5 (study limitations and future directions).

3.2 Air speed in the ceiling plenum

Figure 5 shows the air speed in the ceiling plenum for a selected set of variants. We measured air speed with three sensors whose locations are reported in Figure 2. Due to the limited number of speed measurements, we have only used these results to help interpret the overall magnitude of air movement generated by the different variants and fan types. Our observations assume that air speeds measured by the upper sensors (closer to the radiant panels) will more closely correlate with changes in the cooling capacity from the ceiling. For the variants with no fan, all measurements in the plenum stayed within 0.10 - 0.19 m/s. For the ceiling fan variants, the measured air speed on the upper center sensor (above the blades) stayed at around 0.75 m/s for both blowing directions and coverage. The air speed above the upper side sensor (above one lateral cloud) measured air speeds of 0.60 and 1.13 m/s for the ceiling fan in downward and upward directions respectively. We found that for all ceiling fan tests, the coverage has only a small effect on air speed. If we look at the small fans, the lower side sensor shows a higher air speed than the upper sensors. However, the opposite would be more beneficial to increase cooling capacity. For both the low and medium speeds, the 47% coverage shows overall higher air speed than the 63% coverage, especially for the lower sensor. At 63% coverage, the small fans with low speed have an average air speed of 0.38 m/s for the two upper sensors while the small fans with medium speed average 0.65 m/s. We can conclude that the medium fan speed brings substantially more air flow close to the ceiling. With a denser distribution of small fans at the ceiling level, the medium speed can bring a large increase in cooling capacity.

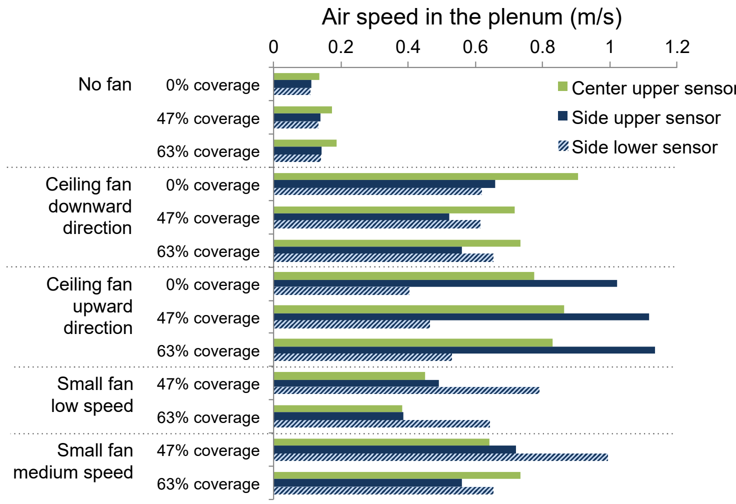
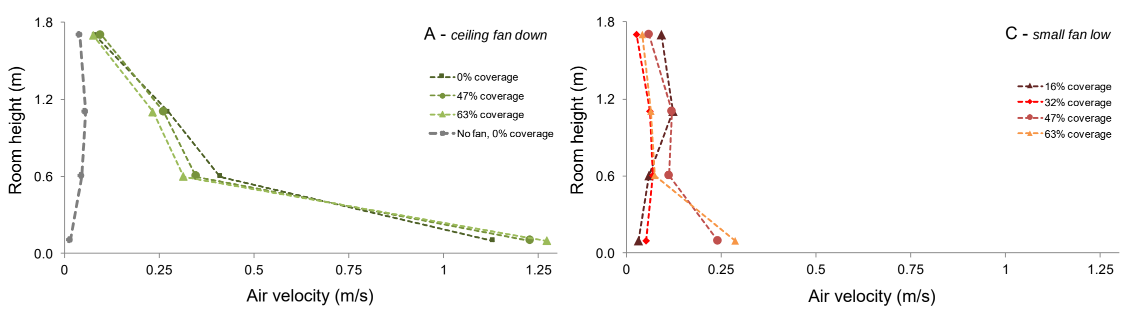


Figure 5: Air speed in the plenum for a selected set of variants. The locations of the sensors are detailed in Figure 2. Lower and upper sensors are within the plenum at 0.17 m and 0.08 m from ceiling respectively (or 0.08 m and 0.17 m from upper side of the acoustical cloud).

3.3 Air speed in the occupied space and its impact on thermal comfort

We measured the thermal comfort parameters at two locations (see Figure 2, center and side trees). The side tree is more representative of a typical occupant location underneath an acoustical cloud, so we conducted our analysis for this location. We measured the air speeds at the different heights typically used in thermal comfort assessment (0.1, 0.6, 1.1 and 1.7 m). Figure 6 presents the results of this analysis for the different fan variants and coverages. The air speed profile for the ceiling fan variants confirms what we would expect from these fans: in the downward direction, the fan pushes the air towards the room and the highest speed is observed nearby the floor (when it meets an obstacle). The ceiling fan in the upward direction is commonly used to counteract stratification. For this variant, the air speed is between 0.10 and 0.33 m/s throughout the measurement locations and the highest speed is observed at 1.1 m height. Figure 6 (C), shows the readings for the small fan at low speed. All the measurements between 0.6 and 1.7 m height stay below 0.12 m/s and the highest air speed is observed nearby the floor for the highest coverage and it reaches 0.29 m/s. In ASHRAE 55, an air speed of 0.2 m/s is the minimum speed to be allowed to cause a cooling effect due to elevated air speed (based on [27]). We can conclude the air speeds for the small fan at low speed have a negligible effect on thermal comfort in the occupied zone. Similar conclusions apply for the small fan at medium speed with 16%, 32% and 47% coverage (see Figure 6 (D)). For 63% coverage, the velocities at 0.1 and 1.1 show an increase of about 0.35 m/s compared to the velocities at 0.6 and 1.7. The heights of 0.1 and 1.1 m can be associated with ground and desk height near the tree. As this variant brought the highest air speed at the plenum level (compared to other small fan variants), we can hypothesize that this irregular profile may be related to more complex air flow patterns and obstacles.



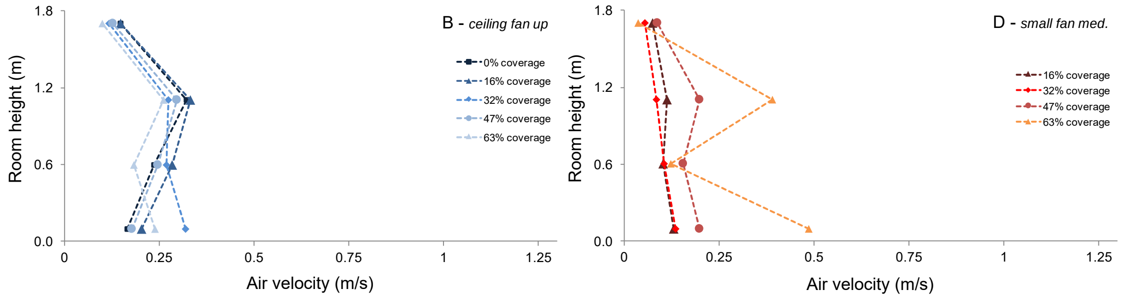


Figure 6: Speed profiles for the different variants and the different height measured at the side tree. This graph comprises 4 parts: (A) base case and ceiling fan blowing down for different coverage, (B) ceiling fan blowing down for different coverage, (C) small fans low speed for different coverage, (D) small fans medium speed for different coverage.

Figure 7 shows how air speed affects the thermal comfort zone. For this assessment, we assumed a seated occupant performing office work (1.1 met) with a total clothing insulation of 0.7 clo (wearing a short sleeve shirt, long trousers, socks, and business shoes (0.55 clo) and sitting in an office chair with a cushion seat and mesh back (0.15 clo)). We used a relative humidity of 50%. For each variant, we calculated the average air speed of 3 heights (0.1, 0.6, 1.1m) representing the occupied zone of a seated occupant using the measurements recorded on the side tree. We used this average air speed as input to the CBE Thermal Comfort Tool [39] to determine the center of the comfort zone based on the following: (1) the PMV model when air speed is below 0.2 m/s, and (2) the PMV+SET models when the air speed is above 0.2 m/s [27], [28]. For simplicity, we used the operative temperature output as the center of the thermal comfort zone. Figure 7 reports acceptable ranges of operative temperature as a function of air speed for each variant. This graph shows that the ceiling fan blowing down allows an increase in the operative temperature of 2.9 ºC compared to the base case with no fan and no coverage. This increase is due to elevated air speed in the occupied zone. The literature has identified this as a relevant potential for energy savings [31], [33]. Increasing the cooling setpoint temperature will likely have a larger savings effect for surface-conditioning systems as they operate over a far smaller temperature differential than traditional air-conditioning systems, and thus an increase of 1 ºC has a proportionally larger effect. Additionally, in many climates, increasing the cooling set point temperature will enable low energy cooling technologies, such as an evaporative cooling tower only approach instead of compressor based cooling, to be used where they would otherwise not be feasible.

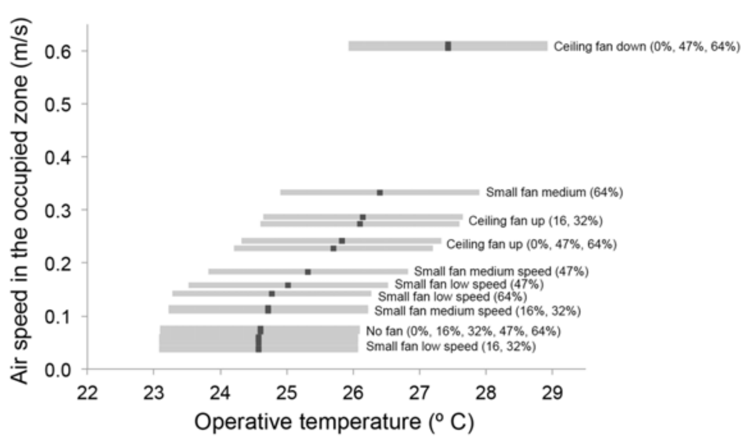


Figure 7: Acceptable ranges of operative temperature for different variants based on the average air speed for a seated person measured at the side tree at 0.1, 0.6 and 1.1 m. Assumptions for calculation: 0.7 clo, 1.1 met, RH 50%

Based on Figure 6 (A), we note that the highest speeds for ceiling fan blowing down occur for the velocities at the ankle level, which may skew the results to a higher value while bringing new concerns in terms of thermal comfort [40]. Depending on the coverage, the ceiling fan blowing up allows an increase in operative temperature of 1.3-1.6 ºC to keep comparable thermal comfort as the base case. It also provides a lower and more uniform air speed at each of the measurement heights, avoiding potential concerns of high velocities at ankle level, or at the head when an occupant is directly below the fan jet. Additionally, having a ceiling fan that has a user selectable direction allows the user to have control over the air speed in the occupied zone, while still maintaining an increase in radiant system cooling capacity. The small fans at medium speed and 63% coverage show the highest air speed of all small fan tests. Yet, based on Figure 6 (D), the air speed profile for this variant is irregular and difficult to explain. It is possible that the walls of the test chamber and the positioning of the desks may have an impact on air speeds in the occupied zone. The other variants using small fan (low and medium speed, all coverages considered) are all below 0.2 m/s and therefore, the effect of air movement in the case of small fan is nearly negligible from a thermal comfort perspective. ASHRAE Standard 55 removed the draft risk model present in EN ISO 7730 (2005) [41], therefore we did not use it. The conditions reported here do not violate the draft local discomfort requirements specified by ASHRAE Standard 55.

3.4 Fan energy implications

Using fans to increase the cooling capacity and potentially expand the comfort zone also has an energy penalty related to the energy consumption of the fan. We looked at this parameter in the results output of our measurements. The ceiling fan used a DC motor and was particularly efficient in terms of energy usage (6 W or 0.3 W/m^2 when normalized by the room area). If we compare the change in cooling capacity to the change in heat losses from the ceiling fan, we are still benefiting from this strategy for all cases. By contrast, the small fans (these were chosen based on their maximum volumetric air flow and operability with the variable transformer of the chamber) used an AC motor and were not energy efficient models. We measured up to 115 W (or 6.3 W/m^2) heat added to the room for the highest fan usage (with 8 small fans at medium speed). This change in heat loads brought a higher penalty to the room conditions than the cooling benefit of increased cooling capacity. Observing this difference in fan power usage between models leads to one of the key conclusion of this study: the use of low-energy (DC) model fans is fundamental to provide energy benefits from increased air movement.

4 Discussion

In this study, we found the cooling capacity coefficient increases for all variants with fans compared to the reference case with no acoustical clouds and fans. The increase reached up to 22% for ceiling fan variants and up to 26% for small fan variants. We did not find prior studies that assessed the effect of fans nearby a radiant cooled ceiling but we found multiple studies have investigated the effect of ventilation systems located nearby a radiant ceiling. Depending on the diffuser and ventilation setting, these configurations brought an increase of the ceiling heat transfer coefficient from 17% (high aspiration diffusor) up to 35% (nozzle diffuser). Although these ventilation set-ups differ from our test configuration, we note a comparable order of magnitude in the results.

We conducted this experiment in a room without a ventilation system to reduce measurement uncertainty. The overall goal was to provide a proof-of-concept for the combined solution of adding acoustical clouds (reducing cooling capacity) and adding fans (increasing cooling capacity). The results are encouraging and warrant further testing and applications that will necessarily include a secondary air distribution system. For this experiment, we decided to focus on cooling capacity and thermal comfort by varying coverage and testing air movement strategies. Adding an air distribution system to our configuration introduces multiple questions, including the type of system (stratified or mixing), the type of diffusers and their location compared to positions of the fan and clouds. As seen in the literature, an overhead system will bring one additional source of air movement that may be nearby the plenum, which may increase the convective heat transfer even more. Combining the different systems at the ceiling will require design integration, both to address air flow dynamics and aesthetic aspects.

The combination of radiant ceiling with stratified ventilation systems (underfloor air distribution or displacement ventilation) have shown to be a reliable solution for energy and comfort aspects [42]–[44]. Yet, as fans at the ceiling will disturb stratification, it is likely that the tested configuration may not work adequately with stratified systems, unless the mixing is confined to a region close to the ceiling. In this regard, small fans at lower speeds or reduced coverage could be compatible as they do not affect the air flow patterns in the occupied zone as much as the ceiling fan variants.

The use of fans has an impact on energy use and fan selection requires careful attention. DC fans can provide a low-energy solution [32], as shown with the ceiling fan used for this experiment. For the small fan variant, there are far more efficient products available than used in this experiment. Depending on their type and design, 0.20 m diameter computer fan models have the potential to offer a low-energy, but also low-cost and silent option (e.g., Antec Big Boy 200 - 200mm Tricool Computer Case Fan). Such types may be worth investigating in the context of practical application or future testing.

The use of fans will bring an additional source of noise in the space. We did not conduct sound pressure level measurement or sound power to determine its impact. Yet, for the types of fans and ranges of speed tested, current low-energy DC fans are much quieter than conventional AC models. Further, radiant systems are sometimes criticized for the lack of 'background noise' due to reduced mechanical systems in the space. In some cases, white noise generators (e.g., active sound masking) are installed to improve sound privacy of spaces with radiant systems. These points lead us to believe in minimal disturbance (if not a benefit) of a moderate fan noise added to the space.

This experiment was conducted for a medium sized office room. In larger spaces (such as open plan offices), we can expect that additional fans would be installed and that the absence of walls would lead to different air flow patterns. Yet, we can also hypothesize that the elevated air speed near the ceiling would bring comparable increases in cooling capacity as the ones observed in our testing. Acoustical clouds and fans could also be a good option for other type of environments (such as conference rooms, patient rooms, etc.).

5 Study limitations and future directions

For the small fan variants, the fans brought an additional source of heat within the chamber located above the acoustical panels and right below the radiant ceiling. The location of this heat source may be another explanation for the higher increase in cooling capacity observed. While we may assume that the convective component of the heat released from the small fans got spread into the full volume of the chamber (because of the well mixed conditions), we may also hypothesize that the radiative component of the heat transfer would mainly stay within the upper part of the room, and be absorbed directly into the radiant panels. The results presented for the small fans may therefore partially overestimate the effect of the small fan at increasing the cooling capacity based on increased air movement. This hypothesis could be verified through additional testing completed with low-energy small fans.

This testing did not focus on the changes in the relative strengths of radiant and convective heat transfer from the radiant ceiling panels. The limited information collected does not allow us to draw conclusive statements on this aspect. Yet, we conducted simplified calculations based on the data available to estimate the relative contributions from convection and radiation. Assuming a constant radiant heat transfer coefficient of 5.5 $W/m^2{\cdot}K$ (as observed in the literature [1], [45]), we derived the values for the convective heat transfer coefficient, which remained below the radiant coefficient for all variants investigated. This simplified calculation deserves confirmation. Nevertheless, it supports the assessment that these combined ceiling systems can still be considered as 'radiant' (i.e., with a minimum of 50% of the load treated by radiation). In practical applications of these design solutions, the share of radiant and convective heat transfer has limited consequences. In other words, the primary outcome of these experiments was a proof of concept that the combined design strategy of radiant chilled ceiling with suspended acoustical clouds and fans (of various types and configurations) has a high probability to be an effective solution in any building application. Knowing the precise split of radiative and convective heat transfer will not change this result

We conducted this experiment in cooling mode. For heating applications, we expect that the reference cases with no air movement would bring a stronger decrease of heating capacity with added coverage. Convective heat transfer coefficients are lower for a warm vs. a cool ceiling surface due to buoyancy effects on natural convection. Thus, for a heating scenario, adding air movement nearby the ceiling will produce an increase in heating capacity that may be larger than the changes observed for a cooling scenario. The air speed strategy (fan type and mode) will need be chosen to avoid increased air movement in the occupied zone as it is counterproductive to cool occupants with air movement while trying to heat the space. Heating applications would be worth exploring and warrant further testing.

Another aspect of using fans to increase cooling capacity is that it offers a new approach for controlling cooling capacity and thermal comfort in spaces with exposed concrete radiant systems (TABS). Activating fans will increase the heat exchange between the room and the ceiling, and therefore, it offers additional control strategies to compensate for higher loads occurring in the space. Ceiling fans used for increased air motion in the occupied zone can also be used independently from the radiant system. Combining these strategies allows them to be used separately or simultaneously, bringing new ways to improve the control of the systems and occupant comfort, which is worth exploring in future work.

6 Conclusions

We conducted laboratory experiments to investigate the change in cooling capacity of an office room with a radiant cooled ceiling, and a combination of free-hanging acoustical clouds and fans. We tested two types of fans independently: a ceiling fan and a series of small fans located above the acoustical clouds. Compared to the reference case with no acoustical clouds and fans, the cooling capacity increases for all variants that use fans and for all levels of ceiling coverage, therefore, using elevated air movement is an effective strategy to compensate for the small reduction in cooling capacity due to the use of acoustical panels.

For the testing conducted with the ceiling fan the highest increase in cooling capacity happened with no coverage. It reached 22.4% for the upward blowing direction and 12.4% for the downward direction. The coverage has only a small effect on air speed in the occupied space. Elevated air motion in the occupied space can help expand the thermal comfort zone for the ceiling fan variants by about 1.4º C for the fans blowing upward and by 2.9º C for the fans blowing downwards. The increase in cooling capacity caused by the air movement generated by the fan far exceeds the heat gain due to the energy consumed by the fan.

For the testing conducted with small fans the highest increase in cooling capacity reached 26% at highest coverage (63%) for the small fans but elevated air movement in the occupied space was not relevant for thermal comfort purposes. In contrast to the ceiling fan variants, the increase in cooling capacity caused by the air movement generated by the fans does not exceed the heat gain due to the energy consumed by the fans. This is due to the particular model selected for this experiment, and a low-energy DC fan would likely yield a similar outcome as the ceiling fan variant, which used a highly efficient ceiling fan. The small fans provide comparable cooling capacity benefits as the ceiling fan, but without bringing a visible addition to the ceiling layout which may be more aesthetically desirable depending on design contexts.

In this study, we showed that combining acoustical clouds and fans not only offsets the modest reduction in cooling capacity from a radiant cooled ceiling caused by the presence of the clouds, but also provides an overall increase in cooling capacity compared to the reference case with no clouds and fan. This study offers a very promising and practical design solution regarding implementation of radiant slab ceiling systems.

7 Nomenclature

Symbol	Quantity	Unit
A	Area	m²
cp_w	Water pecific heat capacity	$KJ \cdot kg^{-1} \cdot K^{-1}$
h_c	Convective heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
h_r	Radiative heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
q	Heat transfer rate	W
q''	Heat flux (= q/Arad)	$W \cdot m^{-2}$
P	Power	W
$\dot{m}$	Water mass flow rate	$Kg \cdot s^{-1}$
U_{cc}	Cooling capacity coefficient	$W \cdot m^{-2} \cdot K^{-1}$
t	Temperature in ºC	°C
T	Temperature in K	K
t_a	Air temperature	°C
t_{op}	Operative temperature	°C
t_{mr}	Mean radiant temperature	°C
t_{globe}	Globe temperature	°C
$t_{w,s}$	Water supply temperature	°C
$t_{w,r}$	Water return temperature	°C
$t_{w,m}$	Mean water temperature	°C
ΔTw	Water temperature difference (ΔTW = tw,r – tw,s)	K
v_a	air speed	$m \cdot s^{-1}$

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