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Ceiling-Fan-Integrated Air Conditioning: airflow and temperature characteristics of a sidewall-supply jet interacting with a ceiling fan

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Abstract

Ceiling-Fan-Integrated Air Conditioning (CFIAC) is a proposed system that can greatly increase buildings' cooling efficiency. In it, terminal supply ducts and diffusers are replaced by vents/nozzles, jetting supply air toward ceiling fans that serve to mix and distribute it within the room. Because of the fans' air movement, the system provides comfort at higher room temperatures than in conventional commercial/ institutional/retail HVAC. We have experimentally evaluated CFIAC in a test room. This paper covers the distributions of air-speed, temperature, and calculated comfort level throughout the room. Two subsequent papers report tests of human subject comfort and ventilation effectiveness in the same experimental conditions. The room's supply air emerged from a high-sidewall vent directed toward a ceiling fan on the jet centerline; we also tested this same jet on a fan located off to the side of the jet. Primary variables are: ceiling fan flow volumes in downward and upward directions, supply air volume, and room-vs-supply temperature difference. Velocity, turbulence, and temperature distributions are presented for vertical and horizontal transects of the room. The occupied zone is then evaluated for velocity and temperature non-uniformity, and for comfort as predicted by the ASHRAE Standard 55 elevated air speed method. We show that temperatures are well-mixed and uniform across the room for all of the fan-on configurations, for fans both within or out of the supply jet centerline. The ceiling fan flow dominates the CFIAC airflow, and even though non-uniform is capable of providing comfortable conditions throughout the occupied area of the room.

Keywords: Ceiling fan; Sidewall vent; Supply nozzle; Thermal comfort; Air distribution.

1. Introduction

This paper is the first of three describing a new air-conditioning concept we term 'ceiling-fan-integrated air conditioning' (CFIAC, pronounced 'siffy·ac'). The paper gives an overview

of CFIAC objectives, its physical and operational implementation, and the types of questions that will need to be answered to ultimately bring it into general practice. The paper then describes the air speed and temperature conditions produced by CFIAC implemented within a characteristic room, and calculates the expected comfort distribution within the occupied zone. The two subsequent papers will describe parallel studies we have done of the human thermal comfort and ventilation effectiveness of the system.

1.1 Description of system

In CFIAC, terminal supply ducts and diffusers are replaced by vents or nozzles directing jets of supply air toward ceiling fans. The ceiling fans serve to mix and distribute the supply air within the room (two conceptual sketches are presented in Fig. 1 for downward and upward fan flow directions). The ceiling fans also increase the air movement throughout the space, which makes warmer room temperatures comfortable. The supply air volume and temperature are coordinated with the room temperature, the ceiling fans' rotational speed, and the fans' up/down direction, in order to optimize velocity and temperature distributions in the room. The ceiling fans also increase the availability of individual or group comfort control within the room. CFIAC promises more user-adaptable comfort, improved energy-efficiency, and also first-cost and aesthetic benefits coming from eliminating the terminal ductwork in the room.
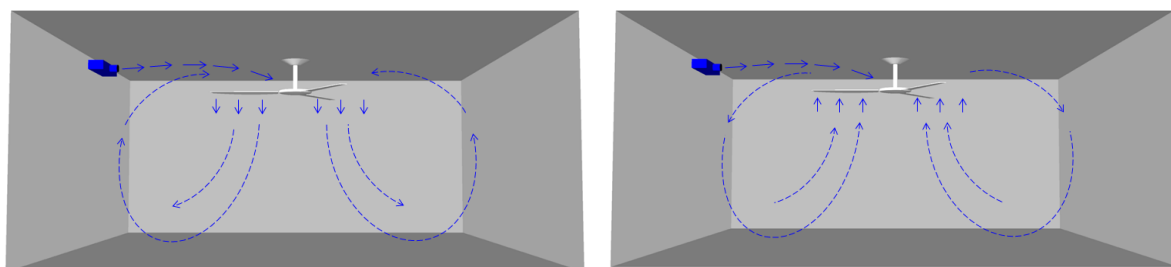
| Fan blowing downward | Fan blowing upward |

Fig. 1. Example of CFIAC system, with supply air jet originating from vent at upper left.

The physical concept is easy to grasp, and some of it may already be encountered in residential spaces containing both split systems and ceiling fans. However, the full scope of CFIAC applied to larger office/institutional/retail spaces involves multiple changes in current heating, ventilation, and air-conditioning (HVAC) design and operation, and these introduce many questions that have not been examined in the past.

1.2 Maintaining comfort in conventional HVAC design

Almost all air-conditioned buildings are currently designed and operated to produce temperatures that are comfortable in still air, by supplying heated or cooled air. In still air scenarios where the room temperature is comfortable, air movement may be perceived as undesirable draft [1]. The mechanical system is therefore expected to minimize air movement within the occupied part of a room, so that the still-air condition is maintained. This requirement is less problematic for heating versus cooling systems, which we will discuss first.

Removing heat loads with cooled supply air presents challenges for designers, operators, and fabricators of air distribution equipment. The relatively dense cooled supply air tends to fall ('dump') into the occupied space from inlets positioned in the ceiling or high on walls, exposing the occupant to cool currents acting locally on the body [1, 2]. To counteract this, supply air is forcibly mixed through diffusers in order to entrain room air, raising the supply jet temperature to reduce its tendency to drop into the room. Diffusers need to be spaced close together to provide uniform coverage of entrained air across the floor area below, requiring lengths of terminal ductwork and substantial numbers of diffusers. In addition, operators have traditionally maintained high volumes of supply air to assure its adhesion to the ceiling via the Coanda effect, a practice that limits their available airflow throttling range and their ability to control the temperature in the space. Elevated flow minima have been demonstrated to be a major cause of the widespread overcooling discomfort occurring in buildings [3, 4].

This cool-temperature-still-air approach to cooling is also energy-intensive. First, the cool temperature setpoint in the space requires higher supply air volumes, which, passing through the sequential pressure drops of terminal ductwork and diffuser grilles imposes a fan energy penalty [5, 6] as well as increasing the required size and first cost of the HVAC system fan and ductwork. In addition, the inherently cooler temperatures of a still-air comfort zone impose sensible and latent cooling loads on the HVAC system, larger than would be imposed by the warmer temperatures of comfort zones in which there is air movement [7, 8].

It is worth considering whether the energy-intensive still-air approach to cooling occupants is in fact superior to other approaches. What if warmer indoor temperatures, together with some air movement, provided equal or better comfort than the cool temperature/still air model?

1.3 Comfort zones and air quality effects under elevated air movement

Extensive field studies in office buildings have shown that occupants prefer to have more air movement than what they are experiencing [9, 10]. Majorities of surveyed occupants in typical air-conditioned environments state that they would prefer higher air speeds at temperatures in which they are 'warm', 'slightly warm', 'neutral', and even 'slightly cool' [11-13]. Their self-reported productivity improves with air speed [14], and their perceived air quality is significantly enhanced, due most probably to disruption of the body's thermal plume [15]. These field study results have led to new provisions in ASHRAE Standard 55 that define comfortable conditions for elevated air movement within building interiors [7]. The air movement offsets the need for air cooling by increasing the convective cooling of the occupants' bodies. It has also been found that a modest air speed significantly reduces a CO_2 bubble that accumulates in a person's breathing zone under still air conditions in workstations [16]. Finally, higher air temperatures and air speeds have also recently been shown to be far more effective (compared to cooler temperatures and still-air) at restoring comfort to occupants who have just entered a space with elevated metabolic rate and stored body heat from walking [17, 18]. Note that in each of these above cases the desirable air flows are isothermal, involving only the room air itself.

Elevated-air-movement can compensate as much as 4 °C temperature difference to provide comfort (depending on the air speed level, see Fig 5.3.3a in Std 55-2017) [7]. Assuming there

is an efficient source of air movement, there is no need for air conditioning within this temperature range; the occupant can be cooled by air movement alone. An HVAC system that integrates air movement and warmer interior temperatures would operate at lower cooling intensity than the still-air system, and also operate for fewer hours in the year.

Room fans, especially ceiling fans, have been used for many years to generate comfort cooling in residential and commercial buildings [19-21]. The human skin surface is relatively warm and moist, and increasing the convection across it provides both sensible and evaporative cooling [22]. A ceiling fan producing a speed between 0.5 m/s and 1.0 m/s in a room's occupied zone compensates for 2 – 3 °C temperature increase in the room [12, 23], and speeds above 1 m/s contribute up to 4 °C. Assorted laboratory studies have confirmed this cooling effect (reference [22] reviews classic ceiling fan studies [24, 25]), and these values have also been consistently observed in field studies of building types such as office, education, fitness centers, factories [12, 14, 26-30].

Ceiling fan cooling to maintain comfort reaches a temperature limit somewhere between 28 and 32 °C [12], depending on environmental humidity, clothing, activity levels; and on practical limits that may constrain air movement: (disturbing paper (~0.8 m/s) or hair or clothing (~1.2 to 1.8 m/s)). Air conditioning is needed to maintain comfort above these air speed limits.

1.4 Energy implications of fan cooling vs air cooling
Fan cooling of occupants is highly energy-efficient compared to cooling the room air to cool the occupants. Modern ceiling fans by themselves use negligible power. For example, a 1.5 m fan, operating at medium speed, can silently move 2 m³/s of air (a fan air speed of 1.2 m/s, see [31]) using just 6 W, equivalent to an LED light bulb. Their cooling per occupant is two orders of magnitude more efficient than using AC to produce the equivalent comfort effect [32].

Although AC also has roles in outside-air ventilation and in dehumidification that ceiling fans do not address, these functions are best treated separately from AC's sensible space cooling role, and will not be discussed here.

Energy analyses have been done for systems that combine fans and AC. The raised thermostat cooling setpoint made possible by the fans provides the primary energy savings. There are system efficiency gains from reduced AC hours, downsized AC equipment, and from the reduced dehumidification needed to stay below a given RH setpoint in a warmer building. Also, having a higher cooling setpoint means the zones are warmer during the afternoon/evening and thus, less likely to hit the heating setpoint the next morning. This happens in the swing seasons across almost all climate zones. Energy simulations show a saving of 15% in residential cooling energy by using ceiling fans and a 1.1 °C increase in thermostat setpoint [33]. Field measurement in 400 Florida households found savings between 17% and 48% [34]. A combination of ceiling fans and AC system in a tropical laboratory study provided maximum comfort at 27 °C at 1 m/s air flow compared to 24 °C in still air, achieving a 25.8% reduction in simulated annual energy consumption [35]. In commercial buildings, detailed simulations indicate that ceiling fan air movement can reduce

total HVAC energy consumption by over 20% by enabling a 2 °C increase in cooling setpoint temperature [36, 37].

1.5 Room airflow under ceiling fans

Because their cooling is more energy-efficient, the ceiling fans should always be at their highest acceptable speed before the AC is turned on, and once AC is on, they should remain at that speed so the person is maximally cooled by convection (one might call this 'fan-first, fan-last' operation). The air temperature in a space cooled this way will be warmer than in the current practice of conditioning for still indoor air.

In rooms with ceiling fans, temperatures in the occupied zone tend to be almost isothermal but the airflow may be highly non-uniform. Depending on the occupancy and building type, such non-uniformity might need to be filled in by personal fans.

A review paper [38] and a field study of a fan-cooled office [39] have demonstrated successful uses of ceiling fans. Office workplace cooling is particularly demanding because people may not be at liberty to move their location, and the fans may be only under group, rather than individual, control.

There have been numerous laboratory studies of velocity distributions in rooms with ceiling fans, including many recently [21, 31, 40]. Paper [31] included fans being operated in both downward and upward flow modes. Models for such isothermal room flow distribution under ceiling fans have been proposed for both empty rooms and for rooms with furniture and partitions [31, 40].

The isothermal nature of pure fan cooling will be affected once the air conditioner comes on, since the introduced cooler supply air will interact with the moving air from the fans, and might produce various levels of temperature difference within the space. Ceiling fan airflow can be expected to entrain, diffuse, and redirect the cool supply air across the room, but how much? There is a need to evaluate the extent of this mixing, and its effects on comfort.

1.6 Need for evaluation metrics

For conventional HVAC systems, the *Air Diffusion Performance Index* (ADPI) is used for evaluating room airflow and temperature distributions. It is used by HVAC diffuser manufacturers to rate their products, whose goal is to create, under a range of heat gains, uniform temperatures (low thermal stratification) and still-air in the room (low draft risk in the cool temperatures) [2]. ADPI has a low maximum allowable velocity (0.2 m/s) that is well below the range of desirable velocities for CFIAC. The ADPI Index cannot be directly converted to apply to warmer temperatures and higher air speeds.

There is also no current index for evaluating the distribution of occupant *comfort* in a space cooled by ceiling fans, with or without the presence of cooled AC air. The index 'predicted percent dissatisfied' (PPD) based on the underlying 'predicted mean vote' (PMV) model is ineffective at predicting convective cooling because it does not realistically account for the evaporation component of convection from the skin; it underpredicts fan cooling by 50% [28]), and has been found inaccurate in field studies [41]. However, cooling under elevated

air speeds has since 2009 been predicted in ASHRAE Standard 55 [7] by the SET model, which has been experimentally verified [28] across a range of conditions. For the combination of conditions created by CFIAC, the elevated air speed method in ASHRAE Standard 55 may provide a comfort index.

1.7 Physical characteristics of proposed integrated system

Putting the above considerations together, the CFIAC system directly mixes conditioned supply air emerging into the room from vents, with supply-air volume and ceiling-fan speeds coordinated to work across a range of indoor- and supply-air temperatures. The vents may be positioned in various locations in the room within range of the fans, such as high on the walls, or in the ceiling. The vents might be part of a central air system, or be the outlets of sidewall- or ceiling-mounted split-system air conditioning units. For most complete mixing, the cool supply jet would be directed into the vicinity of the ceiling fan's upstream inhalation zone, but it might also be directed into the jet produced by the fan downstream of the propeller blades.

The physical layout of such a system would preferably minimize the amount of terminal ductwork and the number of diffusers found in the typical interior space. This would free up the ceiling from suspended ductwork and substitute less-visually-intrusive ceiling fans to perform the diffusing function. This will also likely reduce air-side pressure drop losses in fan power, and reduce the first costs of sheet metal and its installation, both offsetting the cost of the ceiling fans. It may also provide aesthetic benefits of revealing ceilings with less ductwork clutter, *and*--particularly significant but often overlooked--provide more illumination of the ceiling from window daylight that is currently being intercepted by exposed overhead ducts.

We will very briefly discuss CFIAC in heating mode. The physical configuration of ducts and fans remains the same, but in this mode the air movement in the occupied zone has to be kept low. Ceiling fans have long been operated in both upward and downward directions to destratify warm room air in winter, and guidelines for this exist [2]. The supply of heated air into the room's upper regions via shortened CFIAC ductwork will create a stratified condition that is very similar to what happens naturally. The CFIAC ceiling fans will follow conventional practice in destratifying and mixing this heated air by blowing it at low speeds either upward or downward. An interesting economic aspect of CFIAC destratification is that it may eliminate the need in current HVAC heating practice for extra heating sources placed low in the room, such as under windows. It also affects room ventilation effectiveness as prescribed in ASHRAE Standard 62.1 ventilation. The standard assumes a ventilation effectiveness of 0.8 for ceiling supply and return systems in heating mode, which has the effect of requiring 25% higher outside air flow rates during heating mode. Ceiling fan mixing would remove this constraint, allowing higher ventilation effectiveness (1.0) and reduced outside air.

1.8 Needs for CFIAC-specific research

To move this approach into actual widespread practice, new information will be needed at several levels. We foresee the following types of research and development, first in the lab and ultimately in the field:

1) Flow patterns of cold supply air interacting with downward and upward ceiling fan circulation within the room, and the resulting distribution of temperatures and velocities in the space.
2) Human subject tests of comfort under the same conditions.
3) Ventilation effectiveness in rooms with CFIAC.
4) Performance indices for evaluation and design of rooms with CFIAC.
5) Control sequences for optimal operation throughout the range of seasons.
6) Evaluation of first costs, e.g., costs of ceiling fans and savings from reduced terminal ductwork.
7) Evaluation of operational energy savings, and also savings from reducing AC equipment size.
8) Evaluation of energy and comfort of installed systems under long-term operation.
9) Design tools for room layout and system sizing.

This paper addresses 1) and part of 4) above. To the authors' knowledge, velocity and temperature distributions from ceiling fans operating together with HVAC in cooling mode have been rarely reported. We know only of two field studies, one of temperature stratification and comfort in a controlled classroom using ceiling fans and a ceiling-mounted package AC unit [42], and another that measured temperature and speed in spaces operating with fans and central AC via overhead diffusers [21]. A simulation [43] of a ceiling fan interacting in a small room with high-sidewall supply and a low outlet, and with a standing occupant directly under the fan, produced temperature and velocity fields much more extreme than those in the measurement studies, and not validated with experimental data.

This study was performed together with two others using the same experimental configuration: a human subject study of comfort throughout the room under CFIAC conditions addressed the need 2) above, and a study of the CFIAC system's ventilation effectiveness throughout the room using tracer gas addressed research need 3). These will be reported in two subsequent papers. The ultimate intention of the series is to provide preliminary guidelines for CFIAC control strategies and physical designs.

1.9 Objective
The remaining objective of this paper is to characterize the environments created by an archetypal and aerodynamically interesting CFIAC system in cooling mode with supply air coming from a high-sidewall vent. In order to provide generalizable knowledge about the system's flow characteristics, measurements need to consider four factors: (1) ceiling fan speed, (2) ceiling fan airflow direction, (3) supply air volume, and (4) room-supply air temperature difference. The measured results must characterize velocity patterns (mean and turbulence intensity), temperature uniformity and stratification. These will be evaluated with velocity and temperature uniformity indices, and the distribution of computed comfort levels based on ASHRAE Standard 55.

2. Methods
2.1 Experimental setup
2.1.1 Chamber setup: The experiment was conducted at the Center for the Built Environment (CBE) at University of California, Berkeley. The dimensions of the office-style

environmental chamber are 5.5 (X) m × 5.5 (Y) m × 2.53 (Z) m (Fig. 2a). A ceiling fan (Haiku 60, Big Ass Fans, Inc.) [44] of 1.5 m diameter (D) was installed near the center of the room, (described below), and 0.37 m below the ceiling. A supply vent, (0.184 × 0.133 m (equivalent diameter d = 0.155 m), was mounted high on one wall, approximately midway along its length. The distance between the center of the vent and the ceiling was 0.38 m (2.15 m from the floor), and the vent face was mounted proud of the wall in a 0.36 (X) m × 0.33 (Y) m × 0.36 (Z) m small box. The supply vent register (Price Industries 520 Grille), has adjustable airfoil vanes allowing the supply air throw direction to be adjusted vertically. The exhaust grille has a size of 0.61 m × 0.61 m, the distance between the center of the exhaust and the nearest wall is 0.9 m. We spaced 14 heating panels (170 watt each, total 2380 watt ~ 80 W/m²) around the chamber walls 0.25 m away from the wall, in order to simulate office internal loads without creating rising plumes within the measured space. Each heating panel is 0.83 m long, 0.61 m wide and 0.03 m thick, and covered with aluminum foil to reduce radiation asymmetry within the room. The chamber's exterior walls and 3-layer windows have an air gap behind their inner surfaces, through which conditioned air circulates to maintain the chamber's walls at equal surface temperatures.

Most of the measurements were performed for a condition in which the ceiling fan was in the near-center of the ceiling, in line with the supply and exhaust openings. See the black fan in Fig. 2a and 3b). However, for situations where the supply air jet might not be directed toward a ceiling fan, we repeated the tests with the fan moved 1.2 m off centerline into a far corner of the room (see the grey fan in Fig. 3b). In this situation the supply jet enters the ceiling fan circulation indirectly.
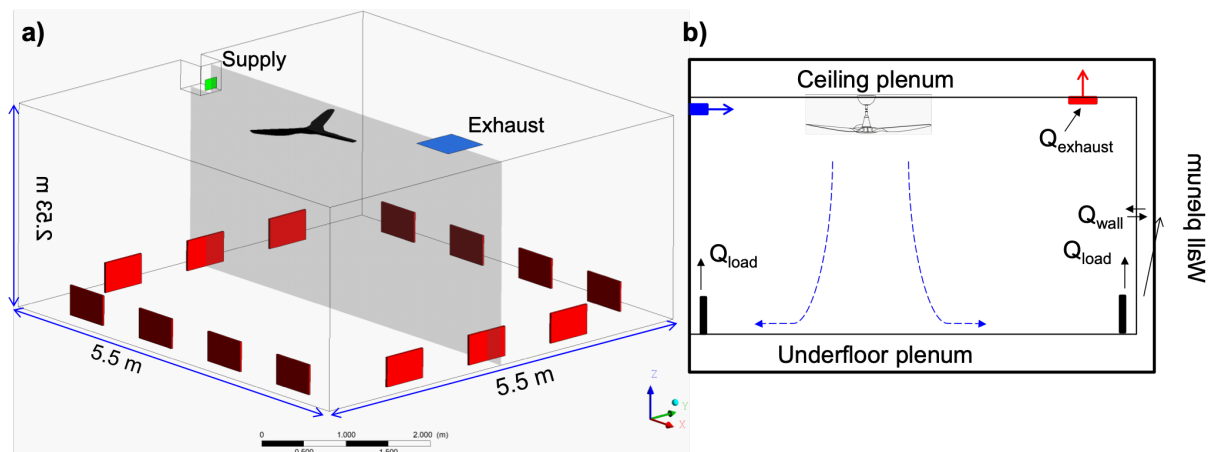

Fig. 2. Schematic diagram of experimental setup, a) layout of the room, b) longitudinal section.
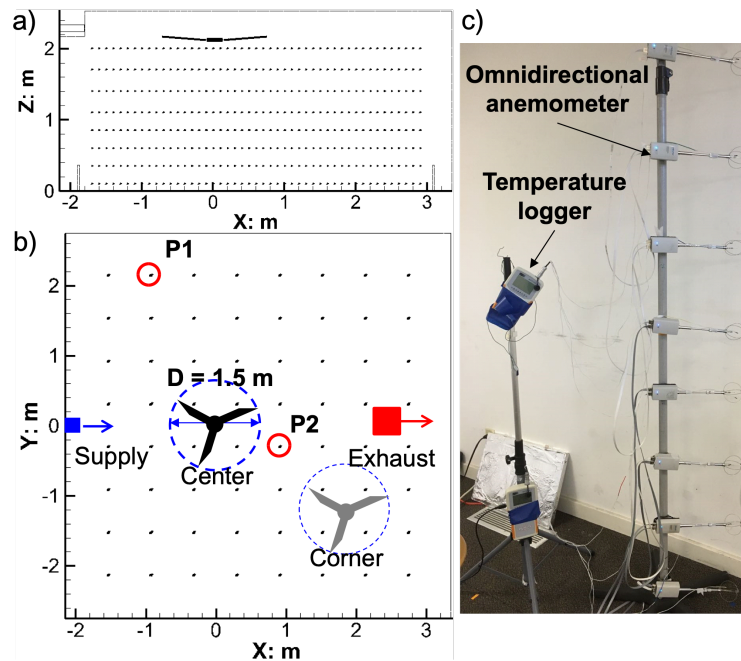
Fig. 3. Measurement points of air speed and temperature in: a) detailed measurements along the centerline of the vent and the fan, b) cross-room measurement points, and c) equipment setup.

2.1.2 Measurement points: Measurements were taken under steady-state conditions, with air velocity and temperature measured simultaneously. There were two types of measurement density. Detailed measurements were made in a vertical plane across the room, through the centerline of the supply vent and the fan (Fig. 3a), to the opposite wall. Readings were taken every 10 cm by an anemometer/thermister tree, with sensors at eight heights (0.1, 0.35, 0.6, 0.85, 1.1, 1.4, 1.7 and 2.0 m). This detailed set of measurements show the interactions of the air from the vent and the fan. The same eight-height measurements were then made across the entire room floor at 2 ft (0.61 m) intervals, shown in Fig. 3b. These lower-density measurements evaluate the velocity and temperature profiles of the entire room. The two points P1 and P2 are chosen to evaluate the air temperature stratifications within the room. P1 is farther away from the ceiling fan, and P2 is near both the ceiling fan and the supply air vent. The air flow was measured by omnidirectional anemometers with a sampling frequency of 0.5 Hz. The anemometer system (Sensor Inc., Gliwice Poland) is designed for the typically low air velocities of room airflow, with an accuracy of ±0.02 m/s or 1% of reading (0.05 - 5 m/s), and a time constant below 0.3 s. It has been shown to measure turbulence frequencies up to and slightly exceeding 1 Hz [45]. It should be noted that 'velocity' as used in this paper is always the measured air speed, a scalar value. Vector directions were obtained from smoke visualization (PT-1500, DJPOWER, Guangzhou). Temperature loggers (WZYCH4; Tianjianhuayi Inc., Beijing) were used for temperature measurement with a precision of ±0.2 °C. The measured duration was 3 and 1 min for each point in the XZ and XY planes, respectively. A picture of the test room and the equipment is shown in Fig. 3c.

2.1.3 Fan operations: The fan was operated at three rotational levels, low (speed-setting level 2, fan rotational speed: 72 rpm, fan airflow: 1.2 m³/s, fan air speed: 0.6 m/s), medium (level 4, 122 rpm, 2.2 m³/s, 1.2 m/s), and high (level 6, 177 rpm, 3.3 m³/s, 1.8 m/s) [31] (The

highest available fan level is 7). Fan air speeds are averaged through the area swept by the fan blades, the airflow rate of downward and upward fan was measured by omnidirectional probes in a planes below or above the fan, and with a 0.15 m distance to the blades [31]. In order to examine the air mixing effect of the upward-blowing fan, we used the level 4 reverse-rotation speed, the highest upward speed allowed by the fan due to UL 507 blade tip thickness and speed constraints. Table 1 summarizes the corresponding airflow rates of each fan speed.

Table 1 Fan air flowrate under each rotation speed (downward flows are EnergyStar fan test data from Big Ass Fans, Inc.; upward flow measured on-site).

Level	2 Downward (L2)	4 Downward (L4)	6 Downward (L6)	4 Upward (L4 up)
Airflow rate (m³/s)	1.2	2.2	3.3	1.3

Table 2 Detailed measurement conditions.

Series	T_{supply} (°C)	Supply angle	Flowrate (m³/s)	Flowrate	V_{supply} (m/s)	Fan place	RPM (rpm)	Measurement Planes
AL0	10	0	0.056	35%	2.27	Center	0	XZ, XYs
AL2	10	0	0.056	35%	2.27	Center	72	XZ, XYs
AL4	10	0	0.056	35%	2.27	Center	124	XZ, XYs
AL4up	10	0	0.056	35%	2.27	Center	124 (upward)	XZ, XYs
BL0	10	30° up	0.056	35%	2.27	Center	0	XZ, XYs
BL2	10	30° up	0.056	35%	2.27	Center	72	XZ, XYs
BL4	10	30° up	0.056	35%	2.27	Center	124	XZ, XYs
BL4up	10	30° up	0.056	35%	2.27	Center	124 (upward)	XZ, XYs
CL0	14	0	0.079	50%	3.25	Center	0	XZ, XYs
CL2	14	0	0.079	50%	3.25	Center	72	XZ, XYs
CL4	14	0	0.079	50%	3.25	Center	124	XZ, XYs
CL4up	14	0	0.079	50%	3.25	Center	124 (upward)	XYs
DL0	14	0	0.114	70%	4.66	Center	0	XZ, XYs
DL2	14	0	0.114	70%	4.66	Center	72	XZ, XYs
DL4	14	0	0.114	70%	4.66	Center	124	XZ, XYs
DL4up	14	0	0.114	70%	4.66	Center	124 (upward)	XYs
DL6	14	0	0.114	70%	4.66	Center	184	XZ, XYs
EL0	17	0	0.163	100%	6.66	Center	0	XZ, XYs

EL2	17	0	0.163	100%	6.66	Center	72	XZ, XYs
EL4	17	0	0.163	100%	6.66	Center	124	XZ, XYs
EL4up	17	0	0.163	100%	6.66	Center	124 (upward)	XYs
EL6	17	0	0.163	100%	6.66	Center	184	XZ, XYs
FL0	10	0	0.056	35%	2.27	Corner	0	XYs
FL2	10	0	0.056	35%	2.27	Corner	72	XYs
FL4	10	0	0.056	35%	2.27	Corner	124	XYs
FL4up	10	0	0.056	35%	2.27	Corner	124 (upward)	XYs

2.1.4 Test conditions: Table 2 describes the detailed measurement conditions. A combination of three supply air temperatures (10, 14, and 17 °C, common supply air temperatures in practice) and four supply flowrates (0.056, 0.079, 0.114, 0.163 m³/s, corresponding to 35%, 50%, 70% and 100% of the maximum flowrate) were measured. The design temperature of indoor air for Cases A, B, C, E and F is 26 - 26.5 °C, the design temperature of indoor air for Case D is 24 °C. Higher supply temperatures were tested together with higher supply flowrates so that the cooling rates would remain similar to those from lower supply temperature and flowrates. The low supply temperature (10 °C) and flowrate were tested because they present the most severe challenge to avoiding supply air dumping into the occupied zone. Series A through E are for the center ceiling fan location. One condition (Series B) was tested with vent louver blades tilted upward 30-degrees, done to engage the Coanda effect along the ceiling and counteract the dumping tendency. This also sends supply air entirely into the feed zone above the (downward-directed) fan; whereas the flow from the level vent sends some of the supply air perpendicularly into the fan jet. For Series D and E (70% and 100% of the maximum supply flowrate), because the supply air velocity is high, we added a Level 6 ceiling fan air speed to balance this.

The Series F is for the corner fan location. The purpose of the corner fan location is to evaluate mixing results by ceiling fans when the supply air is not directly sent to the ceiling fan. In our arrangement, the distances are 1.2 × 4 m away (see Fig. 3b), so none of the supply air from the vent travels directly into the ceiling fan. We tested this fan location together with the lowest supply flowrate and temperature, in which the mixing by the fan would be least effective. The fan ran at Level 0 (off, reference condition), Levels 2 and 4 in the downward direction, and Level 4 in the upward direction.

2.2. Indoor thermal environment evaluation indexes

2.2.1 Supply air jet trajectories: Without the ceiling fan, the airflow from the supply vent behaves as a negatively-buoyant air jet [46]. When the jet penetrates more than half the chamber length, it creates a single circulation airflow pattern in the centerline cross-section of the room. If the jet drops sooner, then two circulation zones are formed. The air velocity from the jet and the temperature difference between the jet and the room air determine the airflow trajectories, following the Archimedes number (Ar), Eq. (1) [46]:

$$Ar = \beta\, g \Delta T_0 d / (u_0 \tan\alpha)^2 \qquad (1)$$

Where: β is the coefficient of expansion of the air, equal to $2/(T_f+T_0+546)$, K^{-1}, α is the supply angle, g is the acceleration due to gravity, m/s^2, T_0 is inlet air temperature, °C, T_f is room averaged temperature, °C, ΔT_0 is the temperature difference between the inlet and the average room temperature, °C, d is the equivalent diameter of diffuser, m, u_0 is the supply air velocity, m/s (see Table 2). When Ar < 0.005, in which either the air temperature difference between the jet and the room is small or the velocity from the jet is large, a single-circulation airflow pattern is formed, and the airflow can be treated as isothermal. When Ar > 0.015, the airflow pattern is governed by buoyancy forces and the air-jet falls on entry. When Ar is between 0.005 and 0.015, an unstable dual-circulation airflow pattern is established [46].

2.2.2 Non-uniformity: Velocity and temperature non-uniformity coefficients (K_v and K_t) [47] are used to evaluate the spatial variability velocity and temperature produced by the system (Eq.s (2) and (3)). The smaller the value, the more uniform the distribution of the variable in the room.

$$K_V = \sqrt{\sum_{i=1}^{n}(V_i - \overline{V})^2 / (n-1)} \Big/ \overline{V} \qquad (2)$$

$$K_T = \sqrt{\sum_{i=1}^{n}(T_i - \overline{T})^2 / (n-1)} \Big/ \overline{T} \qquad (3)$$

where, V_i and T_i are mean air speed and temperature measured for each point, respectively. $\overline{V}$ and $\overline{T}$ are the mean air speed and temperature of all measured points across the room in the XY plane at all heights.

2.2.3 Similarity: To quantify the similarity or differences between velocity profiles and a reference condition, the normalized root-mean-square deviation (NRMSD) [48] is commonly used.

$$NRMSD = \sqrt{(\sum_{j=1}^{n}(\overline{V}_{M,i} - \overline{V}_{N,i})^2)/n} \Big/ (\overline{V}_{M,\max} - (\overline{V}_{M,\min} + \overline{V}_{N,\min})/2) \qquad (4)$$

Where i represents the height and M is a reference case. In our study, we use Series A for the reference conditions under fan levels 2 and 4, because the supply flowrate from the vent is smallest and the velocity profiles are close to those of a single ceiling fan. N represents different series under the same fan speed. $\overline{V}_{\max}$ and $\overline{V}_{\min}$ come from the same height. NRMSD represents normalized velocity variations compared to the reference case (Series A) under fan levels 2 or 4. The larger the value, the greater the difference from the reference condition.

2.2.4 Turbulence intensity (TI): TI indicates the intensity of airflow fluctuations, with

higher values representing more fluctuation. TI over a period of time is defined as the velocity standard deviation divided by mean velocity:

$$TI = \sqrt{\sum_{j=1}^{n} (V_{ij} - \overline{V}_i)^2 / n} \Big/ \overline{V}_i \qquad (5)$$

where, V_{ij} is the instantaneous velocity at point i, $\overline{V}_i$ is the mean velocity measured at point i. Humans perceive turbulent air cooling in the frequency range of 1.5 to 0.2 Hz, with significant cooling effects occurring only between 1.0 and 0.5 Hz [49-54]. For measuring this range, an anemometer time constant of 0.3 s and a sampling rate of 0.5 Hz over 3 minutes are adequate. TI has a small positive effect on cooling by increasing skin heat loss and the perception of cooling [52]. It should be noted that the ISO draft risk model [55] is inappropriate for predicting discomfort in neutral- to warm conditions as in CFIAC.

3. Results

3.1 Vertical distributions of velocity and temperature with CFIAC
3.1.1 Velocities in the XZ (vertical) plane
Fig. 4 presents the velocity fields of all series when the fan is in the XZ plane based on detailed measurement points shown in Fig. 3a. Each column represents a Series at one supply flowrate and temperature. In total, there are 5 columns representing for Series A, B, C, D, and E (see Table 2).

3.1.1.1 Velocity distributions without fan running: The Ar values for Series AL0, BL0, CL0, DL0 and EL0 ("L0" means "fan speed level zero", or "ceiling fan not running") are 0.0165, 0.022, 0.006, 0.0025 and 0.0009, respectively (The room average temperature T_f for case AL0, BL0, CL0, DL0 and EL0 is 26.12, 26.34, 26.04, 24.10 and 26.28 °C, respectively). That means for AL0 and CL0, the air from the supply jet falls after it emerges into the room. For DL0 and EL0, since the Ar is smaller than 0.005, the airflow forms a single-circulation airflow pattern [46].

Figures in the top row are for conditions without the fan running. It can be seen that sectional views of DL0 and EL0 (top two figures in column D: $T_{supply} = 17$ °C, 70% maximum supply flowrate, and column E: $T_{supply} = 17$ °C, 100% maximum supply flowrate) illustrate a similar pattern. The supply air flow is attached to the ceiling due to Coanda effect [56, 57]. The airflow is in a single-circulation pattern because Ar < 0.005 [46].

In AL0 (Ar = 0.0165), the supply air falls downward as a free jet after emerging from the vent, since air flows with Ar > 0.015 are governed by negative buoyancy forces. However, in BL0, the free supply air jet is directed upward by the vent louvers angled 30° above horizontal, so that it engages the Coanda effect when it reaches the ceiling. This delays its descent. The Ar for Series CL0 is 0.006, slightly greater than 0.005, therefore, the airflow is close to being a single-circulation pattern.

3.1.1.2 Velocity distributions with fans (CFIAC):

As depicted in Fig. 4 (L2, L4, L6, rows 2 – 4), with a downward-running ceiling fan, the airflow patterns of CFIAC are transferred into the descending ring jet below the fan blades. The air discharged from the supply vent is inhaled into the ceiling fan, and is then spread downward to the floor.
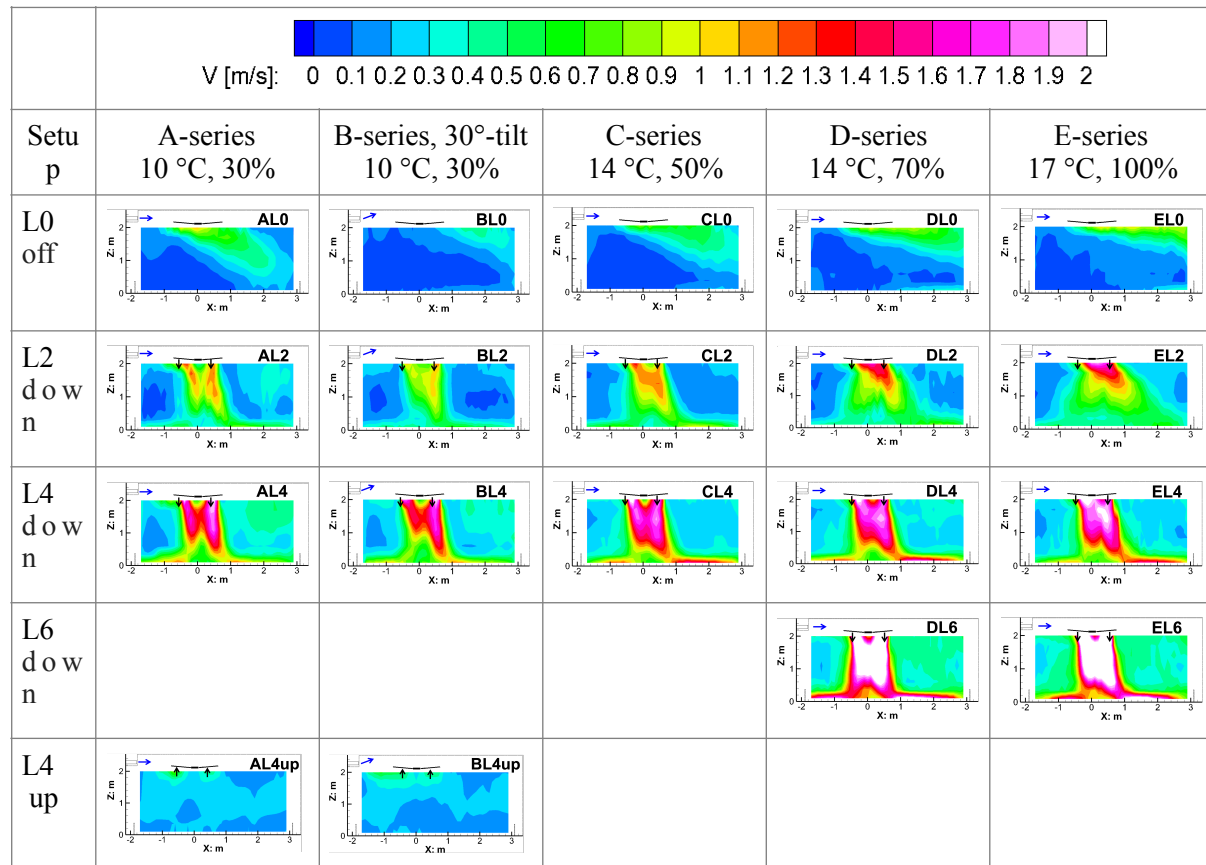

Fig. 4. Velocity distribution of all Series in XZ plane. Test conditions are described in Table 2. Fan air speed: 0.6 m/s (L2), 1.2 m/s (L4), 1.8 m/s (L6).

Please note that the data above 2 m height are not measured. The data ends 35 cm from the two walls on the left and right sides. The room airflow profiles are determined by the interaction of airflows from the vent and the ceiling fan, and in general, they are dominated by the ceiling fan airflows, whether at levels 2, 4, or 6. They are similar to the profile of a fan running alone even at the lowest levels [40, 44]. But looking more closely, at higher supply air flows the room airflow profiles are shifted slightly towards the right side by the momentum of the air from the vent, especially for the low level 2 fan speed and higher supply flowrates from the vent (70% and 100% of the maximum flowrate, DL2, EL2).

The upward-tilted louvers delay the descent of supply air and improve its mixing when the fan is not running (AL0 and BL0). However when the fan is running, the airflow pattern of AL2 and BL2 (fan level 2), or AL4 and BL4 (fan level 4), or AL4 up and BL4 up (fan upward direction), are all almost identical, indicating that the upward-tilt supply air angle (and the resulting Coanda effect) have little effect on the airflow of CFIAC when the ceiling fan is running, even at its low level.

3.1.2 Normalized velocity profiles. Figures 5 and 6 present dimensionless velocity profiles of the different Series, at four heights (0.1 m, 0.6 m, 1.1 m and 1.7 m) and Levels 2 and 4 in the downward fan direction (Fig. 5 covers fan level 2 and Fig. 6 covers fan level 4). We divided by the fan air speed at each fan level, and normalized the radial distance (X) from the fan center by dividing by the fan radius (R). The results are consistent with what we have seen above in Figure 4. When the fan level is low (level 2, Fig. 5) and when the supply flowrate from the vent is high (70% and 100% of the maximum, DL2 and EL2), we see the jet laterally displaced with higher velocities on the right side than on the left side (black and orange lines). At lower supply flowrates (AL2, BL2 and CL2, 35% and 50% of maximum flowrates), the profiles are almost identical, except some small differences under the ceiling fan (-1 < X/R <1). When the fan level is high (level 4, see Fig. 6), the differences from the different flowrates are also very small, happening only under the ceiling fan itself (-1 < X/R <1).

In general, the almost overlapped normalized velocity profiles show that velocity self-similarity exists for different fan levels and different supply flowrates, except for the low fan level combined with highest supply flowrate.
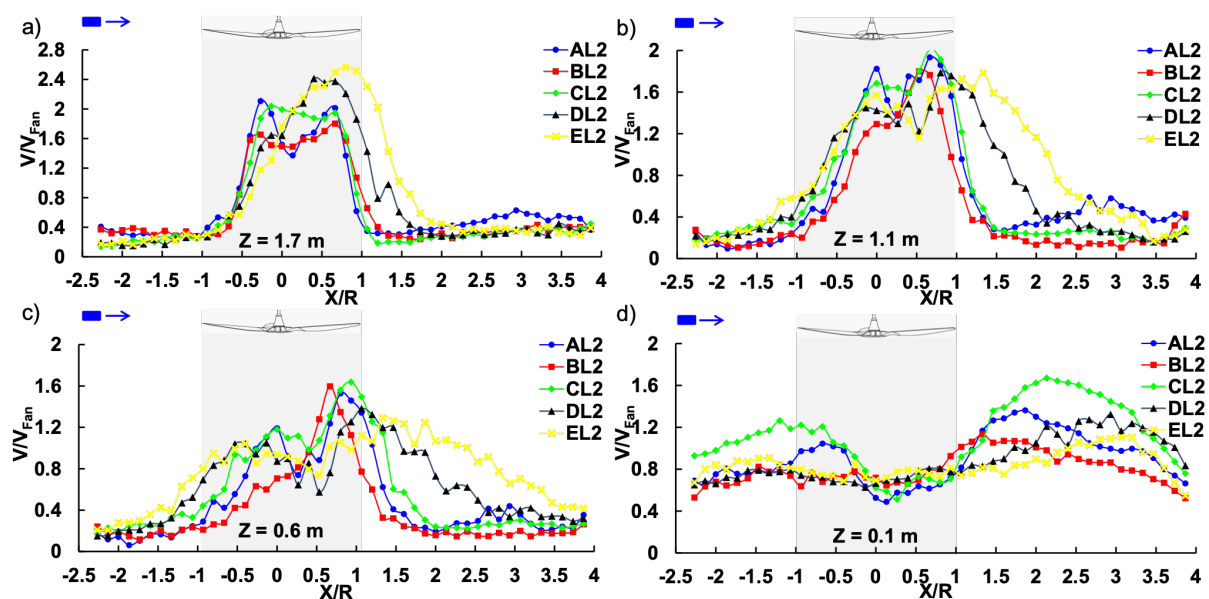

Fig. 5. Normalized velocity profiles of CFIAC for fan air speed = 0.6 m/s in XZ plane: a) Z = 1.7 m, b) Z = 1.1 m, c) Z = 0.6 m, d) Z = 0.1 m.
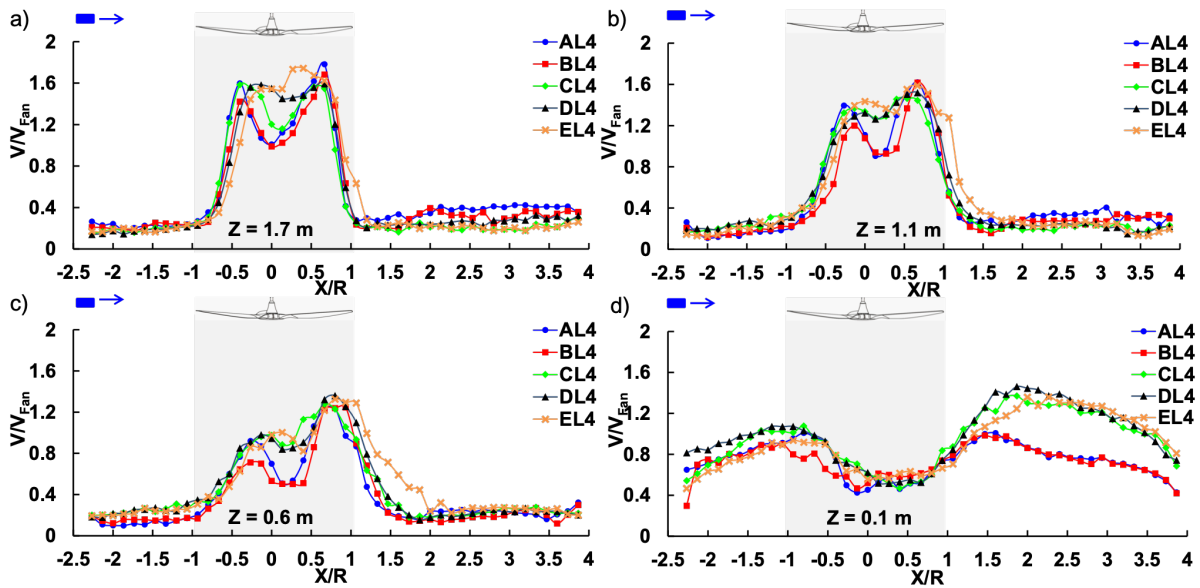
Fig. 6. Normalized velocity profiles of CFIAC for fan air speed = 1.2 m/s in XZ plane: a) Z = 1.7 m, b) Z = 1.1 m, c) Z = 0.6 m, d) Z = 0.1 m.

3.1.3 Similarity evaluation. The calculated NRMSDs for the different flowrates for the two fan levels (L2 and L4) are summarized in Table 3, taking AL2 and AL4 as the reference conditions. They increase with the increased supply air flowrate from the vent (Series A to E), and reduce with fan speed increase (L2 and L4). For fan level 2, the biggest difference is observed at the 0.6 m height, before reaching the floor. For fan level 4, the biggest difference is observed when the air reaches the floor (0.1 m height).

The NRMSDs values for low flowrate from the vent (BL2 to CL2) are lower than 15%, (ranging from 8.6% to 14.6%), indicating that the differences from the reference condition (AL2) are small. Velocity self-similarity with the reference condition is achieved with supply air flowrate $<=$ 0.079 m^3/s (50% of maximum flowrate, 3.8 ACH). It is only at fan level 2 together with 70% and 100% supply flowrates (DL2 and EL2) that the NRMSDs are higher than 15%, indicating that they are not represented by the reference condition AL2. Self-similarity exists for AL4 down except at the 0.1 m height when the supply flowrates are high (DL4, EL4).

Table 3 Root-mean-square deviations (NRMSDs) in the occupied zone for fan speed level 2 and 4.

Case (fan-L 2)	AL2*	BL2	CL2	DL2	EL2	Case (fan-L 4)	AL4*	BL4	CL4	DL4	EL4
1.7 m	Ref.	8.6%	9.7%	16.3%	24.2%	1.7 m	Ref.	5.0%	9.5%	11.8%	14.1%
1.1 m	Ref.	11.3%	8.4%	19.6%	28.9%	1.1 m	Ref.	7.8%	9.6%	9.3%	12.6%
0.6 m	Ref.	14.6%	8.6%	24.4%	37.5%	0.6 m	Ref.	7.7%	9.9%	8.3%	14.8%

0.1 m	Ref.	12.5%	9.1%	15.4%	18.8%	0.1 m	Ref.	9.1%	14.7%	15.0%	19.5%

Note: * Reference condition for the calculation of relative difference. AL2* for level 2 and AL4* for level 4

3.1.4 Turbulence intensity (TI) distributions. The TI distributions in the XZ plane of all Series are compared in Fig. 7. The TI is greatest when the ceiling fan is not running (AL0 – EL0), with highest values in the recirculating region near the floor where there are low velocities and relatively large fluctuations. With the fan running, the velocities near the floor are higher due to the spreading outflow from the fan jet (see Fig. 4 - 6), reducing the TI below that of most of the vertical plane. This is similar to the findings of [40]. The lowest TIs happen in the core zone under the ceiling fan where the velocities are highest (see Fig. 4). The TI fields for the downward ceiling fan are similar at different speed levels and different supply air flowrates (except level 2 with 100% supply flowrate, EL2), indicating that TI self-similarity exists at both fan levels with different supply air flowrates from the vent. Compared to the downward fan cases, the TI distributions for ceiling fan upwards (Case AL4up and BL4up) are more uniform within the occupied zone, ranging from 10% to 20%. In either direction, TI values are low whenever the ceiling fan is running.
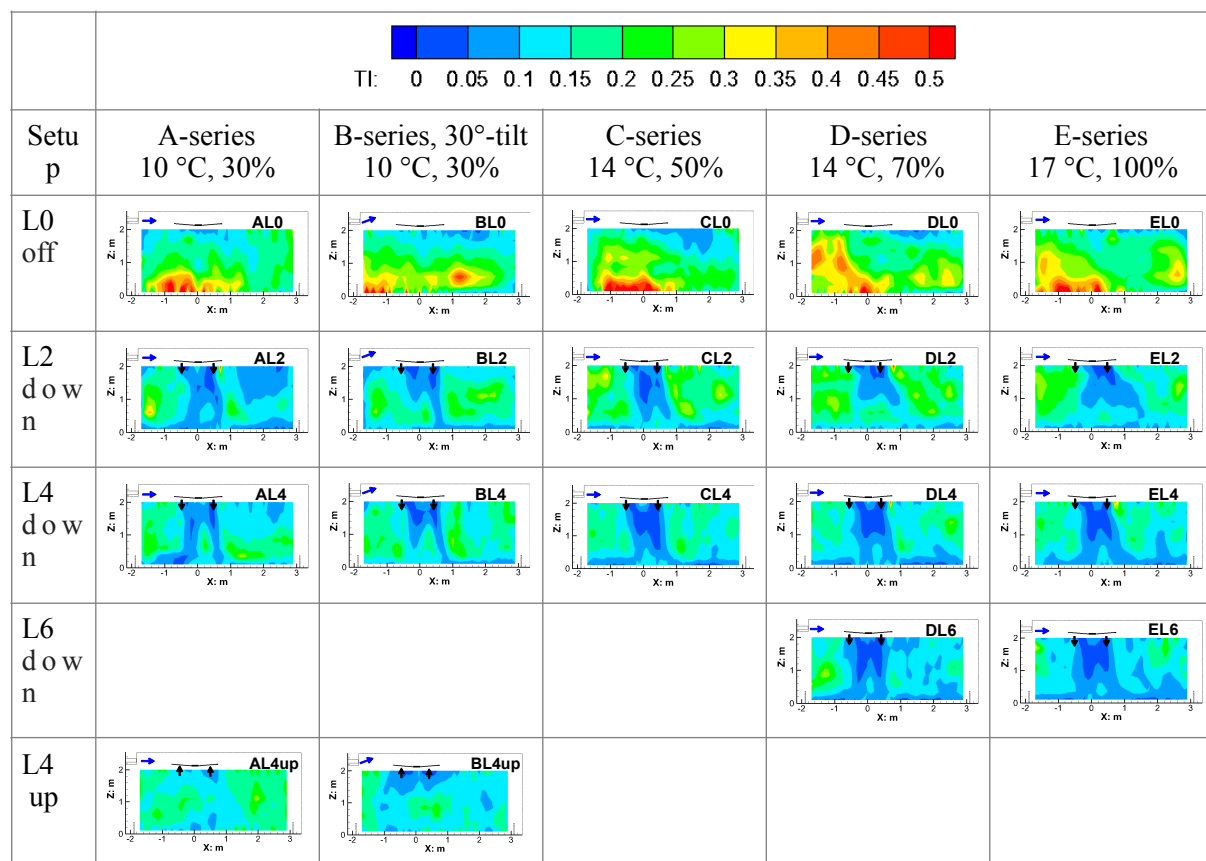

Fig. 7. TI distribution of all Series in XZ plane. Fan air speed: 0.6 m/s (L2), 1.2 m/s (L4), 1.8 m/s (L6).

3.1.5 Temperatures in the vertical plane
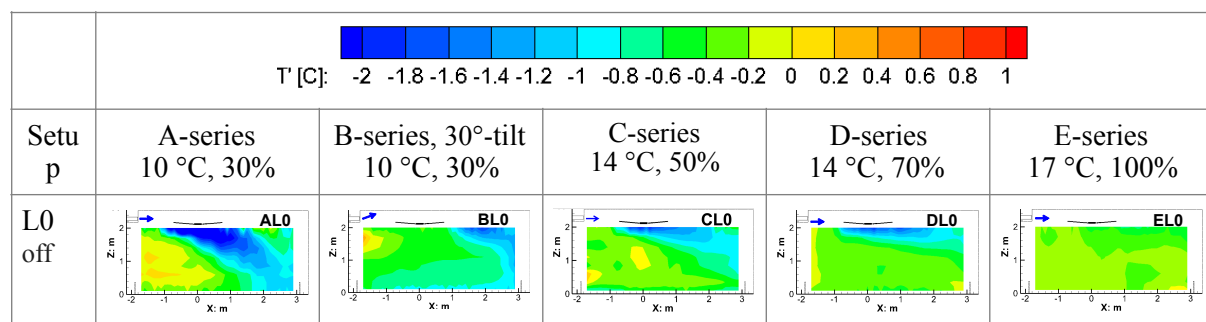*3.1.5.1 Temperature distribution maps*

The distribution of measured temperature variation $T_i{}' = T_i - \overline{T}$ in the XZ plane is shown in Fig. 8 for all Series. T_i is the measured temperature at point i, $\overline{T}$ is the average temperature in the entire chamber at 4 heights (0.1, 0.6, 1.1, 1.7 m; see Fig. 3 and Table 4). Note that $\overline{T}$ is not only the average temperature of the XZ plane. We used the entire chamber to get a reference temperature, because we will compare the temperature distributions in the horizontal plane as well.

For the lowest supply flowrate and temperature (35% AC maximum, 10 °C supply temperature, AL0 with ceiling fan off), there is a large area of low temperature in the occupied zone, about 1.4 K lower than the rest of the area (Fig. 8). The velocity is also high in this area (see Fig. 4), indicating "dumping" of the cold air from the vent into the occupied zone. When the ceiling fan is on, it mixes this temperature significantly and reduces the area of cool air. At fan speed level 2, most of the temperature in the room is within 0.6 °C difference. When the fan is at level 4, the mixing of the ambient temperature is so complete that the cool air zone has disappeared in the occupied zone.

Fig. 8 also shows the nature of the entrainment in the downward ceiling fan jet. Relative to the no-fan situation in Series A, C, D, and E, cooler air appears to discharge from the left side of the ceiling fan. For the B Series with the upward-tilted louvers, this discharge is weaker and appears instead on the right side of the fan. The difference appears to be caused by the supply jet impinging on and merging with the ceiling fan jet below the level of the fan blades (in the A, C, D and E Series), versus the elevated B-series jet being 'inhaled' into the fan from a level above the blades. In general, the temperature differences in the occupied zone are very low, within 0.6 °C, for all modes of flow entrainment into the fan jet.

Table 4 Average room air temperature in different cases.

Setup	Series A	Series B	Series C	Series D	Series E	Series F
L0	26.32	26.42	26.20	24.14	26.38	26.40
L2	26.62	26.48	26.21	24.19	26.48	26.43
L4	26.56	26.55	26.21	24.40	26.46	26.45
L6	-	-	-	24.18	26.43	-
L4up	26.29	26.47	-	-	-	26.45

Setup	A-series 10 °C, 30%	B-series, 30°-tilt 10 °C, 30%	C-series 14 °C, 50%	D-series 14 °C, 70%	E-series 17 °C, 100%
L0 off	AL0	BL0	CL0	DL0	EL0

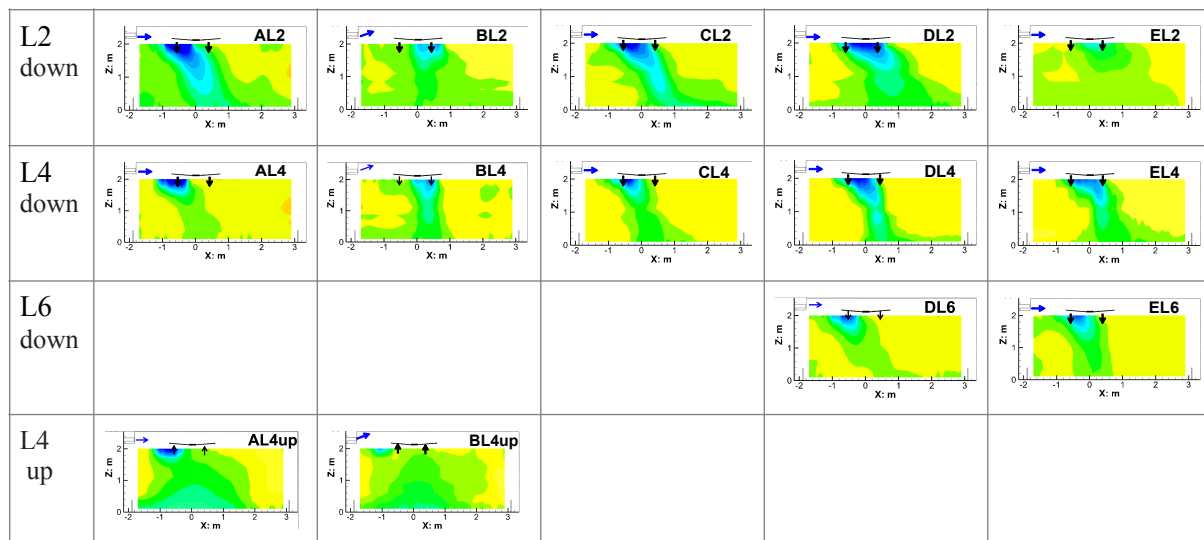|					
L2 down	AL2	BL2	CL2	DL2	EL2
L4 down	AL4	BL4	CL4	DL4	EL4
L6 down				DL6	EL6
L4 up	AL4up	BL4up			

Fig. 8. Temperature variation T' from the average temperature of the entire chamber in XZ plane. Fan air speed: 0.6 m/s (L2), 1.2 m/s (L4), 1.8 m/s (L6).

3.1.5.2 Temperature stratification

The vertical temperature distributions under CFIAC at the two locations labelled in Fig.3: (far from the fan, P1, and close to the fan and along the supply vent centerline, P2), are plotted in Fig. 9a-c for P1 position and Fig. 9d-f for P2 position, for Series A – D and F. (note: the E series is missing here due to a chamber supply fan failure that occurred during testing). These figures separately show the stratifications for no-fan, fan-down, and fan-up conditions.

All the larger stratifications in the two figures occur in cases where the fan is not running (AL0, BL0, CL0, DL0, FL0, lines with solid legends). In position P1, without the fan running, the temperature at the 2.4 m height is approximately 1 °C warmer than the temperature at the 0.1 m height. Because the position P2 is close to the supply vent centerline, the temperature at the 2.4 m height is affected by the supply air and is 1 °C colder than the temperature at the 0.1 m height. In the B series, with the elevated supply jet, the room temperature is very uniform below 2 m, and the top measurement height close to the ceiling (2.4 m) is the coldest. The vertical temperature distribution is very uniform whenever the fan is running. For location P1, all the stratification is within 0.3 °C between the 0.1 and 2.4 m heights, both for the center fan and for the corner fan located on the opposite side of the room from P1 (F-series). For the P2 location, most of the stratifications are also small, around 0.3 °C.
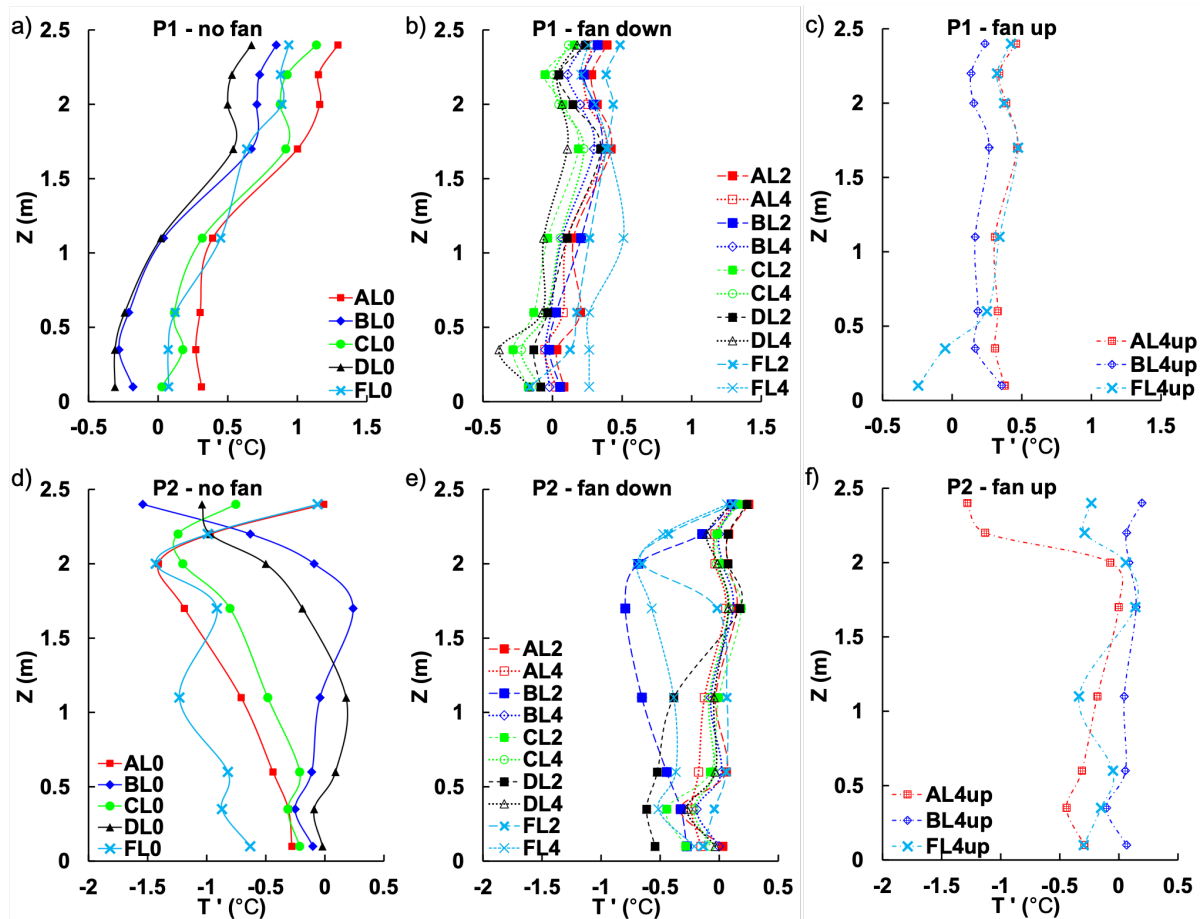
Fig. 9. Vertical temperature stratification of Series A-E in two points, a-c) P1, d-f) P2 (See Fig. 3).

The low temperature stratification that exists whenever the fans are running (both upward and downward, for center and corner fan) suggests that ceiling fans located anywhere in the room (both within or outside the supply air jet) will mix the air temperatures more strongly than buoyancy effects can separate them.

3.2 Horizontal distributions of velocity and temperature

3.2.1 Maps of horizontal velocity distribution

Fig. 10 shows the velocity distributions at the 1.1 m height for all configurations except the lost E series. Without the fan running, Level 0 shows that the up-tilted vent and higher supply flow rates move the supply jet further into the space before descending, as is also seen above in Fig. 4. The velocity across the room is mostly around $0.1 - 0.2$ m/s, with a few high speed areas up to $0.3 - 0.4$ m/s.

With the ceiling fan running in the downward direction (Fig. 10 row 2 and 3), and the fan on centerline (column $1 - 4$) or in the corner (column 5), the high velocity region is limited to a zone directly under the fan, reaching 1 m/s for Level 2 fan and 1.5 m/s for Level 4 fan. Outside of the fan diameter, the fan speed is uniform but low, around 0.2 m/s for Level 2 fan and around $0.2 - 0.4$ m/s for Level 4 fan. When the fan is operated in the upward direction, the velocity is uniform everywhere but also low, around 0.3 m/s. For the downward corner

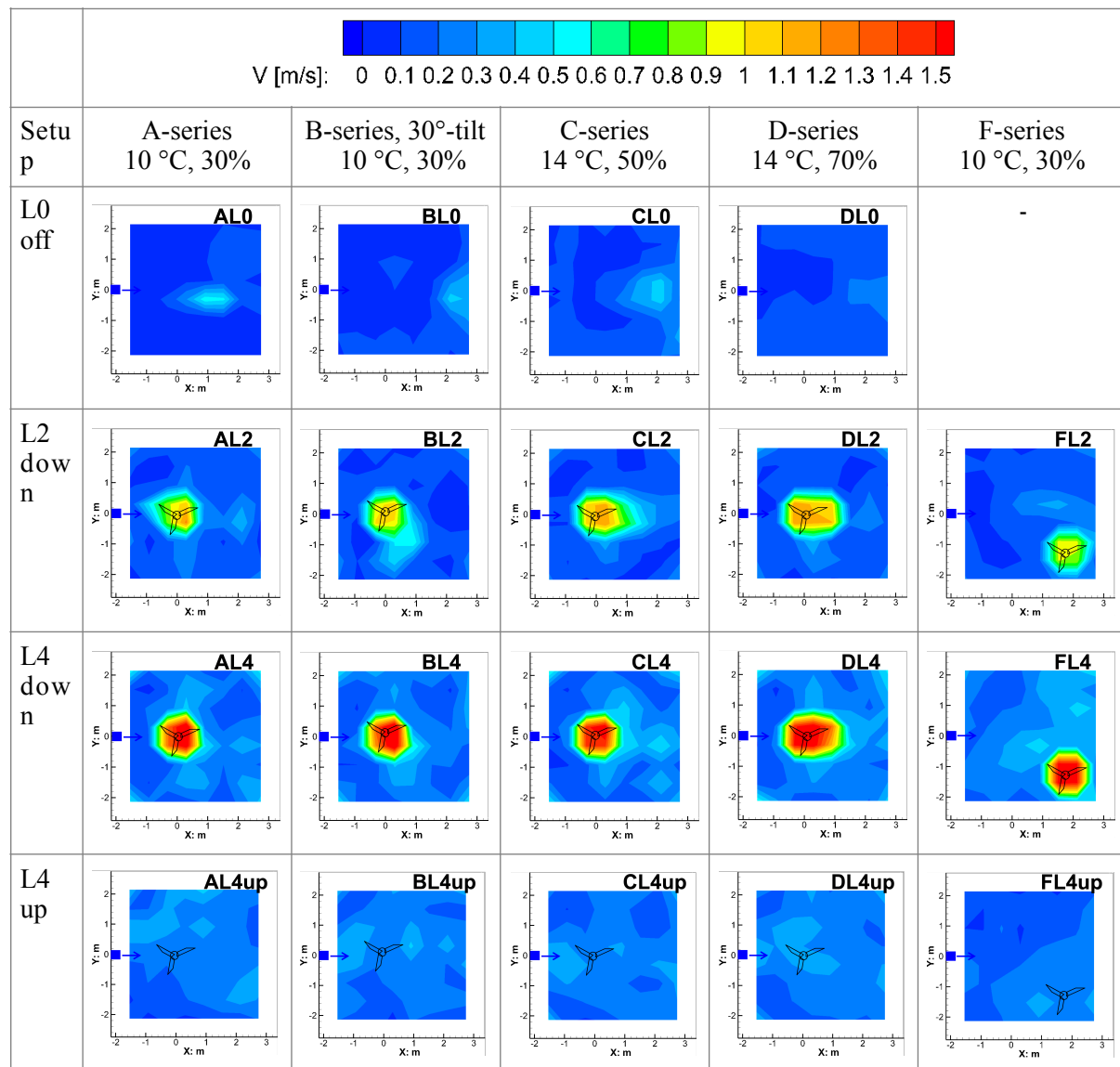fan case at level 4, the velocity in the room is slightly higher (around 0.4 m/s, see FL4 in Fig. 10).

Setup	A-series 10 °C, 30%	B-series, 30°-tilt 10 °C, 30%	C-series 14 °C, 50%	D-series 14 °C, 70%	F-series 10 °C, 30%
L0 off	AL0	BL0	CL0	DL0	-
L2 down	AL2	BL2	CL2	DL2	FL2
L4 down	AL4	BL4	CL4	DL4	FL4
L4 up	AL4up	BL4up	CL4up	DL4up	FL4up

Fig. 10. Horizontal velocity fields at 1.1m height, for center fan (Series A – D) and corner-fan (Series F). Fan air speed: 0.6 m/s (L2), 1.2 m/s (L4), 1.8 m/s (L6).

3.2.2 *Maps of horizontal temperature distribution.*
The temperature distributions at 1.1 m height for all the tested configurations are shown in Fig. 11 (as before, without the E series). With the ceiling fan running, the ambient temperatures all mixed very well, within 0.2 °C (see row 2 – 4). Without the fan (first row), the mixing is poor, reaching 1 °C temperature difference in the least mixed conditions of AL0 and CL0.

When the ceiling fan is not lined up with the supply vent jet (corner fan test case), the temperature mixing is still very good (Fig. 11, F Series). The warmest temperature happens at the opposite side of the room from the corner fan, but it is still only 0.4 °C warmer than the coolest location.
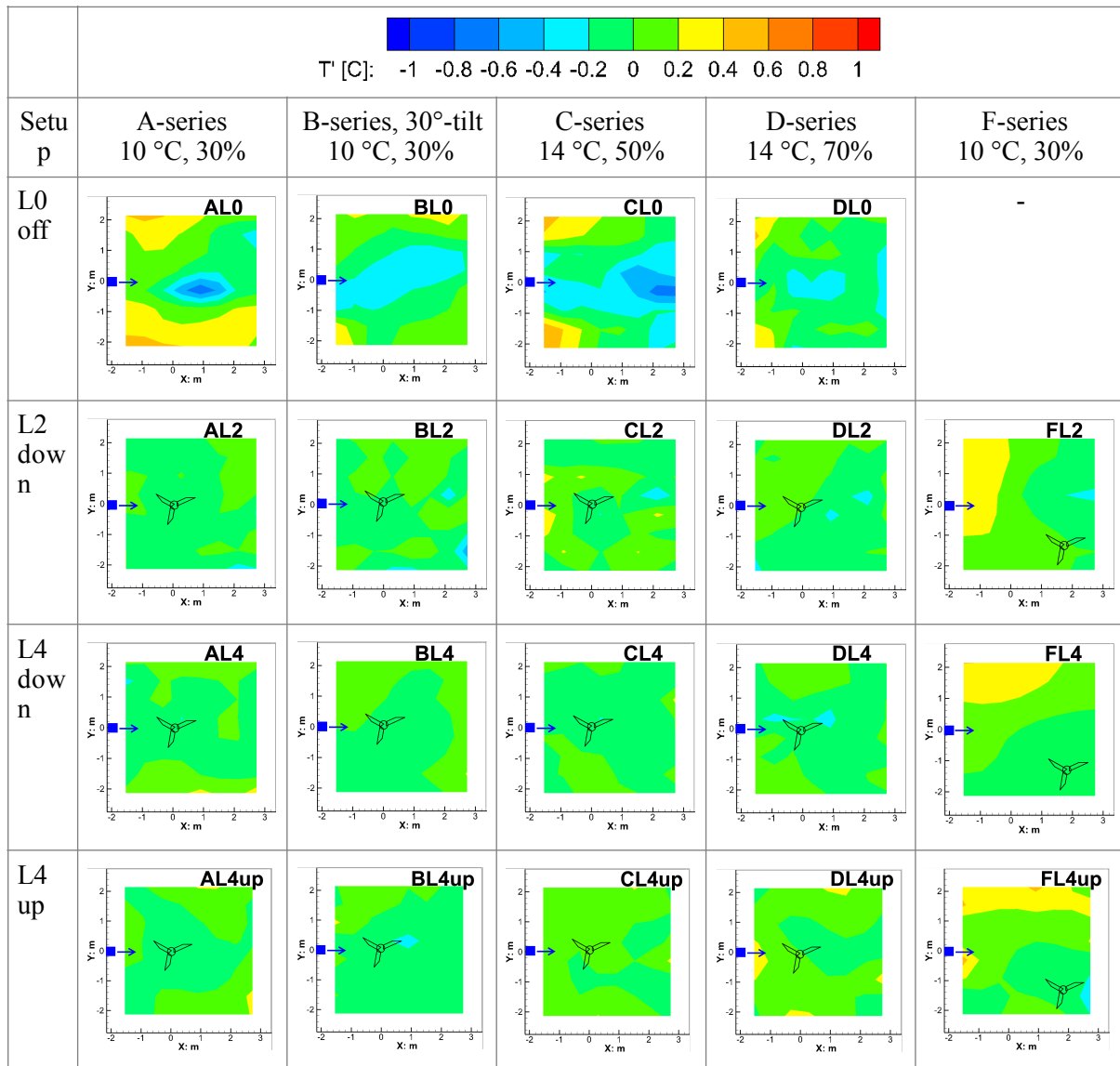
Fig. 11. Horizontal temperature fields at 1.1m height, for center fan (Series A – D) and corner-fan (Series F). Fan air speed: 0.6 m/s (L2), 1.2 m/s (L4), 1.8 m/s (L6).
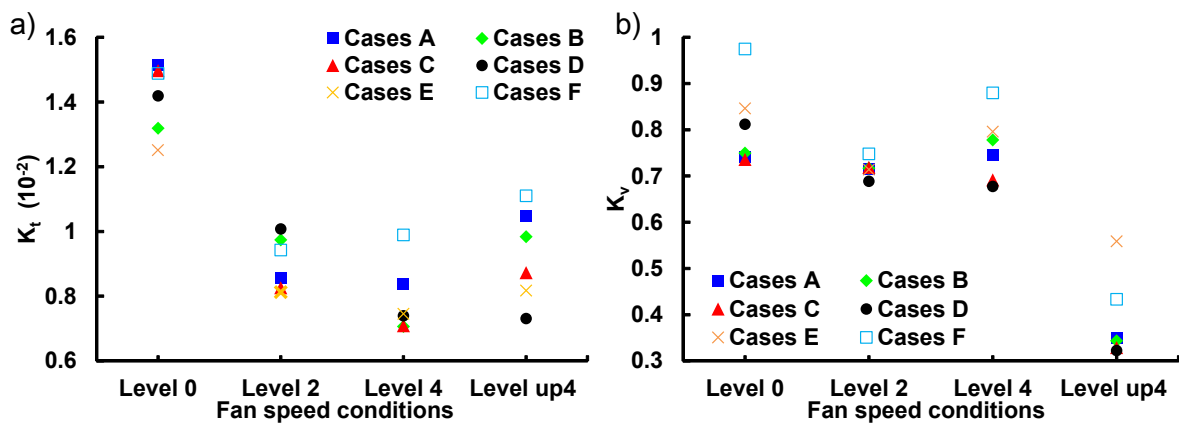

Fig. 12. The non-uniformity coefficient a) K_t and b) K_v under different Series.

3.2.3 *Non-uniformity of velocity and temperature within the occupied space*

The measured velocities and temperatures from 256 points in the occupied zone (0.1, 0.6, 1.1 and 1.7 m) were used to obtain velocity and temperature non-uniformity coefficients (Equations 2 and 3). The temperature non-uniformity K_t (Fig. 12a) is 26% to 43% lower for any condition when the fan is on than for the AC-alone condition. Velocity non-uniformity K_v in contrast is roughly the same for all downward fan speeds as the AC-alone condition (Fig. 12b). The higher velocity in the spreading zone at floor level accounts for much of the K_v. The upward fan condition has about half the velocity non-uniformity as the downward conditions, due to the absence of significant momentum flows in the occupied zone.

3.3 Mapping comfort distribution in the room

Using the ASHRAE elevated air movement procedure for determining comfort, we calculated PMV values for a sedentary occupant positioned throughout the room, at 26 and 28 °C. The calculations were done with the ASHRAE/CBE Thermal Comfort Tool [8], the official computer program for the ASHRAE Standard 55 [2].

The PMV represents the mean thermal sensation of a population, representing 'cold' (-3), 'cool' (-2), 'slightly cool' (-1), 'neutral' (0), 'slightly warm' (+1), 'warm' (+2), and 'hot' (+3).

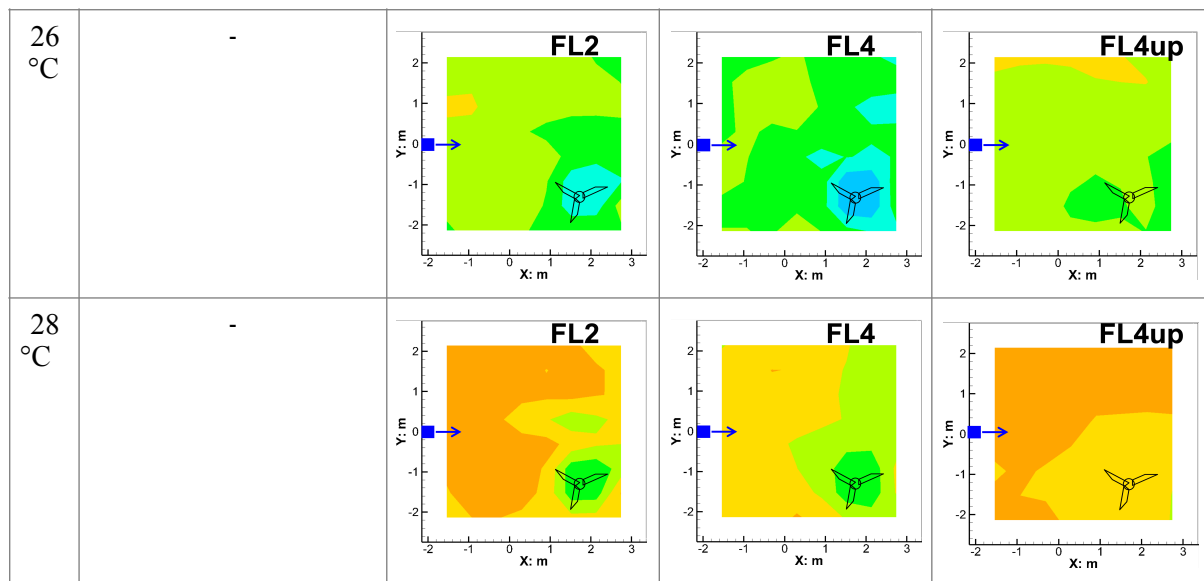|				
26 °C	-	FL2	FL4	FL4up
28 °C	-	FL2	FL4	FL4up
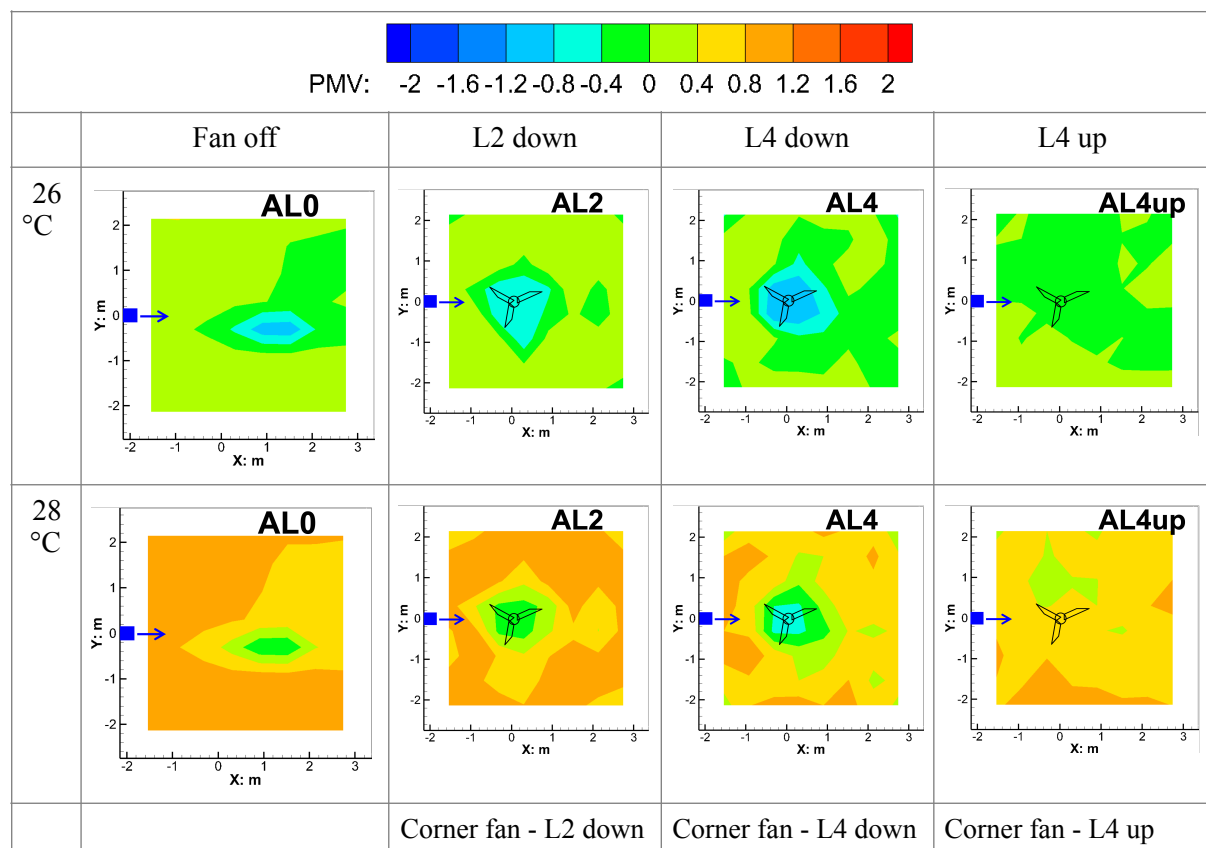
Fig. 13. PMV contours when fan is in the center (rows 1 – 2) and in the corner (rows 3 – 4). For A supply condition: supply temperature = 10 °C, and supply flowrate = 35% of maximum.

The calculations employ room temperatures and velocities averaged at the 1.1 m and 0.6 m heights. They omit ankle-height (0.1 m) temperatures and velocities, following the approach of the ASHRAE Standard 55 ankle draft risk procedure. This is because ankle cooling in warm conditions has a much smaller effect on comfort than cooling the upper body parts [52], and because the presence of office furniture in most real workplaces reduces the air movement seen in the test chamber at that level.

Fig. 13 shows comfort distributions for the A (center fan) and F (corner) series, which with their cold supply air temperature of 10 °C provide the greatest comfort challenge, at room temperatures of 26 and 28 °C. Rows 1 – 2 show central fan and rows 3 – 4 show the corner fan. Column 4 presents the upward fan direction.

The fan creates thermal sensations from 'neutral' to 'slightly cool' at the 26 °C ambient condition, and from 'neutral' to 'slightly warm' at the 28 °C ambient condition. In normal operation at 26 °C, the fan speed would be reduced (to AL2) or reversed (AL4 up) without overcooling anyone. At 28 °C, L4 would be the appropriate speed, but there are spaces where sensation is warm (PMV reaching to 1). In this case, increasing fan density (i.e. adding a fan or increasing the fan diameter) might be more desirable than increasing the fan speed level.

4. Discussion

The experimental work reported in this paper provides insight into how CFIAC will perform under a range of configurations and conditions: with the supply air flow aimed directly at ceiling fans or not, with different ceiling fan speeds directed upward and downward, with different supply flow velocities, and with different supply-air versus room-air temperature differences.

4.1 Physical layout and design of CFIAC systems

A): *Positioning the fans:* as described in *3.1.1 velocities in the XZ (vertical) plane*, the airflow profiles of CFIAC are dominated by the fan airflow profiles (levels 2, 4, and 6). Even at the low fan level the CFIAC profiles are very similar to those of a fan running alone [40, 44]. This suggests that the fans can be spaced and laid out using a pure ceiling fan design tool, such as has been developed from [31], http://cbe.berkeley.edu/fan-tool.

B): *Aiming the supply air vents:* To mix a cool supply air jet using a ceiling fan in the downward direction, is it better to have the jet better 'inhaled' into the fan from above the fan blades, or is it acceptable to have it 'collide' or 'merge' with the downward ring jet that exists below the blades? The temperature profiles in Fig. 8 compare an initially level but descending supply flow (Series A) against an elevated flow passing above the blades but in the fan inhalation zone (Series B). Much of the Series A supply flow merges with the downward ceiling fan jet at a height below the fan blades, cooling the supply vent side of the jet. At fan speed level 2, cool temperature extends furthest into the occupied zone. In contrast, the cooled supply flow in Series B first appears below the fan blades on the opposite side of the jet from the supply air vent, and it is more immediately mixed. The velocity distributions in Fig. 4 show the deformation of the downward fan jet by the 'colliding' supply, which is not present in the jet with the 'inhaled' supply. For downward fans in general, it appears that cool supply air would best be fed *through* the fan, rather than impinging into the downward-descending jet. But the evidence here shows that the fan jet mixes the air very well both ways, and there may not be enough of a difference in either of these issues (temperature and jet distortion) to make supply-jet aiming into a major design issue. For the upward-directed fan, the ideal situation would be to aim the supply jet at or above the fan. Comparing AL0 and AL4up in Figures 4 and 8, one can see that the fan did pull upward and fully inhale the lower half of the cooled supply jet which would otherwise have passed below the fan blades.

4.2 Using ASHRAE Standard 55 comfort analysis as the basis for determining CFIAC comfort

In ASHRAE Standard 55-2017 [7], 0.8 m/s is the upper air speed limit in spaces without occupant control. This elevates the comfort temperature by over 3 °C over that of still air [12, 58]. For spaces *with* occupant control, there is no upper air speed limit specified. Occupant control exists when there is one means of control available for groups of six or fewer, or for less than or equal to 84 m² (900 ft²) floor area or less. Multi-occupant spaces for shared activities, such as classrooms and conference rooms, qualify as occupant-controlled if they have one control available, regardless of room size; but if they are subdividable each division must have its own control. In addition, ASHRAE Std 55's new thermal environmental control classification awards classification credits for fans under individual and group control. These credits could be relevant to CFIAC in the future.

For sedentary activities (1.0 to 1.4 met) there is approximately 3 °C additional cooling provided by air speeds between 0.8 and 1.5 m/s, and experimental subjects have been observed to select fan speeds up to 1.5 m/s [59, 60]. There is very little additional cooling provided by air speeds above 1.5 m/s for sedentary activities, but at higher exercise levels (2, 4, and 6 met, as in a gym setting) speeds up to and over 2.0 m/s are effective and are chosen by subjects [11, 61]. The cooling effects of air movement on activity levels up to 2 met are

covered by the ASHRAE Standard 55 extended air speed provisions. They are simulated using the SET model [1], available in the CBE Thermal Comfort Tool (http://comfort.cbe.berkeley.edu/) [8]. The tool also simulates the effects of air movement cooling at higher met rates beyond the currently applicable range of Standard 55.

This gives us the opportunity to define a Standard 55- based comfort index for CFIAC using the official ComfortTool. With CFIAC operating in both heating and cooling, the different modes of fan use (for destratification in winter, air stirring for temperatures in the neutral zone, and for elevated air movement cooling in the warm season [62-64]), the comfort modeling needs to be able to transition from local draft discomfort considerations, through comfort temperature preferences obtained from field data [9, 10], to the elevated air movement model [65]. This method can now be carried out with the Comfort Tool for year-round simulations. The warm boundary of the comfort zone, above which there is undercooling due to insufficient air speed, would define the limit of CFIAC applicability. It should be noted that our current comfort zone boundary is +/- 0.5 PMV, but there is evidence from field studies [41, 66] that +/- 0.7 provides equal levels of occupant acceptability. This is subject for further study.

4.3 Draft risks at ankle level

Ceiling fan velocity fields are non-uniform, with higher flows near the floor. This poses potential draft risks at the ankle ($z = 0.1$ m) [67]. ASHRAE Standard 55 does not regard this a problem for thermally neutral-through-warm conditions, for which it specifies that its ankle draft risk model does not apply (it supplants it with the elevated air speed cooling procedure used in Fig. 13). In cool winter conditions, however, when fan-flows might be used for ventilation mixing and thermal destratification, the ankle draft risk model would become part of the evaluation metric.

4.4 Findings re comfort and ventilation effectiveness

As mentioned above, the authors have used same test arrangements and conditions reported in this paper to obtain comfort and acceptability perceptions from human subjects, and also to measure the ventilation effectiveness of CFIAC using tracer gas measurement. These two studies are being reported in two subsequent papers. It is worth mentioning two findings from these studies here, since this paper has served to introduce the CFIAC concept:

A) The thermal acceptability ratings of test subjects exposed to air movement are better than those deduced from +/-0.5 PMV obtained through the ASHRAE method used in Fig. 13. The test shows simultaneous satisfaction under both the ceiling fan air jet and the surrounding lower speed areas, under test conditions (Level 4 down-flow at 26 and 28 °C) in which the model predicts as much as 1.5 PMV scale unit difference.

B) The ventilation effectiveness for the CIFAC system is high, due to the high rate of mixing within the room caused by the ceiling fans. The age of air is reduced 20 – 40% for all fan-on conditions compared with the fan-off conditions.

Both the above results above are positive, lending support to the premise that CFIAC systems are a robust and resilient concept.

4.5 Further issues

CFIAC raises many questions about comfort under air movement that have not been studied to date. Answers will be needed in building up the expertise needed to implement optimized systems. How do people perceive spatial variation in room air speed, such as occurs in the fan jet, the low-level spreading zone, and the still air zone, under different ambient temperatures? How readily can varied air flow pattern be infilled (e.g., by adding smaller fans at occupant/desk level or by increasing the ceiling fan density)? In such flow fields, what frequencies of turbulent or oscillating fan-induced flows are best for comfort? How are air movement effects perceived over day-long periods? How much thermal adaptation occurs to higher indoor temperatures when the occupant is involved in cooling by enhanced convection over longer (say weekly) periods? Sound levels of modern fans are almost imperceptible when run in their medium speed ranges as in this study, but when dealing with extreme demands occupants have the ability to run them at higher, audible, speeds—over what lengths of time would this be acceptable? Some of these are of course general and fundamental questions that would benefit all uses of air movement indoors, such as in mixed – mode and naturally ventilated (operable windowed) spaces, as well as CFIAC.

4.6 Limitations

In this study, air speed was measured by omnidirectional anemometers that do not measure airflow direction or details of vortices or turbulence. It was also difficult to do direct measurement in the mixing zone near the ceiling fan. Detailed flow information in the mixing zone might be useful for CFD simulation considering the size of the rotating domain [68, 69]. In developing future CFD models, particle image velocimetry (PIV) [70, 71] or particle streak velocimetry (PSV) [72, 73] might be used to investigate CFIAC flow inputs into ceiling fans.

This study's test fans and supply vent are fairly close to the ceiling. We achieved different degrees of Coanda effect in the various test series. Since Coanda might not occur for rooms with higher ceilings and larger distances between the fan and the ceiling, further details on supply-air-aiming considerations without Coanda are desirable.

Our test chambers did not contain furniture, which would have mixed the air in the lower occupied zone [40] and increased velocity uniformity over the values presented here. Our results are therefore conservative in terms of velocity uniformity. Similarly, positioning the load-balancing heat sources around the walls of the room creates an atypical room but isolates the effect of occupant heat plume locations, which are typically unpredictable within rooms and whose permutations are nearly infinite. This applies also for the plan area of the test chamber, which is smaller than most open-plan office spaces, so the observed wall effects on circulation would be expected to occur differently in larger spaces. Finally, some of the testing was done with a very cold 10 °C supply air temperature to maximize the observed buoyancy and mixing effects. The benefit of these measurement decisions is in providing

conservative estimates of temperature, velocity, and comfort uniformity in the occupied space, relative to the fan jets themselves, and relative to the balance between the cooled supply air jet and the fan jet. But by itself these estimates might result in overdesign, so it will be valuable in the future to obtain detailed field measurements of CFIAC airflow profiles in examples of real furnished and occupied workstations, retail stores, lobbies, etc.

Since ceiling fans are also useful when buildings are in heating condition, the flow patterns of CFIAC with warm buoyant supply air jets need further investigation, for both upward- and downward fan directions. From this study we would expect the room circulation to be dominated by the ceiling fan flows, even at low speeds.

5. Conclusions

The study addresses the room airflows resulting from supply air jets from a high sidewall vent interacting with ceiling fans under CFIAC cooling conditions. The following conclusions are noteworthy.

- Temperature is highly uniform across the room for all the fan-on configurations, including fan speed, up/down direction, and center/corner fan locations. At 1.1 m height, the temperature differences across the room are within 0.2 °C for the center fan location, and within 0.4 °C for the corner fan location. All fan-on conditions reduce the temperature non-uniformity coefficients across the occupied zone by 26% to 43%, and eliminate the cool spot caused by dumping in the fan-off condition. There is virtually no temperature stratification at points in and out of the supply jet centerline when a fan is operating.

- The ceiling fan flow dominates the airflow patterns of CFIAC in the occupied zone, across a wide difference in supply air flowrate/temperature and ceiling fan velocities. The ceiling fan eliminates supply air dumping caused by negative buoyancy, even in the corner position where the fan is not in line with the supply air jet. The air flow patterns in the room for many conditions are largely self-similar even at the lowest fan velocity, and they resemble isothermal fan flow patterns published in the literature [44]. In CFIAC design, this simplifies the layout of ceiling fans and their positions relative to the vents.

- In mapping comfort, even with the largest velocity variations created by the highest downward fan speeds, comfort variations were equivalent to those of AC alone. At 26 °C, predicted thermal sensations ranged from 'neutral' to 'slightly cool'; at 28 °C, the air movement created sensations ranging from 'neutral' to 'slightly warm'. The maximum range seen is 1.2 PMV scale units at 26 °C and 1.5 at 28 °C. The actual comfort importance of these ranges will be seen in human subject tests, but these are encouraging results.

Acknowledgements

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