Ceiling-fan-integrated air-conditioning: thermal comfort evaluations

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ABSTRACT

Ceiling-fan-integrated air-conditioning (CFIAC) is a concept in which terminal supply ducts and diffusers are replaced by vents/nozzles that jet supply air into the vicinity of ceiling fans to be mixed and distributed within the room. CFIAC distributes the supply air within the room and convectively cools the occupants. This could allow raised thermostat set-points and reduced energy for the air-conditioning system’s supply fan and compressor. Previous work on CFIAC shows that the air temperature in the occupied zone is spatially uniform, but the air speed is not. This paper evaluates the thermal comfort performance of a CFIAC system under various ambient temperatures and fan operation modes. Human subject experiments and thermal manikin tests were conducted to characterize how subjects evaluate the thermal comfort performance across the room’s floorplate, and how CFIAC affects human body heat transfer. Despite the spatial variation in air speed across the floorplate, CFIAC created uniform thermal comfort perceptions. Comfort at 28°C was similar to that of 26°C for the overhead supply neutral reference condition. Human subjects preferred having the increased air movement over that of the reference condition. The paper evaluates thermal comfort indexes appropriate for evaluating and designing CFIAC.

PRACTICE RELEVANCE

Despite the spatial variation in air speed across the floorplate, CFIAC created quite uniform thermal comfort perceptions. Conventional air-conditioning systems often create overcooling complaints because supply volumes through diffusers are kept overly high in order to disperse cold temperature dumping in the space. Ceiling fan circulation provides...
sufficient dispersion to eliminate this issue. However, it is important for standards writers and designers to understand that the room temperatures should not be so cold that the highest airspeeds caused by the ceiling fans will be uncomfortable for the occupants in the fan-cooling zone. To evaluate CFIAC systems, the standard effective temperature (SET) model is shown to be useful and the elevated air speed method in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 55 also provides appropriate design guidance.

**NOMENCLATURE**

| Symbol | Definition |
|--------|------------|
| A      | Whole-body surface area (m²) |
| AC     | Air-conditioning |
| ADPI   | Air diffusion performance index |
| AP     | Air movement preference |
| CFIAC  | Ceiling-fan-integrated air-conditioning |
| CP     | Corrective power (°C) |
| D      | Fan diameter (m) |
| EHT    | Equivalent homogenous temperature (°C) |
| I_{clo} | Clothing insulation (clo) |
| HVAC   | Heating, ventilation and air-conditioning |
| MET    | Metabolic rate (met) |
| NV     | Natural ventilation |
| PD     | Predicted dissatisfaction due to draft |
| PPD    | Predicted percent dissatisfied |
| PMV    | Predicted mean vote |
| Q      | Heat loss (W) |
| R²     | Correlation coefficient (r) squared |
| RH     | Relative humidity (%) |
| SD     | Standard deviation |
| SET    | Standard effective temperature (°C) |
| T_{air} | Air temperature (°C) |
| TAV    | Thermal acceptance vote |
| TP     | Thermal preference |
| T_{r}  | Radiant temperature (°C) |
| TSV    | Thermal sensation vote |
| VAV    | Variable air volume |
1. INTRODUCTION

Heating, ventilation and air-conditioning (HVAC) systems account for over 30% of building energy consumption (Pérez-Lombard et al. 2008) and 3–5% of first cost (HomeGuide 2021). Part of the energy and first cost is consumed in order to deliver conditioned air uniformly into the room, using terminal ducts and diffusers. Another portion depends on the size of the air-conditioning (AC) system itself and the thermal loads it must meet.

Ceiling-fan-integrated air-conditioning (CFIAC) addresses both of these energy and cost concerns. Within the room, it eliminates terminal ductwork from the ceiling by using ceiling fans to mix and distribute the air supplied from nozzles in the central supply ducts (Chen et al. 2020). The jets of supply air are first directed into the vicinity of the ceiling fans, then the ceiling fans serve to mix and distribute the supply air within the room. In this new concept, the supply air terminals and the ceiling fans will work in a coordinated manner, running together or separately in different operation modes.

This has several system- and comfort-related benefits:

• The eliminated ductwork reduces the pressure drop for the system fan, and also reduces the cost and visual constraints associated with overhead ducts in the room (Dai et al. 2021).

• The room cooling set-point can be raised because the air movement from the ceiling fan can cool the occupants directly. Raising the cooling set-point can reduce the energy demand, and system size, of the cooling plant (Ghahramani et al. 2016; Hoyt et al. 2015).

• CFIAC can respond to individual occupants’ demands better than conventional AC systems and provide more flexible control over the thermal environment. This is because ceiling fan air movement is spatially localized, and changing fan speeds is much quicker than changing the room temperature through AC (Wang et al. 2020; Yang et al. 2010).

• CFIAC’s elevated air circulation may be generally preferred over still air and cooler temperatures. Building occupants have been found to desire more air movement for thermal comfort and for their sense of air quality (Toftum 2004; Zhang et al. 2007).

To move CFIAC into widespread real-world practice, new information is needed to provide guidelines for its control strategies and physical designs.

1.1 THE AIR-MOVEMENT AND TEMPERATURE CHARACTERISTICS OF CFIAC

Velocity and temperature distributions created by ceiling fans operating together with an HVAC mode have rarely been reported. One field study measured the vertical temperature stratification and thermal comfort in a classroom using ceiling fans and a ceiling-mounted AC unit (Momoi et al. n.d.). Another study measured temperature and air speed in AC spaces operating together with ceiling fans (Present et al. 2019). Both studies installed ceiling fans in AC spaces only to provide additional cooling, not to mix the supply air as in CFIAC, but the observed patterns of ceiling fan cooling are pertinent. In the authors’ previous paper (Chen et al. 2020), testing ceiling fans in a space with a high-side-wall supply vent, the velocity patterns (mean and turbulence intensity), temperature uniformity, and stratification were characterized under different fan operation modes (various fan speeds and both airflow directions) and supply air-conditions (various airflow volumes and air temperatures).

The measurement results (Chen et al. 2020) showed that the airflow patterns of the CFIAC are dominated by the ceiling fan flow patterns across a wide range in supply air flow rate/temperature and fan speeds. The ceiling fan eliminates supply air jets in the occupied zone even for fan locations that are well to the side of the supply air jet. The airflow patterns in different conditions are largely self-similar even at the lowest fan velocity, and they resemble isothermal fan flow patterns published in the literature (Chen et al. 2018). Another study (Wang et al. 2020) integrated a network of smaller fans on the ceiling with an AC vent, reporting that even small fans can create distinct indoor airflow patterns by manipulating the operating direction and air speeds of the fans. The airflow pattern could be significantly modified based on occupants’ ventilation needs, and those fans could serve as air terminals for demand-oriented ventilation.
The temperature field in a room using CFIAC is well-mixed and highly uniform for all the fan-on configurations, regardless of fan speed, operating direction, and fan locations (Chen et al. 2020). At 1.1 m height, the temperature differences across the room were within 0.2°C for a center fan location, and within 0.4°C for a fan located in the corner of the room well away from the supply jet. Turning the fan on reduced temperature non-uniformity across the occupied zone by 26–43%, and eliminated the cool area caused by the descending supply air jet in the fan-off condition. There is virtually no temperature stratification at points either in or out of the supply jet centerline when a fan is operating (Chen et al. 2020).

1.2 THERMAL COMFORT ISSUES RELATED TO CFIAC

The air movement and temperature characteristics of CFIAC raise many questions about an occupant’s thermal comfort. The following questions are particularly important if CFIAC is to move into real-world practice:

- **Will CFIAC prevent the perception of overcooling and drafts in cool conditions?**

  Conventional variable air volume (VAV) systems frequently overcool zones by supplying excessive flows under low occupancy/load conditions. The high supply flows are typically in response to designers’ concern about the ‘dumping’ of a cold supply air into the occupied zone when flow rates are low, since static diffusers relying on flow momentum may not be able to mix the flows sufficiently (Arens 2015). CFIAC could in theory eliminate the designers’ dumping concern by mixing supply flows forcibly via the ceiling fans, permitting lower supply flows and warmer space temperatures. Beyond this, the actual perception of cool supply flows by occupants has been found in field tests to occur more frequently at high supply flow rates, where the negatively buoyant jets reach the occupied zone without sufficient dispersion (Arens 2015). Such cool jets were found (Chen et al. 2020) to be largely dispersed by ceiling fans operating in either direction within the room. The comfort effects under the ceiling fans’ homogenized space temperatures and potentially wide-ranging elevated air speeds have not been established across a range of likely CFIAC operating conditions.

- **Can CFIAC allow operators to raise the cooling set-point to save HVAC energy without decreasing comfort?**

  The thermal corrective power (CP) of ceiling fans operating by themselves has been established in many climate chamber studies of humans (Zhai et al. 2013) and thermal manikins (Luo et al. 2018). In the proposed CFIAC sequence of operation, the fans would operate independently of AC cooling until interior temperatures reached a cooling set-point; above that set-point the fans would remain at their highest practical speed at the same time as air-conditioned air is supplied to maintain the set-point temperature. This type of operation has not been explicitly tested on subjects to establish optimum combinations of temperatures and air speeds.

- **Can CFIAC achieve the same (or better) performance than a conventional overhead diffuser arrangement in neutral ambient temperatures?**

  The field study findings of preference for air movement (Toftum 2004; Zhang et al. 2007) should be tested under controlled conditions. Also, since ceiling fan airflows in the occupied zone are not spatially uniform (especially when fans are operated in the more efficient downward direction), the comfort performance comparison must be established at various locations throughout the room.

- **Which thermal comfort indices might be applied to evaluate thermal comfort within the elevated air movement and temperatures created by the CFIAC?**

  In conventional HVAC systems, interest in air movement has in the past been concerned with drafts in cool temperatures, under which even very low air speed can cause cold discomfort. HVAC diffuser manufacturers use the air diffusion performance index (ADPI) for evaluating the uniformity of room airflow and temperature distributions, with the goal of creating uniform temperatures (especially minimizing thermal stratification) and still-air in the room (low draft risk in cool temperatures) (American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) 2019). The maximum allowable velocity in ADPI is 0.2 m/s, well below
the range of air speeds that are encouraged in CFIAC. The ADPI approach cannot be directly converted to evaluate CFIAC designs (Chen et al. 2020). The index predicted percent dissatisfied (PPD) based on the 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% (Fountain 1991) and has been found to be inaccurate in field studies (Arens et al. 2010). ASHRAE Standard 55-2017 (ASHRAE 2017) has since 2009 predicted cooling effects of elevated air speeds using the standard effective temperature (SET) model (Arens et al. 2009; Gagge et al. 1971), which incorporates the explicit simulation of evaporative effects. Even though it is a whole-body heat-balance model, SET may be effective under the non-uniform combinations of temperatures and air flows created in a room by CFIAC. It has been found to predict comfort well in human subject tests in which horizontal fan air flows cooled only the upper half of the body (Huang et al. 2014). This might be confirmed under the variety of flows experienced under ceiling fans.

This study was planned to address these questions, measuring the thermal comfort performance of a CFIAC system at different locations across the room floorplate under various ambient temperatures and fan operation modes. To do this, thermal manikin tests were first conducted to characterize CFIAC heat transfer effects. Human subject experiments were then conducted to collect the subjective evaluation of the CFIAC thermal conditions. Both down- and upward flow directions were tested because ceiling fans are sometimes operated upwards to reduce the spatial variability and strength of airflows in the occupied zone during neutral and cool conditions.

2. METHOD

2.1 SET-UP OF THE CFIAC SYSTEM

The CFIAC system was set up at the Center for the Built Environment (CBE) at the University of California—Berkeley following the same geometry tested by Chen et al. (2020). The dimensions of the office-style climate chamber were 5.5 × 5.5 × 2.53 m (Figure 1a). A ceiling fan (Haiku 60, Big Ass Fans, Inc.) of 1.5 m diameter (D) was installed near the center of the room and 0.37 m below the
ceiling. A supply vent (0.184 × 0.133 m) of 0.155 m equivalent diameter was mounted 2.15 m high on one wall, approximately midway along its length (Figure 1a, b). The distance between the center of the vent and the ceiling was 0.38 m (2.15 m from the floor), and the grille was mounted proud of the wall in a 0.36 × 0.33 × 0.36 m box. The supply vent register (Price Industries 520 Grille) has adjustable airfoil vanes allowing the supply air throw direction to be adjusted vertically. The 0.61 × 0.61 m exhaust air grille was located in the ceiling on the other side of the room from the supply, 0.9 m from the wall. The chamber’s exterior walls and three-layer windows had an air gap behind their inner surfaces through which conditioned air circulated to maintain the chamber’s walls and windows at equal surface temperatures as in the ambient. By reconnecting air ducts in the ceiling plenum, the room was also operated to create a reference case simulating a conventional AC system using an overhead supply diffuser (Accord Ventilation, 0.61 × 0.61 m) and exhaust.

The supply air temperature was set at 10°C and airflow volume was 0.056 m³/s, 35% of the maximum supply air rate. The low supply flowrate and temperature were chosen because they allow CFIAC to be evaluated for the worse-case scenario in which there is less mixing and potential dumping. In order to simulate office-like internal loads without creating rising plumes within the testing space, 14 electric heating panels (170 W each, total 2380 W, about 80 W/m² floor area) (Figure 1a, d) were arrayed around the chamber’s perimeter 0.025 m from the wall. Each heating panel was 0.83 m long, 0.61 m wide, and 0.03 m thick, and covered with aluminum foil to minimize its radiant exchange with the subjects. The room temperature was controlled to slightly cool (23°C), neutral (26°C), and slightly warm (28°C) conditions. When there were no subjects in the chamber, all the 14 heating panels would be turned on. When a certain number of subjects entered the chamber (one to five people depending on the test cases), the same number of heating panels would be turned off. The temperature was monitored under the supply air vent at three heights (0.1, 0.6, and 1.1 m) (Figure 1a) and on the Y = 0 wall at 1.1 m height (Figure 1c). According to previous measurements (Chen et al. 2020), CFIAC yields a highly uniform temperature distribution across the room for all the tested fan-on configurations.

Figure 1c shows the locations where the manikin and subjects are seated and tested. They were chosen based on the air movement and temperatures measured in the previous study (Chen et al. 2020). P1 and P1’ are locations along the supply jet centerline where subjects would experience air movement. P1 is in the cold supply air dumping zone when the ceiling fan is off; the airspeed is about 0.3 m/s and the temperature is 1–2°C lower than in surrounding locations. P1’ is under the fan at half blade length where the velocity is highest; the air speed ranges from 0.6 (low fan speed downward) to about 1.6 m/s (medium fan speed downward). P2 and P4 are both outside of the ceiling fan and supply air jet zones so that their air speeds are the lowest, usually 0.1–0.3 m/s when the fan is blowing downward. P3 is close to the room perimeter where the test (Chen et al. 2020) showed the potential for air movement caused by the radial outflow from the fan along the floor and then moving up the room walls. When the ceiling fan is reversed to blow upward, the airflow will flow radially outward from the fan along the ceiling and then descend along the wall. For both fan directions, there will be higher air speeds in perimeter zone location P3. Thermal comfort evaluations in locations P1–P3 cover the range of typical air movement and temperature zones created by the CFIAC system when varying the fan operation modes.

2.2 THERMAL MANIKIN TESTS

Before the human subjects, thermal manikin tests were conducted to evaluate how the spatial inhomogeneity in air movement and temperature would affect human body heat transfer at different locations on the floorplate. The test condition, CFIAC set-up, and selected locations are given in Table 1. The reference case is a neutral condition at 26°C with supply air from a normal overhead diffuser so that other CFIAC cases can be compared with it.

A thermal manikin with 16 individual segments (Tanabe et al. 1994) was used to perform the tests. Under the slightly cool (23°C) and neutral (26°C) conditions, the manikin wore normal wintertime office clothing, including T-shirt, long-sleeve shirt, long trousers, and socks. Its insulation was 0.65 clo, excluding the thermal resistance provided by the mesh chair. In the warm (28°C) condition, the manikin wore the same clothing, but without the long-sleeve shirt, an insulation of 0.5 clo. In
all the test conditions, the skin surface temperature $T_{sk}$ of all manikin’s segments was uniformly maintained at 34°C to represent a state of comfort across the body. Each test session lasted for two hours to allow the manikin to reach a stable state when the heat loss change of all body segments was within 3% during the last half-hour. The heat dissipation and skin temperature for each manikin segment were recorded every 1 min. The last 10 min of data were used for further analysis.

### 2.3 HUMAN SUBJECT TESTS

After the manikin tests, human-subject tests were carried out to collect subjective comfort evaluations by the subjects. The test conditions, CFIAC set-up, fan operation modes, and selected locations are given in Table 2. In total, 21 test sessions were performed, with three ambient temperatures (23, 26, and 28°C), four ceiling fan operation modes (low downward, medium downward, medium upward, and fan off), two types of ceiling fan (types 1 and 2), and four locations (P1, P1’, P2, and P3). (P2’ and P4 were assumed to be identical to P2.) The overhead diffuser condition without a ceiling fan served as the reference case. Given that the available labor and financial resources were limited, not all the human subject tests covering a full test matrix were performed. Instead, the test conditions were selected to answer the following questions:

- **Cool condition-related issues**
  The upward-operating ceiling fan tests at 23°C compared with the no-fan case in cool condition examine whether the upward-operating fan can prevent cold drafts in the supply-air dumping zone (P1).

- **Warm condition-related issues**
  Tests in which fan directions and speeds vary in neutral and warm conditions explore whether the CFIAC system can extend the temperature set-point to the warmer side while maintaining the same or even higher thermal comfort level as the neutral reference condition, for different locations where the air movement varies.

- **Fan flow direction-related issues**
  The fan type 2 upward-operating with high-speed level aims to see whether a fan with a higher upward airflow would reduce air speed variability in a room. To compare with a typical

| CFIAC SET-UP               | P1 | P1’ | P2 | P4 |
|----------------------------|----|-----|----|----|
| High-side wall diffuser    | ✓  | ✓   | ✓  | ✓  |
| Medium up                  | ✓  | ✓   | ✓  | ✓  |
| Low down                   | ✓  | ✓   | ✓  | ✓  |
| Supply air from an overhead diffuser with no ceiling fan with a room temperature of 26°C | ✓  | ✓   | ✓  | ✓  |
| Room temperatures          | 23, 26, and 28°C                   |
| Manikin skin temperature   | 34°C                         |

| AMBIENT TEMPERATURES       | 23°C | 26°C | 28°C |
|----------------------------|------|------|------|
| SET-UPS                    | P1   | P2   | P1’  | P2   | P1   | P1’  | P2 |
| High-side wall vent        | ✓    | ✓    | ×    | ×    | ×    | ×    |
| Fan type 1, medium speed, upward | ✓    | ✓    | ●    | ●    |       |
| Fan type 1, low speed, downward | ●    | ●    | ●    | ●    |       |
| Fan type 1, medium speed, downward |       | ●    | ●    | ●    |       |
| Fan type 2, high speed, upward |       |       | (comparing with P3) | (comparing with P3) |       |

**Table 1:** Summary of the thermal manikin tests.
Note: A few test locations were skipped because they were repeated under specific conditions. For example, for low and medium downward fan directions, location P1 was not tested because it is similar to P2 and P4.

CFIAC = ceiling-fan-integrated air-conditioning.

**Table 2:** Test conditions for the human-subject experiments.
Note: ✓ = No fan case and fan type 1 in cool condition; × = no-fan condition; ● = with fan conditions at 26 and 28°C; ∞ = fan type 2 upward-operating at a high-speed level; ☼ = comfort evaluations of an overhead flush-mounted ceiling diffuser under an neutral ambient temperature of 26°C. Fan type 1 (Haiku 60, Big Ass Fans, Inc.) has a diameter of 1.52 m and its maximum airflow under downward operation is 4.07 m³/s. Fan type 2 has a diameter of 1.32 m and a maximum airflow under downward operation of 3.17 m³/s.
HVAC air supply system, the comfort evaluations of an overhead flush-mounted ceiling diffuser under a neutral ambient temperature of 26°C were collected.

- **Test procedure, survey questionnaire, and skin temperature measurement**

  Figure 2 shows a 145-min test procedure. Each subject participated in four formal test sessions after a 15-min preparation and a 30-min acclimation period. Each test session lasted 20 min and was followed by a 5-min break interval. After the first two sessions, subjects stand, take 20 vertical up-steps, and switch their location, for example, from P1/P1’ to P2 or from P2 to P1/P1’.

  ![Figure 2: Test protocol example.](image)

  During the test, the CBE thermal comfort questionnaire tool (Luo et al. 2018) was used to collect participants’ whole-body and local thermal sensation vote (TSV; from very hot to very cold), thermal acceptance vote (TAV; from clearly acceptable to clearly unacceptable), and thermal/air movement preferences (TP/AP; cooler/more, no change, warmer/less). As shown by the inverted triangles in Figure 2, two types of questionnaires were designed to avoid fatigue due to repetitive questions. The short questionnaire (marked in dark grey) included whole-body and facial TSVs and the acceptance votes. This streamlined questionnaire was answered at the 10th and 20th min of the acclimation period and the 0th and 10th min of each test session. The long questionnaire (marked in orange) was the short questionnaire plus other local TSVs for the hand, forearm, thigh, and foot, and TP and AP. It was answered at the end of each test session. For a detailed description of surveyed questions, see the Test Description sheet in the supplemental data online.

- **Participants**

  A total of 24 healthy college-aged subjects (12 females and 12 males) were recruited to participate in the tests under warm, cool, and neutral conditions. The male and female groups comprised an almost identical proportion of Caucasian (five female and five male) and Asian ethnicities (seven female and seven male). Not all 24 subjects were able to participate in all the test conditions listed in Table 2 because of conflicts between their classes and the test schedules, but each test had at least 20 subjects. All the recruited subjects had light-to-no caffeine, alcohol, smoking habits, and had normal exercise intensity (two to four times per week). Before the tests they were asked to wear uniform clothing with the same insulation as in the manikin test, 0.65 clo for 23/26°C and 0.5 clo for 28°C. They were trained to be familiar with the questionnaire, and were informed that they would experience mild cool and warm exposures.

### 2.4 DATA PROCESSING

#### 2.4.1 Manikin tests

The heat dissipation and skin temperature for 16 manikin segments were exported into Excel files. The equivalent homogenous temperature (EHT) (°C) (McGuffin et al. 2002; Wyon et al. 1989) was calculated for local body parts and for the whole body. EHT quantifies the heat loss of a human body in a non-uniform condition by converting it to an equivalent homogenous, still-air ambient environment. A higher EHT represents a warmer ambient environment. For segment \( j \), \( EHT_j \) was calculated by equation (1a), where \( A_j \) is the surface area \((m^2)\) for the segment; \( T_s \) is the skin surface temperature \((\text{Kelvin degree})\); \( Q_j \) is the amount of heat loss \((W)\); \( I_j \) represents the total
clothing insulation (clo) of the segment (including the resistance provided by the air layer); and 0.155 is the ratio for converting the insulation unit clo to W/m². Whole-body EHT is calculated by equation (1b), where $Q_j$, $A_j$, and $I_{clo}$ represent the whole-body heat loss (W), surface area (m²), and total clothing insulation (clo), respectively:

$$EHT_j = Tsk - \frac{Q_j}{A_j} \times 0.155 \text{ (K)} \quad (1a)$$

$$EHT_{whole-body} = Tsk - \frac{Q}{A} \times I_{clo} \times 0.155 \quad (1b)$$

The EHT results can be converted into CP (Zhang et al. 2015). The CP values quantify the ability of the CFIAC to correct ambient temperature towards a person’s thermal neutrality. Equation (2) defines $CP_j$ for each body part $(K)$, derived from $EHT_j$ differences between a test condition and the reference condition (the overhead diffuser case). For the whole-body CP value, the same equation can be used, but the local body parts’ $EHT_j$ replaces the whole-body $EHT_{whole-body}$:

$$CP_{EHT_j} = EHT_j - EHT_{j-reference} \quad (2)$$

### 2.4.2 Human participant tests

For tests involving human participants, subjective voting data were first exported and stored in a .txt document through the user interface of the CBE questionnaire tool. It was then imported and organized in Excel files. The iButton skin temperatures for different body parts were matched with the subjective votes by time and test condition. All these data are given in the Data sheet in the supplemental data online. The data analysis was done in the open-source language R programming (v. R 3.6.1) using its functions of visualization and statistical calculations. Descriptive statistics such as the mean, median, minimum, maximum, and standard deviation were calculated. The acceptable rate was calculated by dividing the number of acceptance votes including and above ‘just acceptable’ by the total number of the votes. When calculating the statistics, evaluations from all the subjects were included.

To test whether thermal sensation and acceptance vary significantly between different conditions, two-way analyses of variance tests with location and temperature as the main factors were used and repeated for different ceiling fan statuses. If there were significant main effects or interactions, Tukey’s HSD tests were applied to identify which interaction caused the difference. Tukey’s HSD test results were considered statistically significant when $p \leq 0.05$. The interpretation code is as follows: $p \leq 0.001$ or ‘***’ means highly significant; $0.001 < p \leq 0.01$ or ‘**’ means significant; $0.01 < p \leq 0.05$ or ‘*’ means weakly significant; and $p > 0.05$ means not significant. In figures comparing different experimental conditions, only results with statistical significance are marked. In the ANOVA tests, only the evaluations from subjects who attended all the experimental conditions were included. To consider the effect size of significance test, Cohen’s $d$ was calculated between groups with significant difference. The $d$-values were then marked on the figures with the significance test.

### 2.4.3 Comparing the manikin and human tests

To compare the results from manikin and human subject tests against thermal comfort related indexes, the ‘comf’ package in R programming (Schweiker 2016) was used to calculate the PMV, PPD, predicted dissatisfaction due to draft (PD), and SET. The CP value based on SET was defined in equation (3) as the difference in SET in the still air and it assumed a uniform temperature case ($SET_{still air and uniform temperature case}$) versus the measured air speed and temperature ($SET_{measured air speed and temperature}$) (Gagge et al. 1971). The mean radiant temperatures were assumed to be the same with the air temperature for the SET calculation. Other parameters were set for a standard office worker (e.g. 50% relative humidity (RH), 1.1 met activity, 0.65 clo clothing insulation for 23 and 26°C, and 0.5 clo for 28°C):

$$CP_{SET} = SET_{still air and uniform temperature case} - SET_{measured air speed and temperature} \quad (3)$$
3. RESULTS

3.1 HEAT TRANSFER EFFECTS FROM THE MANIKIN TESTS

Figure 3 shows the whole-body CP values at different locations under slightly cool (23°C), neutral (26°C), and slightly warm (28°C) ambient temperatures. In the supply air dumping zone (P1), the EHT is 1.6–2.3°C lower than the overhead diffuser condition. Operating the ceiling fan upward can reduce this cold draft and make its EHT close to that of the reference condition. For the area under the fan (P1’), the downward-blowing fan can provide a 2–5°C cooling effect depending on the fan’s speed. For areas outside of the fan covered area and supply air dumping zone, P2 and P4 are almost identical with the reference condition when the ceiling fan is off but the AC is on, and identical with the condition when fan is blowing upward. The EHT differences are within 0.5°C. When the fan operates in a downward direction, the cooling effects in these two locations are 0.5–1.5°C depending on the fan speeds.

Figure 3: Whole-body corrective power (CP) in Equivalent homogenous temperature (EHT) at different test locations.
Note: The reference case is the ‘overhead diffuser with 26°C room temperature’ condition. Error bars represent the standard error among the three temperature conditions.

Figure 4 shows the CP values for different manikin segments in the supply-air-dumping zone (P1), fan-jet zone (P1’), and outside of the fan zone (P2), for three test conditions. In order to compare CFIAC’s effect on the manikin’s heat dissipation, the fan-off condition at each location was chosen as the baseline for calculating the CP. In the fan-jet zone (P1’), the downward-operating fan produced higher cooling effects at the head, forearm, hand, and foot, probably because the

Figure 4: Corrective power (CP) in equivalent homogenous temperature (EHT) at local body parts.
Note: The baseline case is the ‘fan off’ condition at each location. + = Warmer than the baseline case; and ‘−’ = cooler than the baseline case. Highlighted values are the top three affected body parts.
head and forearms of the seated manikin were directly exposed to the downward airflow from the ceiling fan. Outside of the fan zone (P2 and P4), the airflow from the downward-blowing ceiling fan spreads outward across the floor yielding higher airspeeds at ankle height; this causes a stronger cooling of the foot and lower leg of the manikin. For P1, the upward-operating fan blocked the cool supply air from the high-side-wall vent to this location directly, resulting in positive CPs compared with the fan-off baseline condition. The strongest effects happened at the head, hands, and thighs, possibly because these body parts were most directly exposed to the cold dumping air from the supply vent when the ceiling fan was off. These local CPs are about twice stronger than the whole-body average.

3.2 THERMAL PERFORMANCE FROM THE HUMAN SUBJECT TESTS

Table 3 presents descriptive statistics for whole-body thermal sensation, acceptance, and preferences in temperature and airspeed, under the different test conditions. The whole-body acceptance rate varied from 73.7% at lowest to 100%. The lower acceptance rates came from the cases with the fan off but the AC on in the cold dumping zone when ambient temperature was cool (23°C), and in cases with the fan off but warmer ambient temperature (28°C). The higher acceptance rates came from the cases with the ceiling fan on, which received a similar or higher acceptance at 28°C than the reference conditions (overhead diffuser at 26°C without a fan). Addressing which fan operation mode can yield the highest acceptance rate, it depends on the location of the subjects and ambient temperature. For the area under the fan, it was 100% acceptable at 28°C with the medium downward flow. For the cold supply-air dumping zone at 23°C, running the fan upward improved the acceptance by about 16% by eliminating the cool draft. The thermal and airspeed preferences also varied with the test conditions and will be discussed in the following sections.

To visualize the effects of ceiling fan operation mode on subjects’ thermal sensation, Figure 5 shows the distributions of TSV under different test conditions. Each color represents one of the fan operation modes. For the cold dumping zone (see the pink and grey distributions in the first column), the fan running upwards would shift TSV towards the warmer side of the TSV scale in both cases (23 and 28°C), which is as expected because it relieves the person from the cold draft. The difference is that under 23°C ambient conditions, such a shift to the warmer side leads to votes moving from cool towards neutral, while at 28°C the shift is from neutral away towards warm. Under the fan-jet (in the middle column), the low speed-downward blowing can correct 26 and 28°C to thermally neutrality (see the green distribution). The medium speed-downward blowing can even correct the slightly warm temperature (28°C) to a slightly cool condition (see the blue distribution). For areas outside of the fan-jet (third column), different fan operation modes do not produce significant changes.

Figure 6 shows the probability of airspeed preference for different test conditions. In the cold supply air dumping zone (first column), at 23°C, about 30% of subjects wanted less air movement when the fan was off. Running the fan upward reduced the ‘less air movement’ preference by half. At 28°C, there was a substantial portion (30–50%) of subjects in this zone who wanted more air movement when the fan was off or operating in upward mode. Directly under the fan (mid column), the low speed (0.6 m/s) at 26°C and medium speed (1.6 m/s) at 28°C downward fan made 25–45% of subjects prefer less air movement. The low speed at 28°C created an equal preference (25%) for both more and less, and about 50% wanted no change. For areas outside of the fan, there is a dominant preference for ‘more air movement’ over ‘less air movement’, even at 23°C, but this is particularly obvious at 26 and 28°C.

3.3 THERMAL COMFORT EFFECT OF THE CFIAAC

After presenting the findings from the thermal manikin and human subject experiments, the following section answers the questions addressed by the experimental design.
Table 3: Statistical results of the human subject test.

Note: The shaded row represents the reference condition test results. The airspeeds column shows the averaged values at 0.1, 0.6 and 1.1 m heights.

AC = air-conditioning; TAV = thermal acceptance vote; TSV = thermal sensation vote.

| TEST CONDITIONS               | LOCATION                      | TEMPERATURE (°C) | FAN OPERATION | AIR SPEED (M/S) | NUMBER OF SUBJECTS | MEAN   | SD    | MEAN   | SD    | ACCEPTANCE RATE (%) | COOLER NO CHANGE | WARMER LESS NO CHANGE | MORE |
|-------------------------------|-------------------------------|------------------|---------------|-----------------|--------------------|--------|-------|--------|-------|----------------------|-------------------|------------------------|------|
| Whole-body thermal sensation  | Air dump zone (P1)            | 23               | Fan off but AC on | 0.32            | 20                 | -1.00  | 0.82  | 1.21   | 2.03  | 73.7%                | 2                 | 9                      | 9    |
|                               |                               |                  | Medium up     | 0.24            | 20                 | -0.02  | 0.73  | 2.25   | 1.40  | 85.5%                | 3                 | 10                     | 7    |
|                               |                               |                  | Low down      | 0.33            | 20                 | 0.13   | 0.69  | 1.96   | 1.42  | 84.2%                | 3                 | 15                     | 2    |
|                               |                               |                  | Medium up     | 0.34            | 20                 | 0.28   | 0.50  | 2.13   | 1.37  | 94.4%                | 7                 | 13                     | 0    |
| Outside the fan zone (P2 or P4) | 26                           | 23               | Fan off but AC on | 0.03            | 20                 | -0.32  | 0.57  | 2.03   | 1.51  | 89.5%                | 2                 | 10                     | 8    |
|                               |                               |                  | Medium up     | 0.34            | 20                 | -0.58  | 0.81  | 1.88   | 1.76  | 89.5%                | 0                 | 11                     | 9    |
|                               |                               |                  | Low down      | 0.04            | 20                 | 0.56   | 0.68  | 2.16   | 1.40  | 85.9%                | 8                 | 10                     | 2    |
|                               |                               |                  | Medium down   | 0.36            | 20                 | 0.38   | 0.85  | 2.22   | 1.46  | 94.7%                | 6                 | 13                     | 1    |
| Supply air dumping zone (P1)  | 28                           | 26               | Fan off but AC on | 0.03            | 20                 | 1.04   | 0.65  | 0.94   | 1.57  | 77.8%                | 12                | 8                      | 0    |
|                               |                               |                  | Low down      | 0.36            | 20                 | 0.78   | 0.91  | 1.22   | 1.56  | 94.1%                | 7                 | 11                     | 2    |
|                               |                               |                  | Medium down   | 0.54            | 20                 | 0.34   | 0.77  | 2.28   | 1.43  | 100.0%               | 6                 | 11                     | 3    |
|                               |                               |                  | Medium up     | 0.32            | 21                 | 0.78   | 0.91  | 2.13   | 1.43  | 84.4%                | 10                | 11                     | 0    |
| Supply air dumping zone (P1)  | 28                           | 26               | High speed up | 0.25            | 20                 | 0.18   | 1.00  | 2.61   | 1.48  | 94.1%                | 5                 | 12                     | 3    |
|                               |                               |                  | Low speed up  | 0.26            | 20                 | 0.37   | 0.70  | 2.71   | 1.21  | 88.2%                | 8                 | 11                     | 1    |
| Perimeter zone (P3)           | 26                           | 26               | High speed up | 0.34            | 20                 | -0.01  | 0.85  | 2.64   | 1.37  | 94.1%                | 4                 | 14                     | 2    |
|                               |                               |                  | Low speed up  | 0.35            | 21                 | 0.08   | 0.73  | 2.81   | 1.24  | 100.0%               | 6                 | 15                     | –    |
| Overhead diffuser             | 26                           |                  | –              | 0.06            | 20                 | 0.05   | 0.57  | 1.39   | 1.48  | 89.9%                | 8                 | 9                      | 3    |

Note: The shaded row represents the reference condition test results. The airspeeds column shows the averaged values at 0.1, 0.6 and 1.1 m heights.
3.3.1 Preventing cold drafts in the cool condition

This study collected thermal sensation in the ‘dumping’ zone (P1) at 23°C and examined whether the fan operation could reduce the effect of the descending supply air jet. Figure 7 shows the thermal sensation and acceptability when the ambient temperature was cool (23°C). In the supply air dumping zone (in the middle column), the high-side-wall vent created a slightly cool sensation (see the grey box with the fan-off condition). The upward fan indeed reduced the dumping and

Figure 5: Density plots for thermal sensation votes (TSVs) under different test conditions. Note: Shaded areas are the perceived thermally neutral zone with $-1 < TSV < +1$.

Figure 6: Air movement preference (AP) under different test conditions.
created a neutral thermal sensation (see the dark blue box with medium up-blowing fan). It significantly increased the acceptance rate from 73.7% to 85.5%, which is close to that of the overhead diffuser set-up in a neutral condition (26°C, left column). Outside of the fan-jet (last column), the small air movement created by the fan-upward direction did not create much of a difference in thermal sensation compared with the fan-off condition, and the resulting acceptability rates are the same.

3.3.2 Extending the cooling set-point to the warmer side

One of the purported advantages of the CFIAC system is its ability to provide extra cooling via generally elevated air movement, thus allowing the cooling set-point to be raised and saving energy (Hoyt et al. 2015). To test this hypothesis, Figure 8 shows the thermal sensation and acceptability rates for different fan operation modes under the slightly warm condition (28°C). Within the ceiling-fan’s jet (in the middle column), the thermal sensation (top figure) is neutral or slightly cool at low and medium fan speeds downward direction, one sensation scale unit cooler than the fan-off condition (grey box shown in the last column). Compared with the overhead diffuser at 26°C (left column), the low downward and medium downward fan had 0 and 0.38 scale units cooler sensation. Outside of the fan-jet, the thermal sensation is slightly warm for all fan operation conditions (low, medium, downward, and upward). They are 0.29–0.73 scale units warmer than the reference overhead diffuser at 26°C, but 0.26–0.71 scale units cooler than the fan-off condition at the same 28°C ambient temperature.

The thermal acceptability rates are highest (between 94% and 100%) for all the CFIAC downward-flow test conditions (low and medium fan speeds, in- or outside of the fan-jet locations), and lower in the medium upward direction (probably due to warm sensation from less air movement). The acceptability reached 100% in the conditions of low speed-downward in the fan-jet and medium speed outside of the fan-jet. Acceptability was higher than the overhead reference condition at a lower ambient temperature required for the neutral sensation. Air movement outside of the
fan-jet could not bring thermal sensation down to the warmer-than-neutral sensation seen for the still-air reference condition at 26°C, but the higher acceptability rates for this air movement suggest that people are detecting the out-of-jet movement and appreciate it.

The very best acceptability is found to be at 28°C, and that the air-speed differences between being directly under the fan and not being under it does not matter much. Figure 9 shows that all locations, including directly under the fan, had equal or better acceptability rates than the
reference conditions. This is an important finding that supports the first paper (Chen et al. 2020): the larger variation in air speed in the room that will happen under (downward) CFIAC is acceptable for occupants.

3.3.3 Achieving similar comfort performance to that of a normal overhead diffuser at a neutral ambient temperature

_Figure 10_ compares the thermal sensation and acceptability performance of CFIAC with the conventional overhead diffuser under the neutral condition (26°C) and low downward fan speeds. Under the fan-jet, the fan created 0.3 scale units cooler thermal sensation than in the overhead reference condition. Outside of the ceiling fan-jet, the low fan speed-downward direction created similar thermal sensation compared with the overhead reference conditions. The thermal acceptability rates are higher (>94%) for the locations outside of the fan-jet than for the reference conditions, but slightly lower (89%) when people were directly under the fan-jet (probably due to the slightly cool thermal sensation caused by the air movement). Practically, these acceptabilities are virtually all the same; it can be concluded that CFIAC functioned as well as conventional VAV at low air speeds and neutral room temperatures.

3.3.4 Different types of ceiling fans for an upward direction

Different types of ceiling fan may have different airflow volume when operating upward so as to affect the space airflow distribution (Raftery et al. 2019). _Figure 10_ compares the thermal sensation and acceptability for two ceiling fans in different locations. As fan type 2 has a higher upward airflow volume, it produced higher air speeds and stronger cooling effects, especially in the perimeter zone because the airflow flowed radially outward from the fan along the ceiling and then descended along the wall. In the warm temperature of 28°C, the upward-operating fan type 2 was able to achieve a neutral thermal sensation and a high acceptance rate across the floorplate of the room.

4. DISCUSSION
4.1 POSSIBLE THERMAL COMFORT EVALUATION INDEX FOR CFIAC

_Table 4_ compares the thermal comfort evaluation indexes for the CFIAC system under different operation combinations and different measurement locations. _Figure 11_ highlights the PMV and TSV, PPD and thermal unacceptable rate, PD and thermal unacceptable rate, and _CP_{EHT} and _CP_{SET} comparisons. The PMV, PPD, and PD indexes produced differences as much as 0.86 unit scales of TSV, 21% dissatisfaction rate, and 93% dissatisfaction rate when compared with the human subject experiment results, which suggests that these three indexes are not effective to evaluate thermal comfort for the CFIAC system.
Table 4: Comparison of thermal comfort indexes.
Note: The shaded area represents the reference condition test results.
AC = air-conditioning; CP = corrective power; EHT = equivalent homogenous temperature; PMV = predicted mean vote; PD = predicted dissatisfaction due to draft; PPD = predicted percent dissatisfied; TSV = thermal sensation vote.

| TEST CONDITIONS | LOCATION | TEMPERATURE (°C) | FAN OPERATION MODE | PMV | PPD (%) | PD (%) | SET (°C) | CP VALUES BASED ON EHT (K) | CP VALUES BASED ON SET (K) |
|-----------------|----------|------------------|--------------------|-----|---------|--------|----------|---------------------------|---------------------------|
| Fan type 1      | Supply air dumping zone (P1) | 23 | Fan off but AC on | -1.00 | 73.7% | -1.20 | 33.0% | 45.6% | 21.4 | -1.68 | -1.87 |
|                 |          |                  | Medium up | -0.02 | 85.5% | -0.70 | 14.6% | 24.1% | 23.0 | -0.51 | -0.27 |
|                 |          |                  | Fan off but AC on | 0.13 | 84.2% | -0.10 | 5.3% | 33.7% | 24.4 | -1.60 | -1.93 |
|                 |          |                  | Medium up | 0.28 | 94.4% | 0.60 | 12.5% | 25.7% | 26.4 | -2.32 | -2.00 |
|                 |          |                  | Low down | 0.93 | 94.4% | 1.00 | 25.9% | 12.8% | 28.1 | -0.32 | -0.34 |
| Fan-jet zone (P1') | 26 | Low down | -0.25 | 89.5% | 0 | 5.0% | 89.6% | 24.2 | -2.08 | -2.20 |
|                 | 28 | Low down | 0.05 | 100.0% | 0.75 | 17.1% | 65.6% | 26.1 | -2.10 | -2.30 |
|                 |          |                  | Medium down | -0.33 | 94.7% | 0.53 | 10.9% | 98.4% | 25.2 | -4.10 | -3.20 |
| Outside the fan zone (P2 or P4) | 23 | Fan off but AC on | -0.32 | 89.5% | -0.38 | 9.1% | 11.0% | 23.6 | 0.23 | 0.25 |
|                 |          |                  | Medium up | -0.58 | 89.5% | -0.61 | 13.1% | 23.1% | 23.2 | -0.13 | -0.07 |
|                 | 26 | Fan off but AC on | 0.56 | 84.2% | 0.55 | 12.0% | 8.0% | 26.6 | 0.04 | 0.24 |
|                 |          |                  | Low down | 0.38 | 94.7% | 0.27 | 6.7% | 24.8% | 25.6 | -0.81 | -0.71 |
|                 | 28 | Fan off but AC on | 1.04 | 77.8% | 1.17 | 34.0% | 5.9% | 28.7 | 0.20 | 0.24 |
|                 |          |                  | Low down | 0.78 | 94.1% | 1.00 | 25% | 18.3% | 27.7 | -0.97 | -0.76 |
|                 |          |                  | Medium down | 0.34 | 100.0% | 0.88 | 21.6% | 33.1% | 26.9 | -1.41 | -1.50 |
|                 |          |                  | Medium up | 0.78 | 84.4% | 1.10 | 28.4% | 12.1% | 28.2 | -0.34 | -0.16 |
| Overhead diffuser | 26 | – | 0.05 | 89.9% | 0.50 | 10.4% | – | 26.38 | – | – | – |
Among the thermal comfort indexes shown in Table 4 and Figure 11, the $CP_{EHT}$ and $CP_{SET}$ values are consistent with each other, producing differences within $\pm 0.35$ K (mean = 0.1 K, SD = 0.29 K). The thermal sensation differences ($\Delta TSV$) between the tested cases and the baseline cases show good linear relationship ($R^2 = 0.9$) with $CP_{SET}$. This indicates that the cooling effects of the elevated air speeds and temperature distribution created by the CFIAC can be predicted using the SET model (Gagge et al. 1971). This provides possible ways to evaluate the thermal comfort performance of the CFIAC system. One approach is the SET-based cooling-effect calculation used to predict PMV in ASHRAE Standard 55-2020 (ASHRAE 2020).²

For the combination of conditions created by CFIAC, the elevated air speed limits in ASHRAE Standard 55 may provide guidance. In ASHRAE Standard 55, 0.8 m/s is the upper airspeed limit in spaces that do not provide occupant control, which is equivalent to a 3 K reduction in comfort temperature (Zhang et al. 2011). For spaces with occupant control, there is no upper air speed limit specified. In addition, ASHRAE Standard 55’s new thermal environmental control classification awards classification credits for fans under individual and group control. These credits could be relevant to CFIAC systems.

### 4.2 LOCAL BODY-PART THERMAL SENSATION

To investigate how the spatial inhomogeneity in temperature and air movement would affect thermal sensation at different body parts, Figure 12 shows the local TSVs for the face, foot, and hand.
These TSVs overlap with each other and exhibit no obvious difference. Many previous studies have reported that the downward-blowing ceiling fan will yield a higher airspeed at ankle height (Gao et al. 2017), and this may cause cold-draft discomfort complaints at the ankle level (Fanger et al. 1988). But the present results do not show a colder sensation at the foot (see the red distributions) when the fan is operating downward. This is different from what has been observed in the thermal manikin test where the downward-blowing fan produced stronger cooling effects at the foot, head, and forearm (Figure 4). The inconsistency between the physical heat transfer measured by the thermal manikin and the subjective thermal sensations suggests that thermal perception at different local body parts is a complex process. A stronger cooling effect at a certain body part may not necessarily result in a stronger subjective perception. When people feel warmth/cold at a local body part, they may compare it with other body parts, thus perceiving a different level of warmth/cold feeling than derived purely from heat loss or temperature. There is also another factor: the cool sensation in the ankles and feet depends on the type of shoes being worn and the trouser length. In the manikin test, there was a fixed gap between the bottom of the trousers and the shoes, and the manikin was wearing a slip-on canvas shoe. In the human subject tests, most people were wearing laced sports shoes with socks, which provide slightly greater insulation than the manikin's shoes.

4.3 TRANSIENT COMFORT TO CONFIRM THE DURATIONS OF THE TEST DESIGN

To examine whether the 30-min acclimatization and 20-min test periods are sufficient to allow people to reach a stable condition representing longer term conditions, Figure 13 shows the changing trends of whole-body thermal sensation during the acclimation period (Figure 13a) and during the test sessions (Figure 13b). For the acclimation period, the group average (black dots and solid regression line) shows a decreasing trend from the first to the second votes. The average of the third vote is close to that of the second vote. Therefore, overall, the 30-min period appears to provide sufficient acclimatization before performing a test session.

For the test sessions (Figure 13b), the decreasing trend mostly occurred in the cold dumping zone when the ceiling fan was turned off. There is no obvious decreasing or increasing trend in other test conditions. The 20-min duration appears to be enough to reach a stable state. Comparing the red and blue dots, Figure 13b also reveals that individual differences extend across three sensation scale units.

4.4 PRACTICAL IMPLICATIONS

CFIAC creates inhomogeneous temperature and air speed distributions across the room depending on the fan operation modes and location where the occupants sit. As shown in Figure 14, the floorplate of the CFIAC (a high-side-wall vent system, as shown in Figure 1) can be divided into the supply air-dumping zone, the fan cooling zone, the uniform zone (or outside of the ceiling fan zone), and the perimeter zone near the envelope walls.
• For the supply air dumping zone

The high-side-wall vent could ‘dump’ the cool air along the centerline of the vent if both the supply volume and supply air temperature are low, causing draft complaints. In warm ambient temperatures, this dumping effect may be comfortably dispersed by ceiling fan circulation within the room. Under cool conditions, upward-blowing ceiling fans eliminate the cold draft in the dumping zone without worsening the cool sensation in other locations. However, downward ceiling fan circulation may contribute to overcooling in the zone, and so temperatures should not be maintained below neutral.
• **For the fan cooling zone under the fan**
  It can take advantage of the cooling effects from downward-blowing ceiling fans. The magnitude of the cooling effects depends on the fan’s rotation speed. The CBE comfort tool can be used to determine the capability of elevated air speed to correcting raised ambient temperatures.

• **For the uniform zone**
  It was less affected by the vent supply air or the ceiling fan. People in this area do not feel a strong air movement and the fan-cooling effect is not strong; however, the acceptance rate is still high.

• **For the perimeter zone**
  It is similar to the uniform zone when the ceiling fan is blowing downward. When the ceiling fan operates upward, the air movement will flow downward along the walls and produce stronger cooling sensations.

### 4.5 LIMITATIONS

At present, there is no quantified evidence that people will continue to prefer elevated air speed when it occurs over longer periods than used in this study, say over day-long, multi-day, or season-long periods. There may be non-thermal comfort factors that come into play over time above certain air speed limits. Or perhaps people adapt to air speed in either restrictive or expansive directions.

Field studies in naturally ventilated buildings show that occupants rate them highly, but there are multiple possible reasons for this that may not pertain to CFIAC. The analysis of long-term comfort among fan-exposed occupancies in tropical countries would be useful, but the authors are not currently aware of published data on this. Additionally, the sample size and sample constitution (rather homogeneous age range, limited subject numbers) in the study may affect the generalizability. It is worth further investigation in future.

### 5. CONCLUSIONS

This study evaluated thermal comfort performance at different locations across the floorplate under various ambient temperatures and fan operation modes of a ceiling-fan-integrated air-conditioning (CFIAC) system with high-side-wall vent. The following findings are noteworthy:

• For neutral and slightly warm (28°C) temperatures, within the fan-jet or outside of it, the downward operating ceiling fan can create thermal acceptance above or close to that of the neutral overhead reference condition (26°C), and that of the same ambient temperature with the ceiling fan off. The optimum fan speed may vary with the ambient temperature.

• In slightly cool environments, an upward-directed fan can reduce the overcooled sensation caused by cold supply-air dumping from the side-wall vent, the worse-case scenario tested in this study, without imposing a significant cold sensation in areas elsewhere in the room.

• The standard effective temperature (SET) can predict the cooling effects of the elevated air speeds and temperature distributions produced by high-side-wall vent supplying ceiling fans, and be a useful thermal comfort evaluation tool for CFIAC systems.

### NOTES

1. The slightly cool (23°C) temperature with running ceiling fans was designed purposely to investigate the comfort effects of such a temperature combined with elevated air speeds, not to encourage overcooled operating conditions in summer.

2. The ASHRAE approach, and the SET model itself, may be accessed via the CBE/ASHRAE Thermal Comfort Tool ([http://comfort.cbe.berkeley.edu/](http://comfort.cbe.berkeley.edu/)) (Tartarini et al. 2020).
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Maohui Luo, Hui Zhang, and Edward Arens conceived the study; Luo and Zhang conducted the thermal manikin tests; Luo, Zi Wang, and Zhang conducted the human subject tests; Luo, Wenhua Chen, and Zhang conducted the data analysis; Zhang and Arens guided and interpreted the analysis; Luo drafted the manuscript; Zhang, Arens, Fred Bauman, and Paul Raftery revised the manuscript; and Zhang and Arens finalized the reviewing and approved the final manuscript.

COMPETING INTERESTS

The authors have no competing interests to declare.

ETHICAL APPROVAL

The experimental protocol was reviewed by the University of California—Berkeley’s Committee for the Protection of Human Subjects (approval number 2015-08-7882). Written informed consent was obtained before all the human subject experiments.

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SUPPLEMENTAL DATA

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