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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, which could allow raised thermostat setpoints 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 quite uniform thermal comfort perceptions. Comfort at 28 ºC was similar to that of 26 ºC 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.

POLICY 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 to the occupants in the fan-cooling-zone. To evaluate CFIAC systems, the SET model is shown to be useful and the elevated air speed method in ASHRAE Standard 55 also provides appropriate design guidance.

KEYWORDS

air movement; air distribution; energy efficient; high-side wall vent; thermal comfort

NOMENCLATURE

A	Whole-body surface area, m^2
AC	Air conditioning
AP	Air movement preference
CFIAC	Ceiling fan integrated air conditioning
CP	Corrective power, ºC
D	The fan diameter, m
EHT	Equivalent homogenous temperature
I_{clo}	Clothing insulation, clo
HVAC	Heating, Ventilation, and Air Conditioning
MET	Metabolic rate, met
NV	Naturally Ventilation
PMV	Predicted Mean Vote
Q	Heat loss, W
R^2	Correlation coefficients (r) square
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 (*2021 HVAC System Costs | Installation & Replacement Cost Estimator*). 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 system itself, which depends on 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 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, and 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 coordinately, running together or separately in different operation modes. This has several system- and comfort-related benefits. *First*, the eliminated ductwork reduces the pressure drop for the system fan, and also reduces the cost and visual constraints associated with ducts overhead in the room (Dai et al., 2021). *Second*, the room cooling setpoint can be raised, as the air movement from the ceiling fan can cool the occupants directly. Relaxing the cooling setpoint can reduce the energy demand, and system size, of the cooling plant (Ghahramani et al., 2016; Hoyt et al., 2015). *Third*, 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 air-conditioning (Wang et al., 2020; Yang et al., 2010). *Fourth*, 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 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 these 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 our previous paper (Chen et al., 2020), testing ceiling fans in a space with a high-sidewall supply vent, we characterized the velocity patterns (mean and turbulence intensity), temperature uniformity, and stratification under different fan operation modes (various fan speed levels and both airflow directions) and supply air conditions (various airflow volumes and air temperatures).

The measurement results in (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 air flow 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 speed levels 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 1.1m 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 and 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 occupant's thermal comfort. The following questions are particularly important if CFIAC is to move into real-world practice.

First, 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 'dumping' of 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 in (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.

Second, can CFIAC allow operators to raise the cooling setpoint to save HVAC energy, without decreasing comfort? The thermal corrective power 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 setpoint; above that setpoint the fans would remain at their highest practical speed at the same time as air-conditioned air is supplied to maintain the setpoint temperature. This type of operation has not been explicitly tested on subjects, to establish optimum combinations of temperatures and air speeds.

Third, 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, 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.

Fourth, 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 about 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) ("HVAC Applications," 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 inaccurate in field studies (Arens et al., 2010). ASHRAE Standard 55 (*ASHRAE Standard 55-2017*) 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, 1971), which incorporates 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 subjective evaluation of the CFIAC thermal conditions. Both downward- and upward flow directions were tested, because ceiling fans are sometimes operated in the upward direction to reduce the spatial variability and strength of airflows in the occupied zone during neutral and cool conditions.

2 METHODS

2.1 Setup of the CFIAC system

The CFIAC system was set up at the Center for the Built Environment (CBE) at University of California, Berkeley, following the same geometry that was tested in our previous study (Chen et al., 2020). The dimensions of the office-style climate chamber are 5.5 m × 5.5 m × 2.53 m (see Figure 1.a). 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 (see Figure 1 a and 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 m × 0.33 m × 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 m × 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 3-layer windows have an air gap behind their inner surfaces, through which conditioned air circulates 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 air conditioning system using an overhead supply diffuser (Accord Ventilation, 0.61 × 0.61 m) and exhaust.

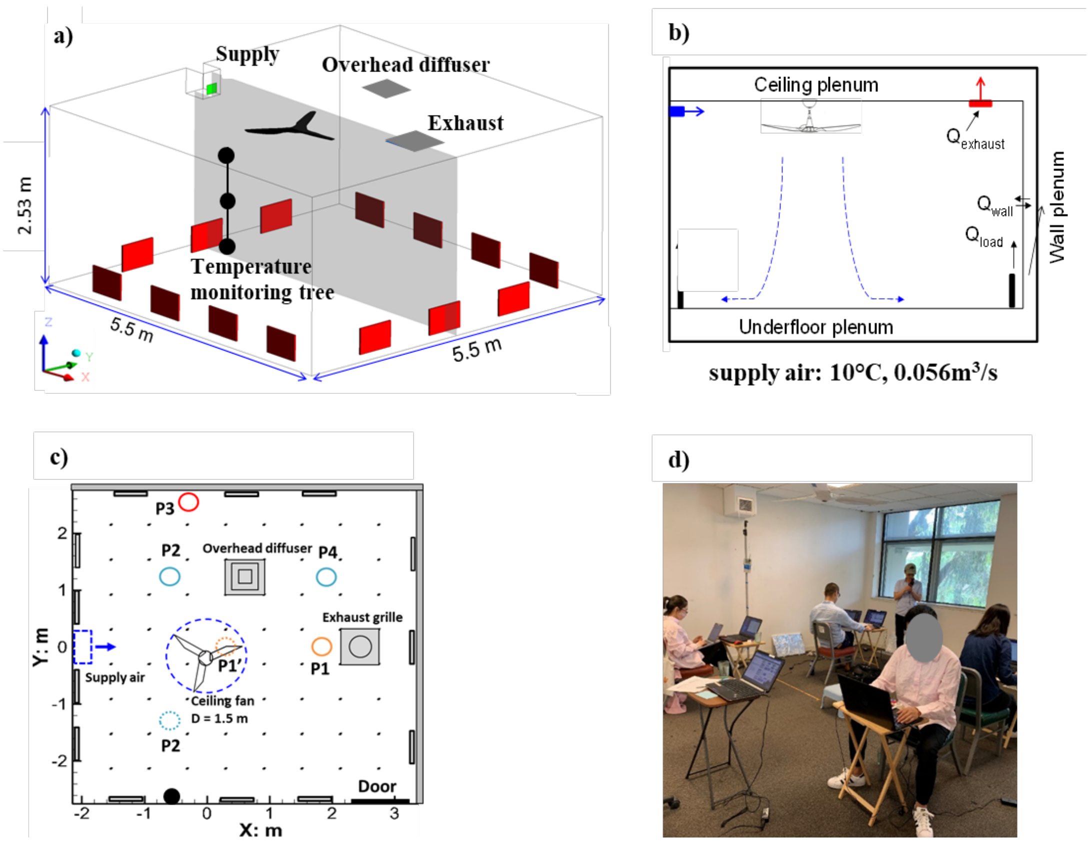

Figure 1 Configuration of the CFIAC system and the climate chamber: a) 3-D layout of the chamber room; b) Vertical section; c) Horizontal section; d) Setup for the human subject test.

The supply air temperature was set at 10 ºC and airflow volume was 0.056 m^3/s, 35% of the maximum supply air rate. The low supply flowrate and temperature were chosen because it allows 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, ~ 80 W/m^2 floor area, see Figure 1.a and c) were arrayed around the chamber perimeter 0.025 m away from the wall. Each heating panel is 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 ºC)[a], neutral (26 ºC), and slightly warm (28 ºC) conditions. When there were no subjects in the chamber, all the 14 heating panels will be turned on. When a certain number of subjects entered the chamber (1 to 5 people depending on test cases), the same number of heating panels will be turned off. The temperature was monitored under the supply air vent at three heights (0.1, 0.6, and 1.1 m, Figure 1.a) and on the Y = 0 wall at 1.1 m height (Figure 1.c). According to our previous measurements (Chen et al., 2020), CFIAC yields a highly uniform temperature distribution across the room for all the tested fan-on configurations.

Figure 1.c shows the locations where the manikin and subjects were 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 ~ 0.3 m/s and the temperature is 1 ~ 2 ºC lower than in surrounding locations. P1' is under the fan at ½ blade length where the velocity is highest; the air speed ranges from 0.6 (low fan speed downward) ~ 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 blowing downward. P3 is close to the room perimeter where test (Chen et al., 2020) showed 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 the perimeter zone location P3. Thermal comfort evaluations in locations P1 through 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

Prior to 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 of the floorplate. The test condition, CFIAC setup, and the selected

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

locations are given in Table 1. The reference case is a neutral condition at 26 ºC 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 do the tests. Under the slightly cool (23 ºC) and neutral (26 ºC) conditions, the manikin wore normal wintertime office clothing, including T-shirt, long-sleeve shirt, long pants, and socks. Its insulation was 0.65 Clo, excluding the thermal resistance provided by the mesh chair. In the warm (28 ºC) 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 *Tsk* of all manikin's segments were uniformly maintained at 34 ºC to represent a state of comfort across the body. Each test session lasted for 2 hours to allow the manikin to reach to stable state when the heat loss change of all body segments were within 3% during the last half-hour. The heat dissipation and skin temperature for each manikin segment were recorded every 1 minute. The last 10 minutes of data were used for further analysis.

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, we didn't test in location P1 because this location is similar to P2 and P4.

CFIAC setup		P1	P1'	P2	P4
High-side wall diffuser	fan off	√		√	√
	medium up (L4)	√		√	√
	low down (L2)		√	√	√
	medium down (L4)		√	√	√
Supply air from overhead diffuser with no ceiling fan with room temperature of 26 ºC		√	√	√	√
Room temperatures		23/26/28 ºC			
Manikin skin temperature		34 ºC			

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 setup, fan operation modes, and the selected locations are given in Table 2. In total, 21 test sessions were performed, with three ambient temperatures (23/26/28 ºC), four ceiling fan operation modes (low downward, medium downward, medium upward, and fan off), two types of ceiling fan (Type 1 and Type 2), and the four locations P1, P1', P2, and P3. (P4 and P2' were assumed to be identical with P2). The overhead diffuser condition without ceiling fan served as the reference case. Given that the available labour and financial resources were limited, we didn't perform all the human subject tests covering a full test matrix. Instead, we selected the test conditions 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 (23 °C, marked as '√' in Table 2) 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 speed levels vary in neutral and warm conditions explore whether the CFIAC system can extend the temperature setpoint 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. We mark 'X' for the no-fan condition and '●' for the with-fan conditions.

Fan flow direction related issues: The fan type 2 upward-operating with high-speed level (marked as '∞') aims to see whether a fan with higher upward airflow would reduce air speed variability in a room. To compare with typical HVAC air supply system, we collected the comfort evaluations of an overhead flush-mounted ceiling diffuser under neutral ambient temperature 26 °C (marked as '☼').

Table 2. Test conditions for the human-subject experiments

Ambient temperatures		**23 °C**		**26 °C**			**28 °C**		
Setups		**P1**	**P2**	**P1**	**P1'**	**P2**	**P1**	**P1'**	**P2**
	No fan	√	√	X		X	X		X
High-side wall vent	Fan type 1 medium speed upward (L4)	√	√				●		●
	Fan type 1 low speed downward (L2)				●	●		●	●
	Fan type 1 medium speed downward (L4)							●	●
	Fan type 2 high speed upward (L6)			∞ (comparing with **P3**)			∞ (comparing with **P3**)		
Overhead diffuser				☼	☼	☼			

The 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 m^3/s. Fan Type 2 has a diameter of 1.32 m and a maximum airflow under downward operation of 3.17 m^3/s.

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

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, we designed two types of questionnaires to avoid fatigue due to repetitive questions. The short questionnaire (marked in dark grey) included whole-body and facial thermal sensation votes and the acceptance votes. This streamlined questionnaire was answered at the 10th and 20th minute of the acclimation period and the 0th and 10th minute of each test session. The long questionnaire (marked in orange) was the short questionnaire plus other local thermal sensation votes for the hand, forearm, thigh, and foot, and thermal and air movement preference. It was answered at the end of each test session. A detailed description of surveyed questions can be found in Appendix A, 'Test Description' sheet.

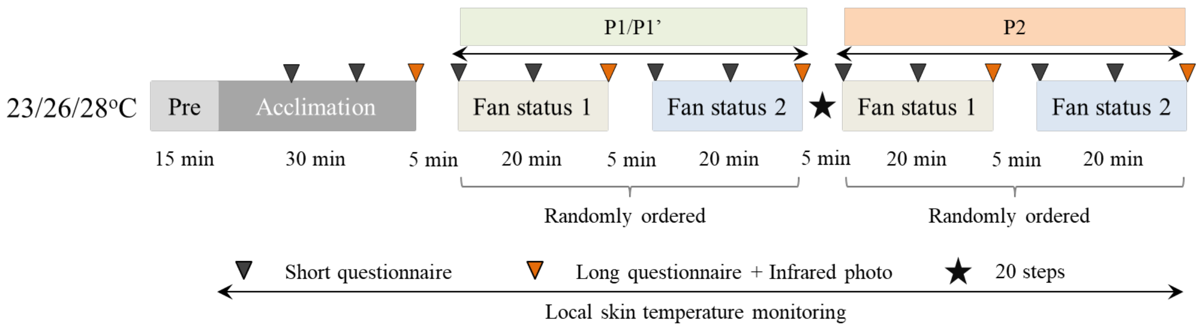

Figure 2 Test protocol example

Participants. Twenty-four 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 (5F/5M) and Asian ethnicities (7F/7M). Not all the 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-none caffeine, alcohol, smoking habits and had normal exercise intensity (2~4 times per week). Prior to 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. The experimental protocol was reviewed by University of California Berkeley's Committee for the Protection of

Human Subjects (Approval # 2015-08-7882). Written informed content was obtained prior to all the human subject experiments.

2.4 Data processing

For manikin tests, the heat dissipation and skin temperature for 16 manikin segments were exported into Excel files. The equivalent homogenous temperature (EHT) (Mcguffin et al., 2002; Wyon et al., 1989) were calculated for local body parts and for the whole-body. EHT quantifies heat loss of 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 (1.1a), where A_j is the surface area (m^2) for the segment, Tsk is the skin surface temperature (kelvin degree), Q_j is the amount of heat loss (W), and I_j represents the total clothing insulation (clo) of the segment (including the resistance provided by the air layer); 0.155 is the ratio for converting the insulation unit 'clo' to 'W/m^2'. Whole-body EHT is calculated by Equation (1.1b), in which Q, A, and I_{clo} represent the whole-body heat loss (W), surface area (m^2), and the total clothing insulation (clo) respectively.

$$EHT_j = Tsk - \frac{Q_j}{A_j} \times I_j \times 0.155 \text{ (K)} \qquad (1.1a)$$

$$EHT_{whole-body} = Tsk - \frac{Q}{A} \times I_{clo} \times 0.155 \qquad (1.1b)$$

The EHT results can be converted into corrective power (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 1.2 defines CP_j for each body part in (K), derived from EHT_j differences between a test condition and the reference condition (the overhead diffuser case). For the whole-body CP value, we can use the same equation but replace the local body parts' EHT_j with whole-body $EHT_{whole\text{-}body}$.

$$CP_{EHT_j} = EHT_j - EHT_{j\_reference} \qquad (1.2)$$

For human-subject tests, subjective voting data was first exported and stored in .txt document through the user interface of the CBE questionnaire tool. Then, it was 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 Appendix A, 'Data' sheet. The data analysis was done in the open-source language - R programming (version 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 ANOVA tests with location and temperature as 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. The 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, $0.01 < p \leqslant 0.05$ or '*' means weakly significant, and $p > 0.05$ means not significant. In figures comparing different experimental conditions, only results with statistical significance are marked on the figure. 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, the Cohen's d was calculated between groups with significant difference. Then, the Cohen's d values were marked on figures with significance test.

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, PD (predicted dissatisfaction due to draft), and SET. The CP value based on SET was defined in equation 1.3 as the difference in standard effective temperature at the still air and assumed 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, 1.1 met activity, 0.65 clo clothing insulation for 23 and 26 ºC, and 0.5 clo for 28 ºC).

$$CP_{SET} = SET_{still\ air\ and\ uniform\ temperature\ case} - SET_{measured\ air\ speed\ and\ temperature} \quad (1.3)$$

3 RESULTS

3.1 Heat transfer effects from manikin tests

Figure 3 shows the whole-body CP values at different locations under slightly cool (23 ºC), neutral (26 ºC), and slightly warm (28 ºC) ambient temperatures. In the supply air dumping zone (P1), the EHT is 1.6 ~ 2.3 ºC 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 2 ~ 5 ºC cooling effect depending on the fan speed level. 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 AC on, and identical with the condition when fan is blowing upward. The EHT differences are within 0.5 ºC. When the fan operates in downward direction, the cooling effects in these two locations are 0.5 ~ 1.5 ºC depending on the fan speed levels.

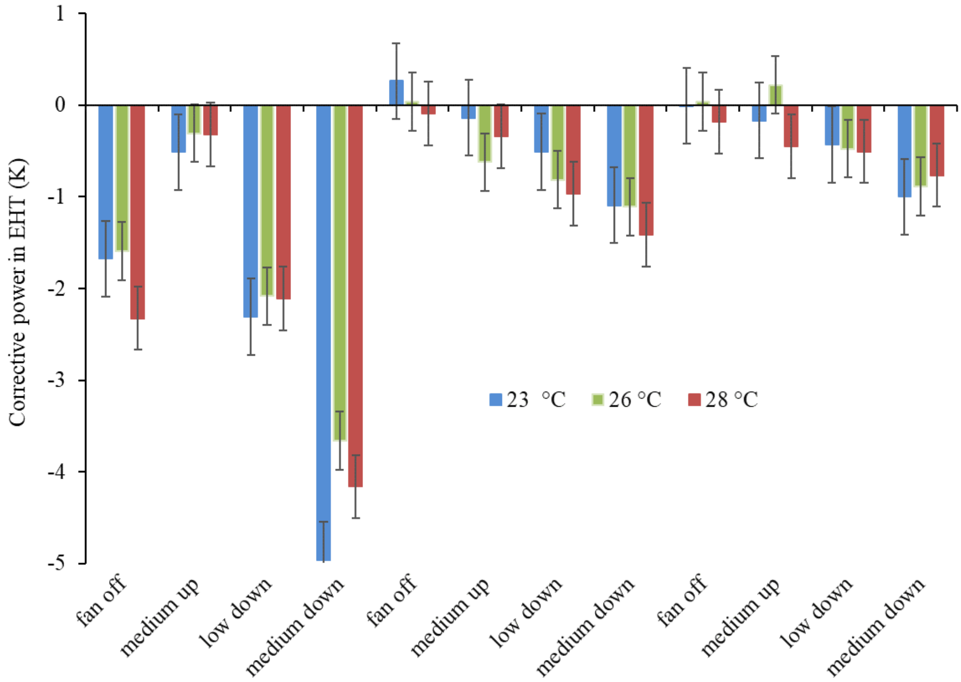

Figure 3 Whole-body corrective power in EHT at different test locations. *The reference case is the 'overhead diffuser with 26 ºC room temperature' condition. The error bars represent the standard error among the three temperature conditions.*

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, we chose the fan-off condition at each location 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 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 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 fan-off baseline condition. The strongest effects happened at the head, hand, and thigh areas, 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 2 times stronger than the whole-body average.

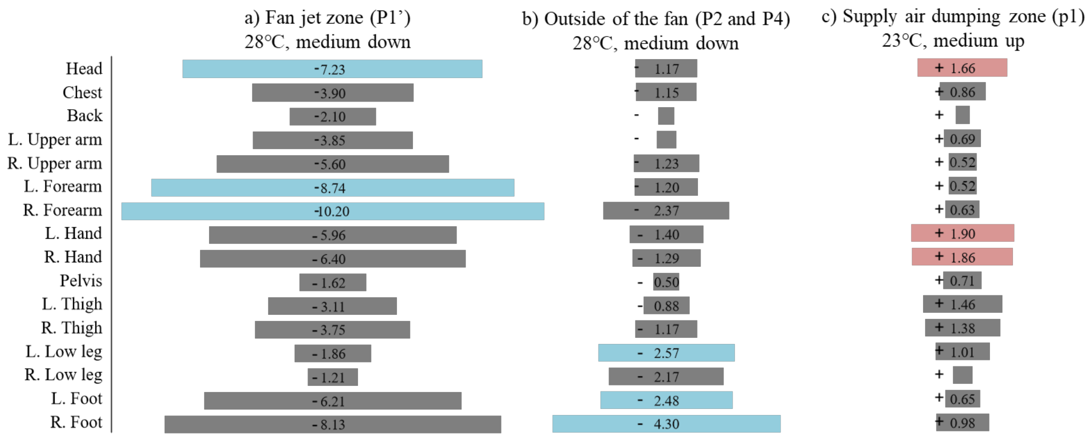

Figure 4 Corrective power in EHT at local body parts. *The baseline case is the 'fan off' condition at each location.* The '+' and '-' marks mean warmer or cooler than the baseline case, respectively. The highlighted values are the top three affected body parts.

3.2 Thermal performance from 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 fan off but AC on in the cold dumping zone when ambient temperature was cool (23 ºC), and in cases with fan off but warmer ambient temperature (28 ºC). The higher acceptance rates came from the cases with the ceiling fan on, which received similar or higher acceptance at 28 ºC than the reference conditions (overhead diffuser at 26 ºC without 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 ºC with the medium downward flow. For the cold supply-air dumping zone at 23 ºC, running the fan upward improved the acceptance by ~ 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.

Table 3. Statistical results of the human subject test

Test conditions					Whole-body thermal Sensation (TSV)			Whole-body thermal Acceptance (TAV)			Thermal preference			Airspeed preference		
	Location	Temp (°C)	Fan operation	Air speed (m/s)	Number of subjects	Mean	SD	Mean	SD	Acceptance rate	Cooler	No change	Warmer	Less	No change	More
Fan type 1	Supply air dumping zone (P1)	23	fan off but AC on	0.32	20	-1.00	0.82	1.21	2.03	73.7%	2	9	9	6	12	2
			medium up	0.24	20	-0.02	0.73	2.25	1.40	85.5%	3	10	7	3	15	2
		26	fan off but AC on	0.33	20	0.13	0.69	1.96	1.42	84.2%	3	15	2	3	14	3
		28	fan off but AC on	0.34	20	0.28	0.50	2.13	1.37	94.4%	7	13	0	2	12	6
			medium up	0.22	20	0.93	0.74	1.98	1.38	85.5%	10	10	0	0	10	10
	Fan jet zone (P1')	26	low down	0.73	20	-0.25	0.81	2.03	1.51	89.5%	2	10	8	7	10	3
		28	low down	0.72	20	0.05	0.67	2.50	1.12	100.0%	5	11	4	5	10	5
			medium down	1.05	20	-0.33	1.25	2.32	1.34	94.7%	4	12	4	9	8	3
	Outside the fan zone (P2 or P4)	23	fan off but AC on	0.03	20	-0.32	0.57	2.03	1.51	89.5%	2	11	7	0	13	7
			medium up	0.34	20	-0.58	0.81	1.88	1.76	89.5%	0	11	9	2	16	2
		26	fan off but AC on	0.04	20	0.56	0.68	2.16	1.40	85.9%	8	10	2	0	8	12
			low down	0.36	20	0.38	0.85	2.22	1.46	94.7%	6	13	1	1	13	6
		28	fan off but AC on	0.03	20	1.04	0.65	0.94	1.57	77.8%	12	8	0	0	8	12
			low down	0.36	20	0.78	0.91	1.22	1.56	94.1%	7	11	2	0	11	9
			medium down	0.54	20	0.34	0.77	2.28	1.43	100.0%	6	11	3	3	11	6
			medium up	0.32	21	0.78	0.91	2.13	1.43	84.4%	10	11	0	0	12	9
Fan type 2	Supply air dumping zone (P1)	26	high speed up	0.25	20	0.18	1.00	2.61	1.48	94.1%	5	12	3	2	13	5
		28		0.26	20	0.37	0.70	2.71	1.21	88.2%	8	11	1	2	10	8
	Perimeter zone (P3)	26	high speed up	0.34	20	-0.01	0.85	2.64	1.37	94.1%	4	14	2	2	10	8
		28		0.35	21	0.08	0.73	2.81	1.24	100.0%	6	15	---	1	14	6
Over-head diffuser	---	26	---	0.06	20	0.05	0.57	1.39	1.48	89.9%	8	9	3	2	9	9

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

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 as it reliefs the person from the cold draft. The difference is that under 23°C ambient conditions, such 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 ºC and 28 ºC to thermally neutrality (see the green distribution). The medium speed downward blowing can even correct the slightly warm temperature (28 ºC) to a slightly cool condition (see the blue distribution). For areas outside of the fan jet (third column), different fan operation modes does not produce significant changes.

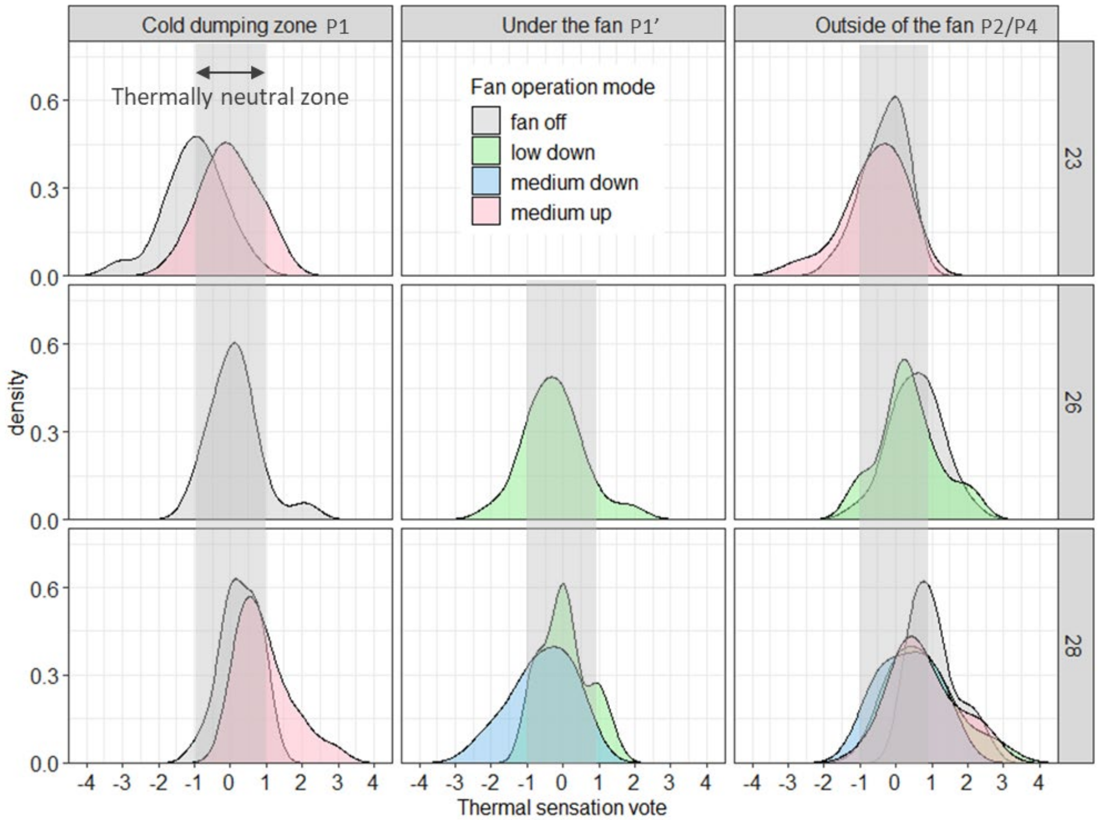

Figure 5 Density plots for thermal sensation votes under different test conditions. The shaded areas are perceived thermally neutral zone with -1 < TSV < +1.

Figure 6 shows the probability of airspeed preference for different test conditions. In the cold supply air dumping zone (first column), in 23 ºC, there were ~30% of subjects who wanted less air movement when the fan was off. Running the fan upward reduced the 'less air movement' preference by half. At 28 ºC, 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 ºC and medium-speed (1.6 m/s) at 28 ºC downward fan made 25%~45% subjects prefer less air movement. The low-speed at 28 ºC created an equal preference (25%) for both more and less, and about 50% wanted no change. For areas outside of the fan, we can see dominant preference on 'more air movement' over 'less air movement', even at 23 ºC, but this is particularly obvious at 26 and 28 ºC.

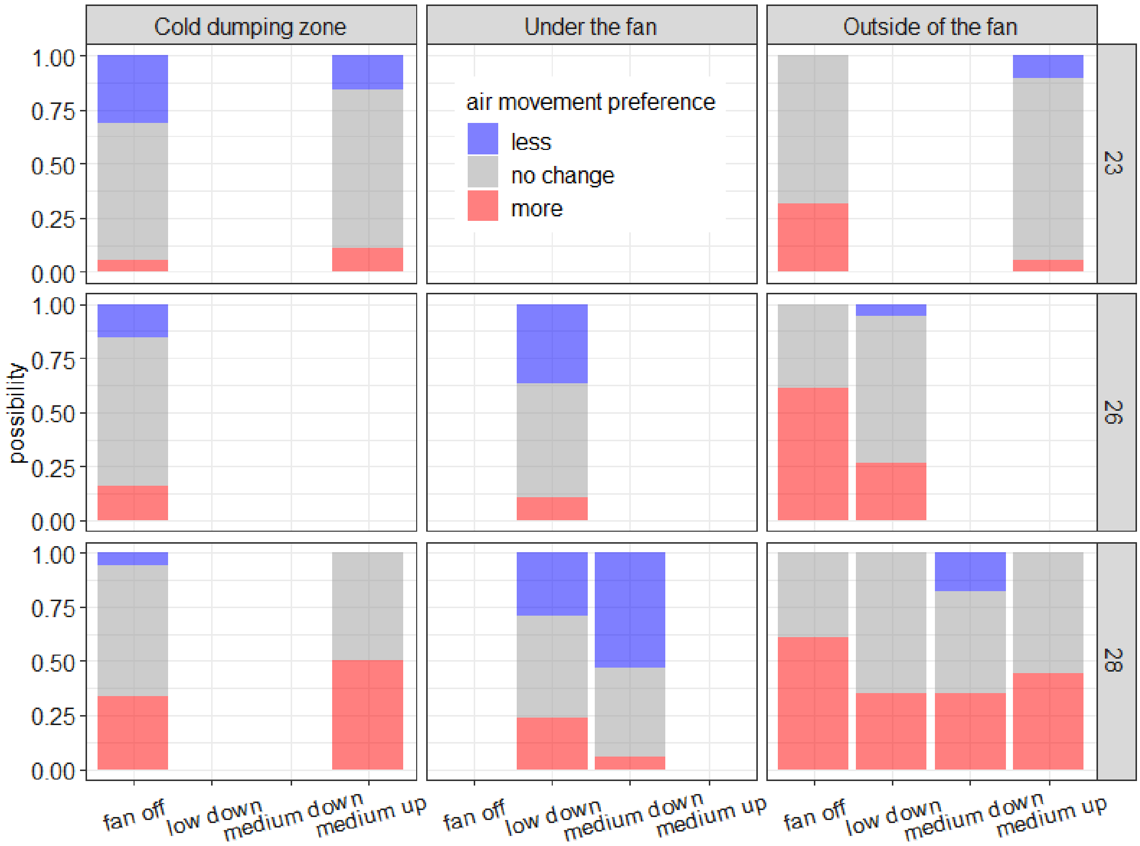

Figure 6 Air movement preference under different test conditions.

3.3 Thermal comfort effect of the CFIAC

After presenting the findings from thermal manikin and human subject experiments, the following part aims to answer the questions addressed by the experimental design.

3.3.1 Preventing cold drafts in cool condition

This study collected thermal sensation in the "dumping" zone (P1) at 23 ºC and examined whether the fan operation could reduce the effect of the descending supply air jet. Figure 8 shows the thermal sensation and acceptability when the ambient temperature was cool (23 ºC).

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 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 setup in neutral condition (26 ºC, the left column). Outside of the fan-jet (the last column), the small air movement created by the fan upward direction didn't create much difference on thermal sensation compared with the fan off condition, and the resulting acceptability rates are the same.

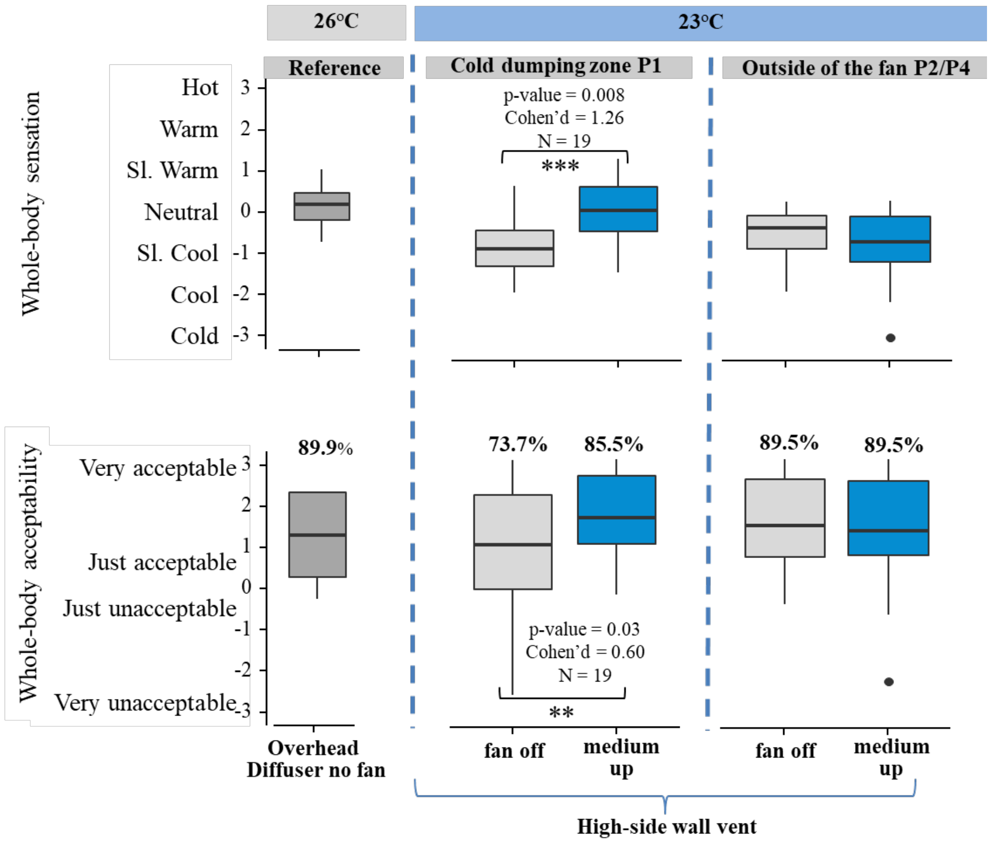

Figure 7 Thermal sensation and acceptability from the human subject tests under 23 ºC ambient temperature. The grey boxes represent results of reference conditions - the overhead diffuser and the high-side wall vent with the ceiling fan off. The blue boxes are for fans blowing upward.

3.3.2 Extending the cooling setpoint to 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 setpoint to be raised and saving energy (Hoyt et al., 2015). To test this hypothesis, Figure 9 shows thermal sensation and acceptability rates for different fan operation modes under the slightly warm condition (28 ºC). 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 ºC (left column), the low downward and medium downward fan had 0 and 0.38 scale unit 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 unit warmer than the reference overhead diffuser at 26 ºC, but 0.26-0.71 scale unit cooler than the fan-off condition at the same 28 ºC ambient temperature.

The thermal acceptability rates are highest (between 94 - 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 to 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 ºC, but the higher acceptability rates for this air movement suggests that people are the detecting out-of-jet movement and appreciate it.

We see is that the very best acceptability is at the 28 ºC temperature, 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 not unacceptable to occupants.

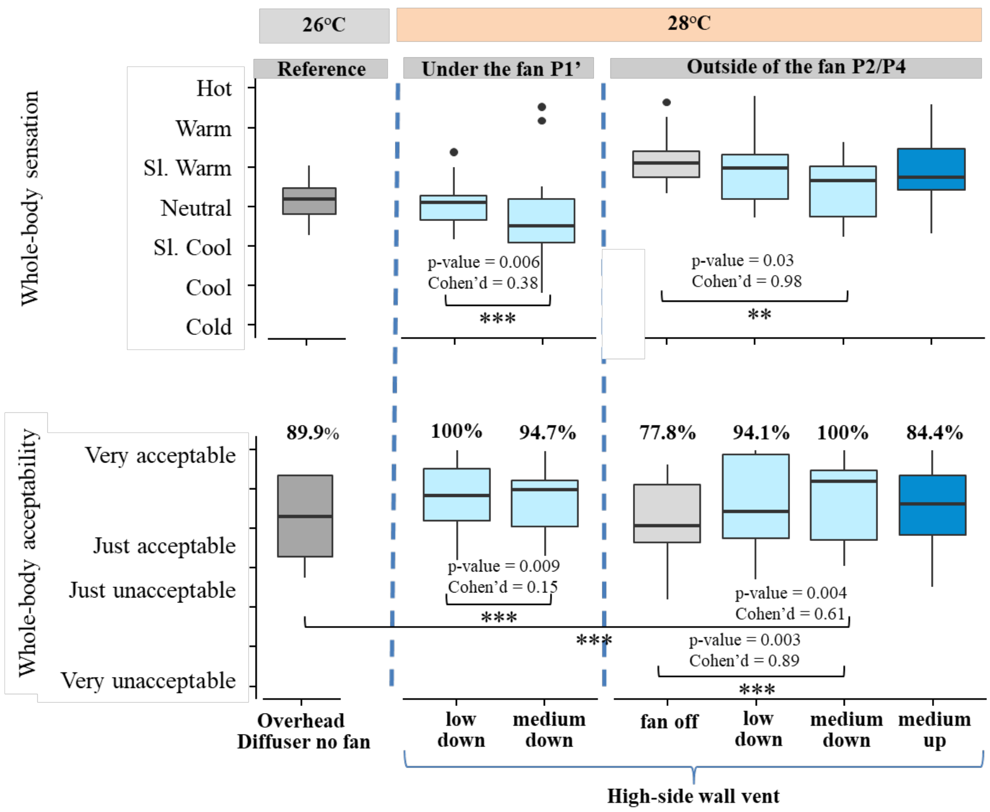

Figure 8 Thermal sensation and acceptability from the human subject tests under 28 °C ambient temperatures under different fan operation modes. The grey boxes represent results of reference condition: 26 °C with overhead diffuser (high-side-wall vent and ceiling fan off). The lighter-blue boxes show results for fan downward direction, and the darker-blue boxes for fan upward direction. The fan level (low or medium) is marked under X-axes and the two test locations (outside or under the ceiling fan-jet) are mentioned at the top of the figure.

3.3.3 Achieving similar comfort performance to that of normal overhead diffuser in 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 unit 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 (above 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; one can conclude that CFIAC functioned as well as conventional VAV at low air speeds and neutral room temperatures.

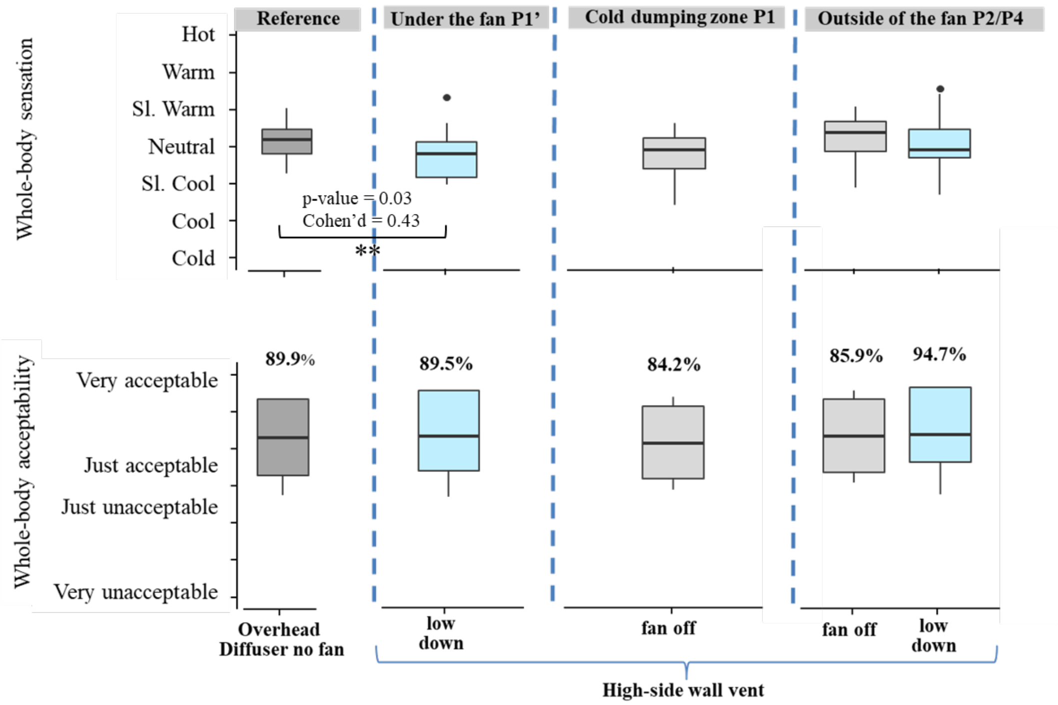

Figure 9 Thermal sensation and acceptability from the human subject tests under 26 ºC ambient temperatures under different fan operation modes.

3.3.4 Different types of ceiling fans for upward direction

Different types of ceiling fan may have different airflow volume when operating upward so that to affect the space airflow distribution (Raftery et al., 2019). Figure 11 compares the thermal sensation and acceptability for two ceiling fans in different locations. As the 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 ºC, the upward-operating fan type 2 was able to achieve neutral thermal sensation and a high acceptance rate across the floorplate of the room.

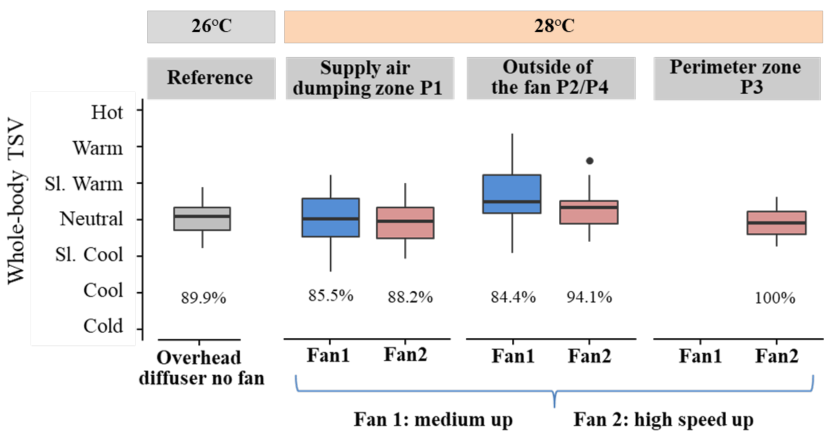

Figure 10 Thermal sensation and acceptable rate from two ceiling fans. The grey boxes on the left represent results of reference condition in neutral ambient temperature. The blue boxes show results for fan type 1 with medium upward flow rate. The pink boxes are from fan type 2 with high upward flow rate. Percent values are acceptability.

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 12 visually highlights the PMV and TSV, PPD and thermal unacceptable rate, PD and thermal unacceptable rate, CP_{EHT} and CP_{SET} comparisons. The PMV, PPD, and PD indexes produced differences as much as 0.86-unit scale of TSV, 21% dissatisfaction rate, and 93% dissatisfaction rate when comparing with the human subject experiment results, which suggests that these three indexes are not effective to evaluate thermal comfort for the CFIAC system.

Among the thermal comfort indexes shown in Table 4 and Figure 12, the CP_{EHT} and CP_{SET} values are consistent with each other, producing differences within ±0.35 K (mean = 0.1 K, SD = 0.29 K). The thermal sensation differences (ΔTSV) between the tested cases and the baseline cases show good linear relationship ($R^2 = 0.9$) with CP_{SET}. Which 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 gives us possible ways to evaluate 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 Standard 55-2020*). The ASHRAE approach, and the SET model itself, may be accessed via the CBE/ASHRAE Thermal Comfort Tool (http://comfort.cbe.berkeley.edu/) (Tartarini et al., 2020).

For the combination of conditions created by CFIAC, the elevated air speed limits in ASHRAE Standard 55 may provide guidance. In Standard 55, 0.8 m/s is the upper airspeed limit in spaces that do not provide occupant control, which is equivalent to 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.

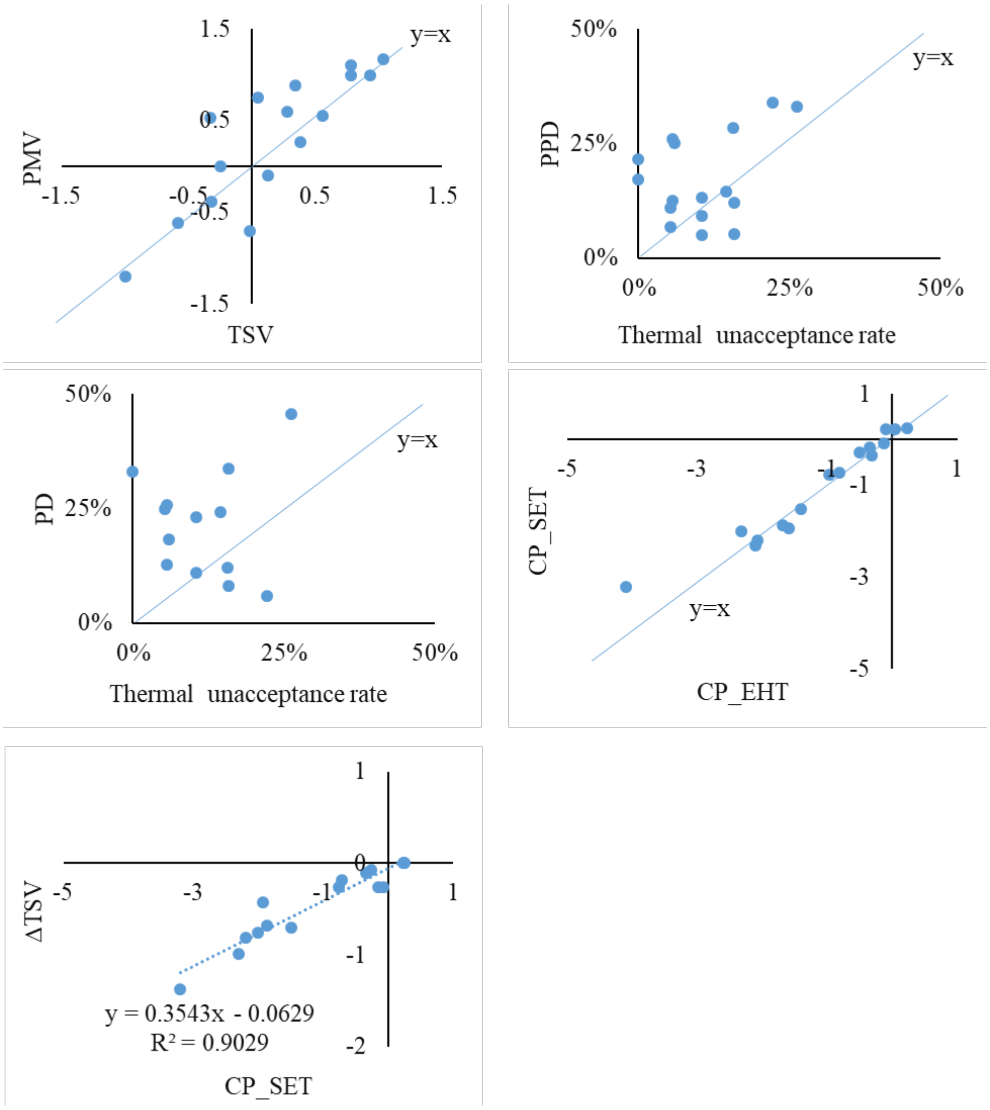

Figure 11 Comparing thermal comfort evaluation indexes

Table 4. Comparison of thermal comfort indexes

Test conditions				Whole-body thermal sensation (TSV)	Whole-body thermal acceptance rate	PMV	PPD	PD	SET (°C)	CP values based on EHT (K)	CP values based on SET (K)
	Location	Temperature (°C)	Fan operation mode								
Fan type 1	Supply air dumping zone	23	fan off but AC on	-1.00	73.7%	-1.2	33.0	45.6 %	21.4	-1.68	-1.87
			medium up	-0.02	85.5%	-0.7	14.6 %	24.1 %	23.0	-0.51	-0.27
		26	fan off	0.13	84.2%	-0.1	5.3%	33.7	24.4	-1.60	-1.93

	(P1)		but AC on					%			
		28	fan off but AC on	0.28	94.4%	0.6	12.5%	25.7%	26.4	-2.32	-2.0
			medium up	0.93	94.4%	1.0	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	-2.2
		28	low down	0.05	100.0%	0.75	17.1	65.6%	26.1	-2.1	-2.3
			medium down	-0.33	94.7%	0.53	10.9%	98.4%	25.2	-4.1	-3.2
	Outside the fan zone (P2 or P4)	23	fan off but AC on	-0.32	89.5%	-0.38	9.1%	11%	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%	8%	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%	5.9%	28.7	0.2	0.24
			low down	0.78	94.1%	1.0	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	-1.5
			medium up	0.78	84.4%	1.1	28.4%	12.1%	28.2	-0.34	-0.16
Over-head diffuser	---	26	---	0.05	89.9%	0.5	10.4%	---	26.38	---	---

Note: Shaded areas represent the reference condition test results.

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 6 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 higher airspeed at the 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 colder sensation at the foot (see the red distributions) when the fan 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 (see 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 ankle and feet depends on the type of shoes being worn, and the length of the pants. In the manikin test, there was a fixed gap between the bottom of the pants 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, which provide slightly higher insulation than the manikin's shoes.

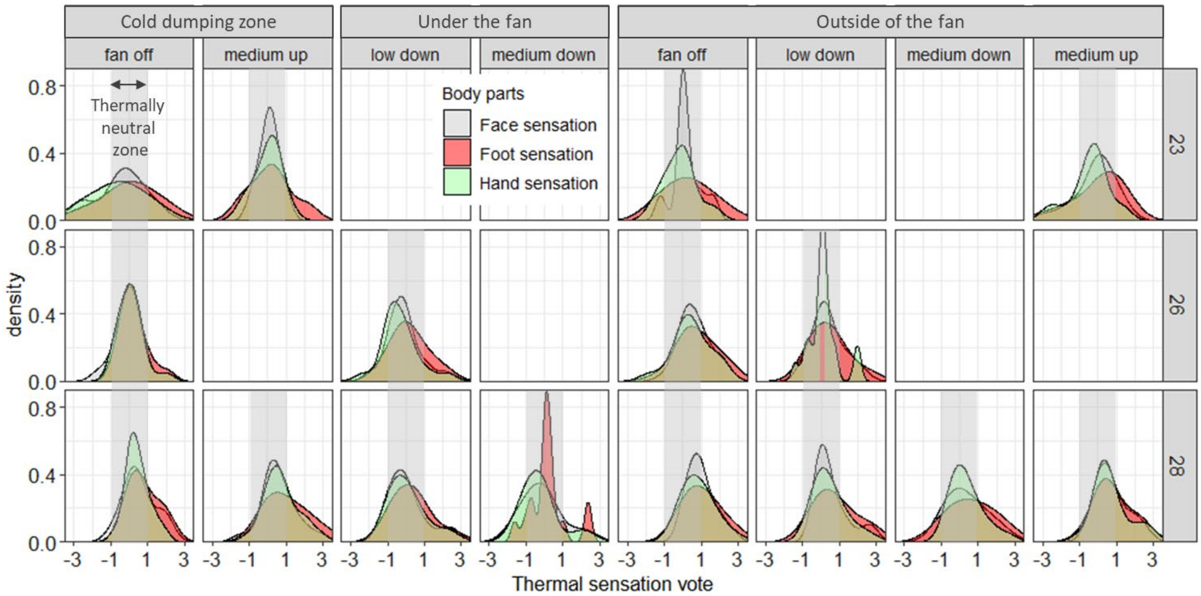

Figure 12 Local thermal sensations at the face, foot, and hand under different test conditions. The shaded areas are thermally neutral zone with -1 < TSV < +1. To make the plots readable, we dropped the local sensation at the forearm and thigh. Their distributions can be found in Appendix A, 'Extra materials' sheet.

4.3 Transient comfort to confirm durations of the test design

To examine whether the 30 min acclimatization and 20 min test periods are sufficient for allowing 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 (see Figure 13.a) and during the test sessions (see Figure 13.b). For the acclimation period, the group average (the black dots and the solid regression line) shows a decreasing trend from the first vote to the second vote. 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 13.b), 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 minutes duration appears to be enough to reach a stable state. Comparing the red and blue dots, the figure also reveals that individual differences extend across 3 sensation scale units.

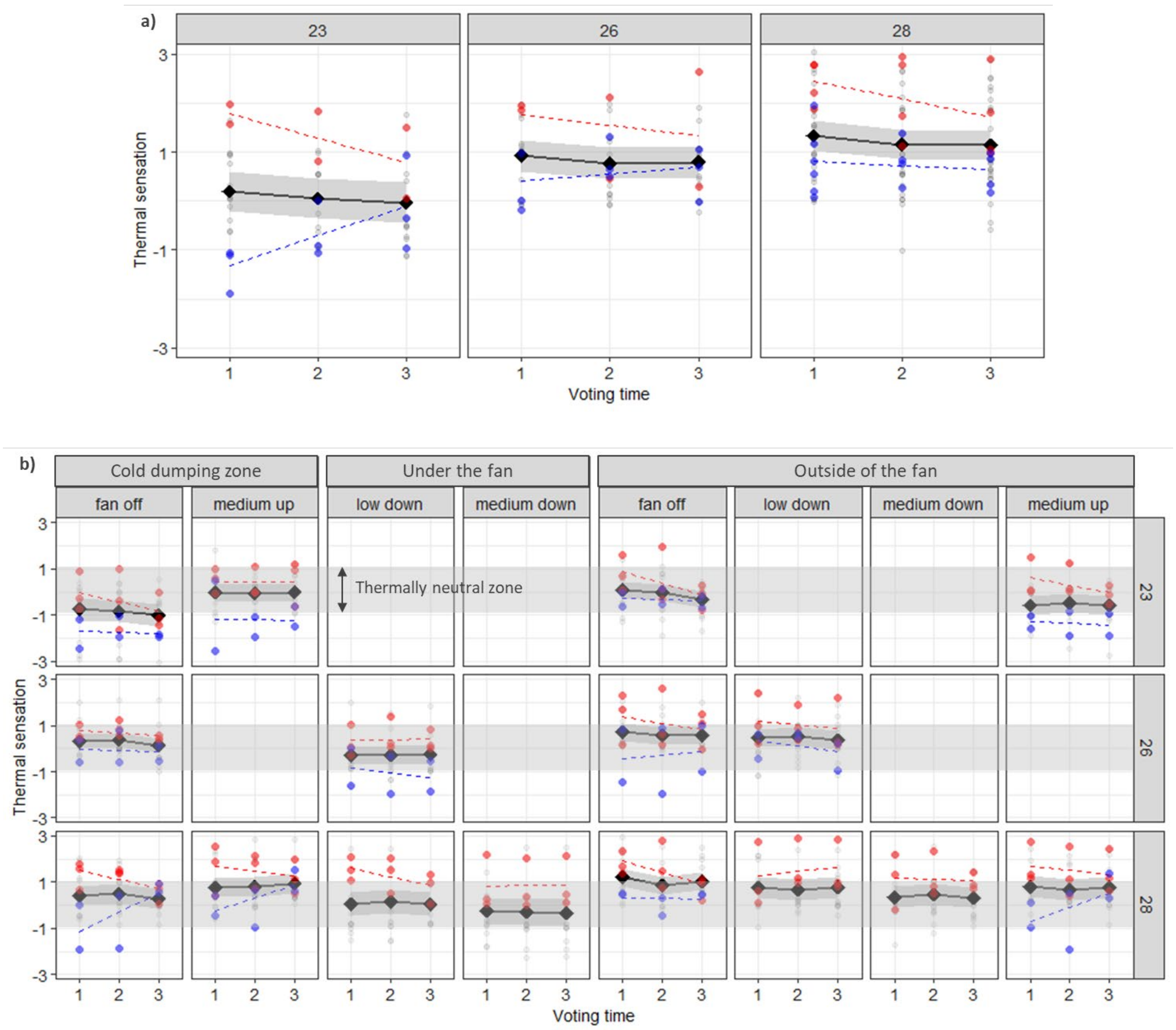

Figure 13 Whole-body thermal sensation changes under different test conditions: a) acclimation period; b) test sessions. For acclimation period, the first, second, and third vote corresponds to the 10th, 20th, and 30th minute. For test sessions, they correspond to the 0th, 10th, and 20th minute. The red dots represent the top three subjects who tend to feel warmer than others. The blue dots are the top three subjects who tend to feel colder than others. The black dots and the solid regression lines are the average TSV from all the subjects.

4.4 Practical implications

CFIAC creates inhomogeneous temperature and air speed distributions across the room depending on the fan operation modes and location where 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 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 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 don't feel 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 blowing downward. When the ceiling fan operates upward, the air movement will flow downward along the walls and produce stronger cooling sensations.

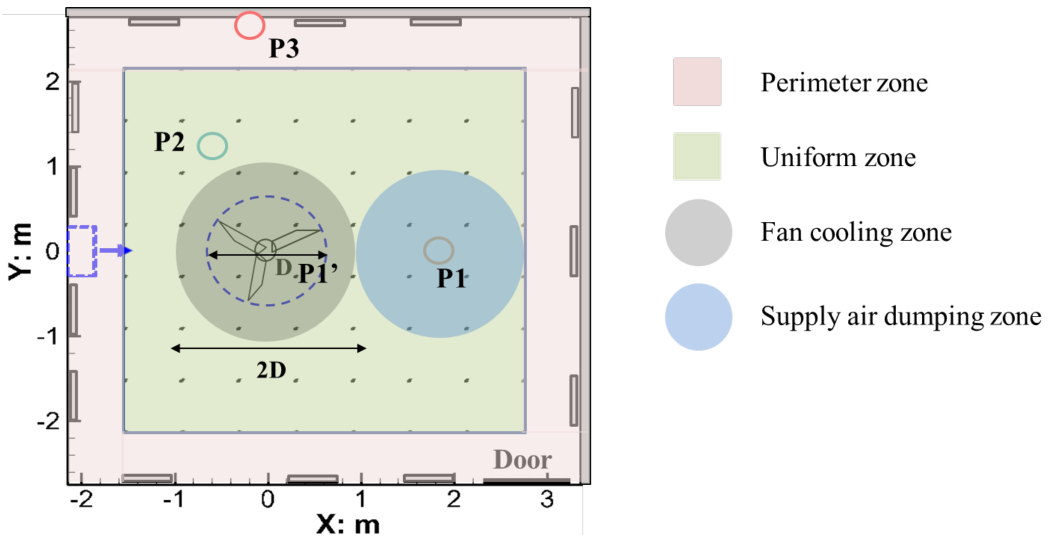

Figure 14 Locations and operation modes of the high-side wall vent system

4.5 Limitations

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

5 CONCLUSIONS

This study evaluates thermal comfort performance at different locations across the floorplate under various ambient temperatures and fan operation modes of a CFIAC system with high-side wall vent. The following findings are noteworthy.

1) For neutral and slightly warm (28 ºC) 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 ºC), and that of the same ambient temperature with the ceiling fan off. The optimum fan speed level may vary with the ambient temperature.
2) 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 significant cold sensation in areas elsewhere in the room.
3) The SET can predict cooling effects of the elevated air speeds and temperature distributions produced by a high-side wall vent supplying ceiling fans, and can be a useful thermal comfort evaluation tool for CFIAC systems.

DECLARATION of COMPETING INTEREST

We declare that no conflict of interest exits in the submission of this manuscript, and manuscript is approved by all authors for publication.

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AUTHOR CONTRIBUTION CLARIFICATION

Luo, Zhang, and Arens conceived the study; Luo and Zhang conducted the thermal manikin tests; Luo, Wang, and Zhang conducted the human subject tests; Luo, Chen, and Zhang did the data analysis; Zhang and Arens guided and interpreted the analysis; Luo drafted the manuscript; Zhang, Arens, Bauman, and Raftery revised the manuscript; Zhang and Arens finalized the reviewing and approved the final manuscript.

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