Title
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Final Report

Air movement as an energy efficient means toward occupant comfort

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Executive summary

Fans provide a major opportunity for buildings in that they enhance both energy efficiency and occupant comfort. Very low-wattage fans (3 watts) producing 1 m/s (2 mph) air movement near each occupant are capable of offsetting a 6ºF (3K) increase in indoor air temperature, while improving the comfort perceived by the occupants. Increasing the indoor temperature reduces a building's total HVAC energy about 5% per degree F, or 10% per degree C, and even more in some climate zones if the higher indoor temperature permits natural ventilation or evaporative cooling systems to be used instead of compressor-based cooling.

The opportunities for new design approaches and new fan products are considerable. Room fans have many applications in both new and retrofit designs since they do not involve other HVAC systems, can be easily turned on and off (as with occupancy sensors or wireless controls), yet they can provide an instantaneous cooling effect for the occupant. They strengthen the effectiveness of other energy-efficient measures which may be inherently slow-acting or unpredictable, such as radiant ceilings/floors, and buildings designed to rely on natural ventilation. The technologies involved are especially applicable in California climates.

The overall goal of this study is to encourage the broader adoption of room fans, especially ceiling-integrated fans, in commercial building design. The project addresses a wide range of topics ranging from fundamental human subject studies to new product design and the removal of barriers to implementation in comfort and energy performance standards. 11 subprojects were accomplished.

- Projects 1 through 4 are human-subject *laboratory tests* examining fan-related comfort at a fundamental level. Several types of fan configurations were tested. The projects establish air temperature and humidity comfort thresholds for spaces with fans, obtain subjects' preferred air speed at different ambient conditions, test for whether air movement causes dry-eye discomfort, and examine whether the comfort model for elevated air movement in ASHRAE Standard 55, which assumes that the entire body is exposed to uniform low-turbulence air movement, is applicable to realistic environments with fans. The results for each of these issues are encouraging to fan applications in building design and operation. The bottom line is that fans are shown to provide comfort up to at 28C at 80% RH and 30C at 60% RH. The fan energy per person may be very low, ranging from 2-8W.

- Projects 5 and 6 include *field studies* of naturally ventilated or mixed-mode (in which natural ventilation is combined with air-conditioning) buildings that include ceiling fans. Project 5 summarizes field studies of ceiling fan usage in one naturally ventilated building in Alameda CA, and two mixed-mode buildings in Phoenix AZ and Seattle WA. The project was part of the California Energy Commission project "Natural ventilation for energy saving in California commercial buildings" which focused on window ventilation, but this CARB project allowed us to monitor and examine the cooling performance of fans present in the three buildings.

  Results from these studies show that the Alameda and Phoenix buildings ranked in the 95th percentile of buildings in the CBE survey database, which includes 650 buildings. The Alameda building had temperatures above 80's during warm weather, and the Phoenix building upper temperature setpoint was 82F. The Seattle building ranked in the 70th percentile, with dissatisfaction coming not from overheating but from a low temperature setpoint (72F) during air conditioned periods.

  Project 6 is a field study in a school in Oakland Ca which installed ceiling fans. We surveyed teachers' responses for fan uses and tested the performance of ceiling fans at circulating warm air downwards to the occupied zone in the heating season.

- Project 7 works toward a fan evaluation index for use by manufacturers, and designers positioning fans in buildings. Currently this information is unavailable or represented by incomplete methods based on volumetric-flow-rate. ASHRAE Standard Committee TC 5.1 (Fans) has just approved a decision to develop such an index. Work done in this project should contribute to the development of the new index.

- In Project 8, our team designed two versions of an entirely new occupant-tracking fan system which approaches the maximum possible cooling efficiency. Whereas a ceiling fan provides a fixed area of cooling within the velocity cone, an occupant-tracking fan continuously cools an occupant for only 3-6W, regardless of their position or movement within the zone of the fan. Corrective power of the fan is about 3K (6F). A provisional patent has been filed by the University and discussion with manufacturers is now in progress.
- There is a potential acoustical problem with radiant ceilings, an efficient space-conditioning approach that requires thermally conductive surfaces to function. The surfaces must be hard and therefore reflect sound. The least expensive and most effective approach to absorbing sound is acoustical panels, but when they are attached to or suspended from the ceiling, they act to intercept radiant and convective heat transfer to and from the radiant ceiling. Project 9 proposed a solution, to integrate ceiling or nozzle fans into the plane of suspended acoustical ceilings. A range of designs were parametrically analyzed using computational fluid dynamics. Most of them eliminate the problem, and even add 50% to the total heat exchange achievable without acoustical panels.
- Project 10 estimates the energy savings of fan systems when used in buildings with different types of central HVAC systems. The study uses detailed EnergyPlus simulations of the DOE reference commercial building prototype positioned in representative climate zones. In California, each 5F that the indoor temperature can be raised saves between 5% (Fresno) to 7% (San Francisco) of total annual HVAC energy.
- Project 11 summarizes this project team's work on the ASHRAE Standard 55 Committee ('Thermal Environmental Conditions for Human Occupancy') to remove unwarranted barriers to fan use. A great deal of our work has just been implemented in the new code-compliant ANSI/ASHRAE Standard 55-2013, published in December. Air speed corrections suitable for fans are added to both the PMV and Adaptive Models within the standard. We developed (with concurrent funding) a new openly available web tool for calculating and visualizing the provisions of the standard, including its airspeed algorithms. It will be adopted as the official comfort tool. We are also on the team developing the official Standard 55 Users' Guide, which will contain instructions useful to fan implementation in buildings.

In overview, the results from these projects are very promising. They show that acceptable office comfort can be provided by fans drawing 2 - 8 watts per person, in a wide range of interior temperatures up to 30C (86F) and 60% RH (relative humidity). Fans can be used to augment the performance of other energy-efficient technologies, such as radiant cooled ceilings and natural ventilation. The computer models used for evaluating air movement comfort effects are found to give robust predictions even though they simplify the complexity of actual air flows produced by fans. Computer control over fan speed and direction will enable a new generation of extremely efficient and effective devices for providing air movement cooling within buildings.

The HVAC energy saving possible from increased fan use is very substantial, exceeding that of almost any other current technological improvement to buildings.

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1 Introduction and background

1.1 Statement of Significance

Compressor-based cooling is the second largest consumer of electricity in US commercial buildings, exceeded only by electrical lighting (481 trillion BTU vs. 1340 trillion BTU) (EIA 2009). In office buildings, the single largest energy user in California's commercial sector and the scope of this study, compressor-based cooling constitutes roughly 15% of total electricity consumption, and 5% of that of the State as a whole. In addition, the California Energy Commission has identified maintaining adequate electricity supplies as the state's most pressing challenge; since air conditioning is the largest contributor to buildings' peak demand (Brown and Koomey 2003) it is a prime target for demand reduction strategies.

The amount of energy used to condition commercial buildings is increased by the tendency of building managers to operate buildings at too-low ambient temperatures during the warm season (Mendell and Mirer 2009). The overcooling has been found to significantly reduce occupant comfort and even health symptoms (Moezzi and Goins 2011). Overcooling has a number of causes, but an important one is the unavailability in conventional sealed office buildings of sufficient air movement around their occupants (Hoyt 2009a, Zhang et al. 2007). Sealed designs with their low air movement consistently show lower occupant satisfaction than offices with operable windows (Fisk et al.1993, Brager and Baker 2009). Although buildings with operable windows tend to have warmer interior temperatures than sealed buildings, the modest increase in indoor air movement that they provide indoors gives them comfort ratings superior to those of sealed buildings.

This suggests one can reduce cooling energy demand by allowing a building to float within an expanded indoor temperature range while maintaining the occupants' thermal comfort by increasing air movement in their vicinity. Fans of very low wattage (as low as 3W) have been shown to yield the equivalent of 3K (6ºF) offset of air temperature within an individual workstation (Zhang et al. 2009). Buildings employing such fan cooling promise substantial savings in their HVAC energy, more than 30% below that of conventionally conditioned buildings (Hoyt et al. 2009b). The energy savings may be even greater if the warmer setpoint temperatures enable the primary cooling source to be switched from a compressor-based system to one of the more efficient but lower-power approaches, such as natural ventilation, evaporative cooling, or radiant ceiling/floor systems. Such technologies are especially applicable in California climates. Room fans may be readily applied in both new and retrofit designs since they only require reprogramming HVAC system setpoints.

Several large field studies in offices buildings showed that occupants of air conditioned buildings had significantly more sick building symptoms than occupants of naturally ventilated buildings (Fisk et al. 1993, Burge et al. 1987). The increased sick building syndrome symptoms were found to be associated with poorly maintained HVAC systems (Mendell et al., 2009, Bennett et al. 2012), some due to microbiological contamination of the supply air from chillers and humidifiers (Burge et al. 1987).

For HVAC buildings, several studies show that increasing fresh air supply by increasing ventilation rate reduces SBS symptoms (Sundell et al. 2011, Mendell and Heath 2005). Reviewing studies in schools, Sundell et al. (2011) pointed out that low ventilation rates are associated with increased absenteeism and more respiratory symptoms in school children, but that the available data are too limited for firm conclusions.

Since the1960's, the use of air movement to maintain thermal comfort in warm conditions was impeded by strict limits to air movement in thermal comfort standards (e.g. ASHRAE 55-1992, 2004). This has led to almost no innovation in designing interiors that use air movement, and only modest innovation in industrial products that move air. Recently, based on extensive field and laboratory studies, the air movement requirements in these standards were revised to permit higher indoor air speeds (ASHRAE 55-2010, Arens 2009). The revisions pave the way for the abovementioned 30% reduction in HVAC to become reality, and also for significant reduction in their peak power demand. They also enable more individual environmental control and the higher levels of occupant comfort that are already observed in buildings with operable windows.

The challenge is how to implement indoor air movement devices within ceiling and interior space, so that they are: highly energy efficient, comfortable and acceptable to occupants, visually attractive to building management and designers, and straightforward to design. This research project addresses each of these issues with respect to air movement systems that serve both individuals and groups of occupants. The results and intellectual property from this project are intended to inform the HVAC and architectural design communities, and the building-interiors, furniture, and appliance industries, helping them develop new designs and products. We have also worked to have the project's results incorporated into the new code-compliant version of ANSI/ASHRAE Standard 55-2013, in order to facilitate their adoption by LEED and Title 24.

1.2 Background

A great deal of the knowledge about occupant air movement preferences comes from analysis of the ASHRAE database of indoor thermal comfort field studies (de Dear 1998) and a few subsequent studies, which together relate physical measurements and occupant comfort votes obtained in a large number of buildings. Most of the air speeds measured in these studies were low; air velocities in mechanically ventilated buildings (HVAC) are two- to three times lower than in buildings with natural ventilation (NV) where the average itself was only 0.3 m/s (60 fpm). However, this average level of NV air speed produced the equivalent of 2K (4ºF) cooling. The air movement recorded in the database was primarily due to flow through windows, and therefore was probably less than optimal in warm conditions.

At the higher end of indoor temperatures, the acceptable temperature range can be expanded by cooling the occupants convectively with fans, from above (ceiling fans) or the side or front (stand and desktop fans, as well as local air jets from personal comfort systems (PCS). The combination of NV and fans is very energy-efficient. The fans act as backup to less-predictable wind-driven flows through windows, and in California climates with high daily temperature range, they allow the windows to be closed during the hotter afternoons while maintaining convective cooling indoors. Certainly, most responses in the ASHRAE database indicate a preference of more air movement, with very few wanting less (Paliaga et al. 2004, Zhang et al. 2007, Hoyt et al. 2009, Toftum 2004).

Air movement has long been shown to be effective at increasing convective (and evaporative if sweating occurs) heat loss in warm environments (e.g. McIntyre 1978, Rohles et al. 1983, Tanabe et al. 1987, Kimura et al. 1993). The Center for the Built Environment (CBE) has a long history of studying air movement for comfort in both laboratory and field studies. Early laboratory studies showed that comfort can be well-maintained by air movement in ambient temperatures as high as 27.8ºC (82ºF) and 30ºC (86ºF) (studies summarized in Fountain et al. 1994, Arens et al. 1995, Arens et al. 1997). Recent laboratory studies have shown that a 3W personal fan maintains a neutral thermal sensation at 28ºC (Zhang et al. 2010, Zhang et al. 2009). These studies also showed that air movement significantly improved people's perceived air quality, possibly by disrupting the body's naturally-occurring thermal plume through which body odors and skin bioeffluents are carried to the breathing zone. Field studies have shown that personally controlled air movement systems have been highly popular in offices (Bauman et al 1998).

Another argument for shifting the comfort strategy in commercial buildings is found in Mendell and Mirer (2009). Based on their analysis of data from the US EPA Building Assessment Survey and Evaluation Study (BASE) involving 100 office buildings, they suggested that reducing the amount of summer cooling would benefit occupant health symptoms and thermal comfort while also saving energy.

Altogether, the cited research proves that the air movement can increase thermal comfort and perceived air quality, decrease occupant symptoms and reduce building energy consumption. However, studies for hot and humid environments (29C or higher and 80% RH) have only been done with wind over the whole-body from a box fan (Kubo 21997, Tanabe 1988), not from ceiling or floor fans which only provide wind on part of the body. Are the differences between such fans, or the wind exposure they provide, significant? There is also no information available to examine human thermal comfort under oscillating fans, in which the air movement experienced by the occupants cycles over time. Better comfort information is needed for: vertical (ceiling fan) vs. horizontal (floor or desk fans) air movement in warm and humid environments; static vs. oscillating fan; modeling comfort using the whole-body model vs. the reality that wind produced by fans only reaches part of the body which can be nude or clothed.

In addition, there are at present there are very few examples of efficient air movement implementation in ceiling fan design. There are also no evaluation metrics that focus on a fan's capacity to provide comfort. Fan performance is quantified in terms of volumetric flow capacity (e.g., cubic feet per minute) which is not directly related to the quantities that determine comfort or the efficiency of the fan: airspeed, width of jet, turbulence intensity, and turbulent spectral distribution at various speed levels and distances from the fan, and the electrical power required for the different fan speed levels. In addition, since furniture such as desk/table surfaces are ubiquitous in the work environment, the fan airflow redirection due to such surfaces should be accounted for in a fan performance index.

1.3 Objectives of Project

- Quantify occupant comfort and satisfaction in these systems under exposure to air movement and elevated temperatures, and characterize occupants' use of personal environmental controls.

- Characterize the air movement patterns and cooling effects of integrated fan systems through physical measurements in laboratory test installations

- Design efficient ways of integrating ceiling fans and nozzles into ceiling systems, and develop efficient ways to deliver the airspeed to the occupant directly, and not to unoccupied space.

- Investigate, through CFD modeling, the impacts of fans into acoustical ceiling panels on radiant slab systems; verify if the improved convective heat exchange created by the fans can offset the reduced radiation heat exchange caused by the presence of the acoustic panels.

- Using EnergyPlus, assess the energy saving potential of shifting the building operation strategy from strict temperature conditioning to higher indoor temperatures and increased air movement by simulations of a selected example building located in a range of California climates.

- Investigate a future fan evaluation method for use by manufacturers and designers of interior environments.

- Work with ASHRAE Standard 55 Standing Project Committee to codify the new information on fan cooling produced by this project. Provide material and tools to assist designers of air movement cooling systems.

2 Human comfort under various airflows – laboratory studies

2.1 Introduction

This section describes 4 human subject studies conducted to improve fundamental understanding of human thermal comfort in the presence of airflow provided by fans. Both horizontal and vertical air flows were tested and compared. The studies address the effects of partial exposure of the body to the airstream, of various forms of time-varying airflow such as turbulence intensity and intermittent flows, of personal control options, of possible non-thermal sources of discomfort such as eye dryness, and of combinations of temperature and humidity that are at or above the comfort zone boundaries prescribed for elevated airspeeds by ASHRAE Standard 55.

The test conditions for the four studies are shown in Figure 1. They are all outside the warm boundary of the traditional still-air thermal comfort zone (shaded zones in Figure 1), but are with the exception of the 86F/80% RH condition within or on the comfort boundary for elevated air speed (dotted line in Fig 1). Although the humidities are on the high side for California, there have been previous studies of fan performance under conditions. We intended to extend the envelope of previous studies in this work, and at higher humidities the comfort limits are most severely tested.

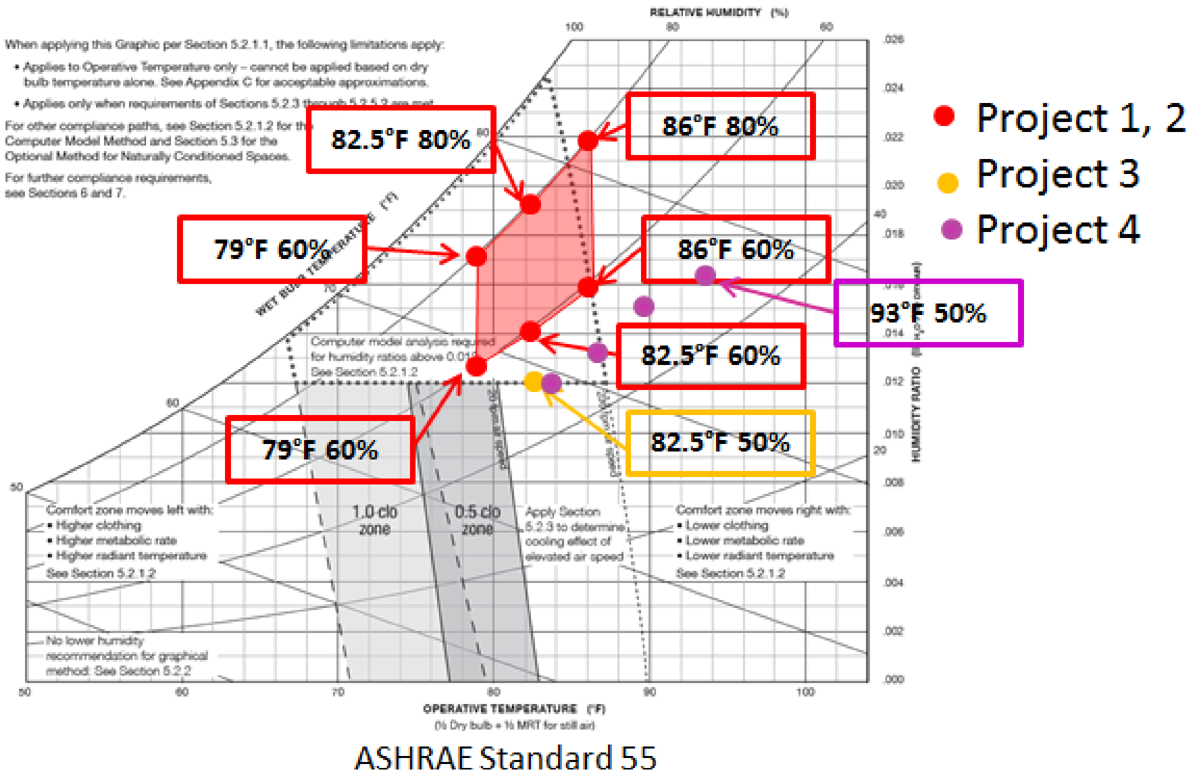


Figure 1 Air temperature and humidity conditions for the four human subject tests described below

2.2 Fan Selection for Testing

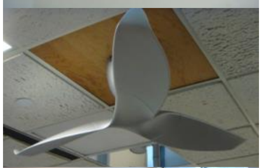
We considered it essential that the fans tested had the characteristics necessary for successful adoption in commercial buildings. The fans need to be: effective at producing comfortable air movement, energy efficient, visually attractive, and acoustically silent.

Efforts were made to find the best fans available on the US and Japanese markets. We identified 4 fans (Table 1) with exceptional performance in some or all of the four characteristics.

The three propeller fans all employ highly efficient DC motors, magnetically levitated bearings, and have great air moving ability. Two of the three fans won the EnergyStar most efficient fans award in 2013. The DC motor and the low wattage of these fans provide an additional potential benefit: they could be operated under battery power during demand response events.

The Dyson fan has no visible moving parts, which opens numerous aesthetic opportunities for interior designers. It is inherently somewhat less energy-efficient because of the larger pressure required to induce airflow through its induction ring. The impeller also generates more noise.

Table 1 Selected 4 fans to consider for the study

manufacture	Photos	Wattage (W) for normal range of use (maximum)	Visually attractive	acoustical
Big Ass Fans, US		2 – 8 (max. 18)	good	Quiet
Aeratron Inc., Australia		2 – 8 (max 14.5)	good	Quiet
Toshiba, Japan		2 – 8 (max 16)	good	Quiet
Dyson, UK		9 – 30 (max 35)	Special –without propeller blades	Noisy, even at low speed levels

We decided to use the Haiku and Toshiba fans for the human subject testing. The Haiku blades are beautiful and have achieved commercial success and recognition in high-end commercial and residential design. The Toshiba fan blades are smaller but transparent and so are not visible especially when in motion. The Toshiba was tested as a floor fan (as intended) but we also modified it to be mounted on the ceiling as an oscillating vertical source. In this mode, it provided intermittent airspeed at multiple occupants' locations.

The Aeratron fan was not commercially available in the US at the time of the study. We found it very effective and we encouraged the manufacturer to expand its market. (It is now prominently available at Home Depot as 'Hampton Bay Aeratron', in two- and three-bladed models).

The Dyson fan though effective and reasonably efficient was judged too noisy for use in human subject tests. The noise could be significantly reduced by putting more distance between the motor/impeller and the discharge ring, say by a duct above the suspended ceiling, but this involved a level of customization that might not occur in actual future building designs. We did not wish our findings to be tied to a design concept that might not be available in the future.

2.3 Project 1: Human comfort and perceived air quality in hot and humid conditions with vertical airflow (ceiling fans)

(This paper is being submitted to the journal *Building and Environment*)

Abstract
The effects of air movement from ceiling fans on subjective thermal comfort and perceived air quality (PAQ) were examined for warm-humid environments. In a climate chamber controlled at elevated temperatures (26°C, 28°C and 30°C) and relative humidities (RH 60% and 80%), sixteen subjects (8 males and 8 females) were exposed to air speeds ranging from 0.05m/s to 1.8m/s provided by ceiling fans. Subjects voted their thermal sensation, comfort, PAQ, air-movement acceptability, humidity sensation, eye-dryness and noise satisfaction during 2-hour long tests. Air movement is seen to significantly improve the subjects' thermal comfort, PAQ, and acceptance of environmental air movement and humidity, without causing discomfort from eye-dryness or noise. When the ceiling fans were off, subjects were comfortable at 26°C/80% and 28°C/60%RH. With air movement, subjects were comfortable up to 30°C/80%RH, which is beyond the current ASHRAE summer comfort temperature and RH limits. The preferred air speeds for ceiling fans were in many cases higher than the airspeed limits specified in ASHRAE.

Keywords: Warm-humid environment, Air movement, Ceiling fan, Thermal comfort, Perceived air quality

2.3.1 Background

The ceiling fan is a common cooling strategy around the world, especially in hot-humid South and Southeast Asia. As an alternative to air conditioning, ceiling fans save energy by allowing the thermostat to be set at a higher temperature, and by allowing an extended naturally ventilated season when air conditioning is not needed. Simulations for hot-humid Florida have shown that raising the thermostat cooling setpoint 2°C while using ceiling fans saves approximately 15% of cooling energy use while providing occupants the same level of comfort (James et al. 1996).

Research into the effects of ceiling fans on thermal comfort was first instigated by the oil crisis in the 1970's. Studies conducted in Australia (Burton et al. 1975), UK (McIntyre 1978), and US (Rohles et al. 1983, Scheatzle et al. 1989) investigated the effect of air movement from ceiling fans on subjective thermal comfort in warm environments. It was found that an air speed of 0.8m/s to 1.05m/s could maintain comfort between 28°C and 29.5°C at 50% relative humidity.

Less attention has been paid to warm-humid environments. A recent Chinese study of ceiling fan on comfort at high temperature and humidity found that an upper acceptable air speed 0.8m/s kept occupants comfortable at 29.3°C and 70% RH (Ji et al. 2003).

This study tests the ability of a very low-energy ceiling fan to produce comfort for occupants under warm and humid conditions, probing the acceptable limits of temperature, humidity, and air speed. Its objectives were to determine:

1. the effects of air movement from ceiling fans on thermal sensation, comfort and perceived air quality.
2. subjects' acceptance of humidity, air movement, eye-dryness and noise, with and without air movement
3. the comfortable temperature and humidity limits with and without air movement, compared to comfort zones in standards

2.3.2 Method

2.3.2.1 Subjects

Sixteen paid volunteers (8 males and 8 females) participated in the experiments (Table 2). During the tests they were instructed to wear summer clothes (0.5 clo), a T-shirt or short sleeve shirts, jeans or light pants, underwear, light socks, and sandals.

The study was approved by the University of California, Berkeley Committee for the Protection of Human Subjects. All participants provided written informed consent before the tests.

Table 2 Subjects' Anthropological data

Sex	*Sample size*	*Age*	*Height (m)*	*Weight (Kg)*	*BMI*
Female	8	29.5±4.6	1.63±0.46	54.4±6.6	20.4±1.7
Male	8	27.8±4.9	1.76±0.65	71.2±6.8	22.9±1.8
Female + Male	16	28.7±4.6	1.70±0.86	62.8±10.8	21.6±2.2

2.3.2.2 Test facilities

The experiments were conducted in a climate chamber measuring 5.5m*5.5m*2.5m (Figure 2), located at Center for the Built Environment, University of California at Berkeley. The chamber's temperature control accuracy is ±0.5°C, and humidity ±3% RH. The fresh air flow rate was around 180-220 cfm – 85-104 L/s. Since the number of occupants was 3 in the chamber (2 subjects and 1 researcher), the minimum outdoor air supply was 28 l/s to 35 l/s per person, much higher than the 2.5 l/s person requirement of the current standard for office buildings (ASHRAE 62.1 2004).

Two ceiling fans installed in the chamber provided two spots with similar air speed under two fans where two subjects were located (Figure 2). The subjects were tested in groups of two, experiencing almost the same temperature, humidity and air speed conditions throughout the two-hour sessions.

The ceiling fan has 6 speed settings controlled via a remote controller. The energy consumption ranges from 2W at speed level 1to 16 W at level 6.

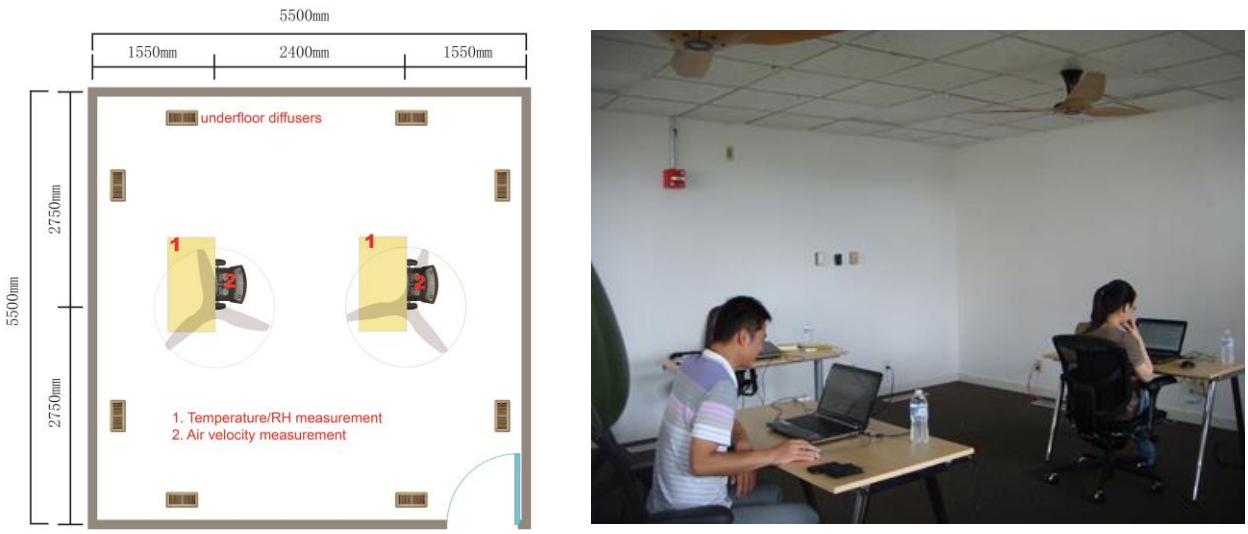


Figure 2 Test climate chamber configuration

2.3.2.3 Test conditions

The test conditions were selected to represent typical temperature and humidity levels in free-running buildings in a warm-humid climate, using the ASHRAE RP 884 database and extensive field data from naturally ventilated building in Guangzhou (project 1 in Figure 1). Three temperatures (26°C, 28°C and 30°C) and two RH (60% and 80%) were chosen. Four levels of air speeds were tested in each test condition, in a randomized sequence (Table 3 Test conditions

Each test condition is represented by the metric effective temperature (ET*). ET* is the air temperature with relative humidity at 50%, in which people would have the same heat loss as in the real environment. Since the test conditions in this study are at RH levels 60% and 80%, there are for each air temperature two corresponding ET* values, each higher than the air temperature (Table 3).

Table 3 Test conditions

Test conditions	T (°C) and RH (%)	ET* (°C)	Air speed (m/s)						
			No fan	0.30	0.70	0.85	1.20	1.60	1.80
A	26°C, 60%	26.3	X	X	X	X			
B	26°C, 80%	27.0	X	X	X	X			
C	28°C, 60%	28.5	X		X	X	X		
D	28°C, 80%	29.7	X			X	X	X	
E	30°C, 60%	30.7	X			X	X	X	
F	30°C, 80%	32.6	X				X	X	X

2.3.2.4 Survey questions

The survey questionnaires were designed to cover acceptance of thermal environment, PAQ, humidity and air movement. Because of the elevated temperature and humidity levels, the normal ASHRAE seven-point thermal sensation scale was expanded to nine points by including 'very hot' and 'very cold' (Figure 3). The sensation scale units are: -4 very cold, -3 cold, -2 cool, -1 slightly cool, 0 neutral, 1 slightly warm, 2 warm, 3 hot, 4 very hot. There is a break in the middle of the comfort scale. Scale units +0 to 4 range from just comfortable to very comfortable, and -0 to -4 range from just uncomfortable to very uncomfortable. For the satisfaction scale, positive value means satisfied: the higher the value, the better the satisfaction; negative values means dissatisfied: the lower the value, the lower the satisfaction.

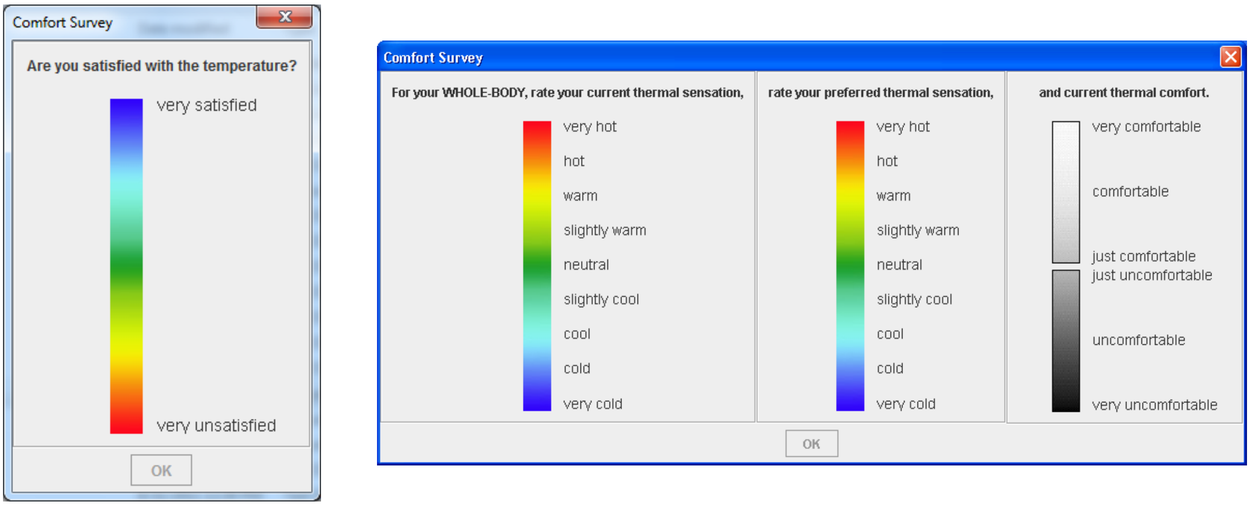


Figure 3 Sample survey questions

2.3.2.5 Procedure

On arrival, the subjects were asked to rest seated outside the chamber for 10 to 20 minutes to stabilize their metabolic rate. Then the subjects entered the chamber and were exposed to the first air speed. After 30 minutes they were instructed to stand up and take a one-minute break, during which the experimenter changed the fan setting. Then they returned to their workspace and experienced the second air speed. In this way the subjects were exposed to air speeds in four consecutive 30-minute periods. While exposed to each air speed condition, the subjects answered the survey questions at 1 minute, 15 minutes and 30 minutes after being seated.

2.3.2.6 Statistical analysis

Analysis was done for each test condition with repeated measures one-way analysis of variance. Acceptability votes were treated as binary variables for analysis of percent dissatisfied (PD, -4 to -0 unacceptable, +0 to 4 acceptable), and Fisher's exact test was employed to test such data. All differences were accepted as significant at the 0.05 level. Analyses were made with Graphpad software.

2.3.3 Results

2.3.3.1 Overall thermal sensation Thermal comfort

Mean whole body thermal sensation votes were plotted against effective temperature (ET*, Figure 4). In the control conditions (no fan), subjective votes increased from neutral at 26ºC (60% and 80%) to slightly warm at 28ºC (60% and 80%), and to warm at 30ºC (60% and 80%). For all tests with air movement, TS (thermal sensation) votes were well within the neutral zone (-1 to 1).

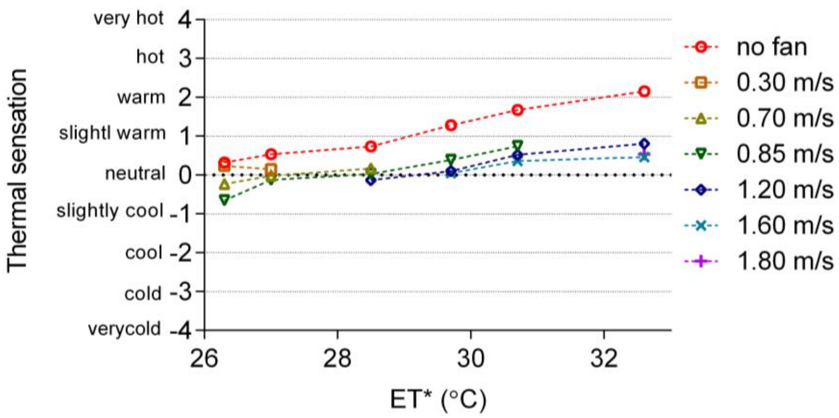


Figure 4 Mean thermal sensation votes and ET*

Statistical analysis for each test condition show no difference between "no fan" and "0.3m/s" at 26ºC, 60%, the same is true for "0.7m/s" and "0.85m/s", however there is significant difference between "no fan, 0.3" and "0.7, 0.85", indicating higher air speed induced cool sensation at this specific test condition (26ºC, 60% RH). At 26ºC, 80% no significant difference was found between all three air speeds. At this test condition (26ºC, 80% RH), the air speeds at the three levels didn't cause slightly cool thermal sensation as at 26ºC, 60% RH condition, therefore no significant difference was found among different speed levels. At the rest test conditions, significant differences were found only between "with air movement" and "no fan".

In the no-fan condition, mean thermal comfort votes decreased rapidly from comfortable to uncomfortable as ET* increased. Subjects were comfortable at the lower temperatures 26 ºC (60% and 80%) and 28 ºC/60%, but were uncomfortable when ET* went beyond 29.7 ºC with no air movement.

With air movement, the mean thermal comfort votes were comfortable (mean comfort vote above 1) for all test conditions (Figure 5). No significant differences of air speeds on thermal comfort votes were found at 26 ºC (60% and 80%). At higher temperatures, comfort votes at elevated air movement were significantly better than “no fan” (Friedman test, p<0.001). No differences were found among the conditions with different ceiling fan speeds. Air movement highly improves thermal comfort especially at 30ºC (60% and 80% RH), increasing mean comfort votes by 1.5 and 2.5 comfort scale unites respectively.

Condition 3 (28ºC and 60%RH) with 0.7 and 0.85m/s air speeds seems the highest mean thermal comfort votes (2.1 scale units), which was even higher than lower temperatures (26ºC and 60%RH with air movement), although the difference was not statistically significant.

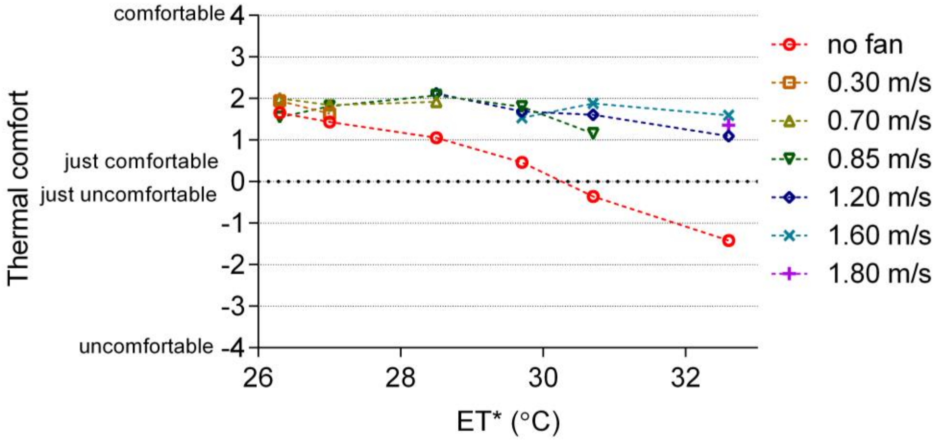


Figure 5 Mean thermal comfort votes and ET*

When there was no air movement, nearly 100% of subjects wanted to be “cooler” (Table 4, “C” means “Cooler”, “N” means “No change”, “W” mean “Warmer”). When there was elevated air movement in test conditions 1 to 5, most subjects preferred “no change” at the higher air speed conditions, which means that subjects felt fine with higher temperatures with elevated air movement. However, 50% of subjects wanted to be “cooler” at 30 ºC and 80% RH even at the maximum air speed of 1.8m/s, indicating that although air movement could maintain comfort (Figure 5) at this condition, subjective preference was toward a cooler environment with lower temperature and/or relative humidity.

Table 4 Percentage of thermal preference

	Thermal preference (%)											
	C	N	W	C	N	W	C	N	W	C	N	W
26°C		No fan			0.3m/s			0.7m/s			0.85m/s	
60%RH	25	75	0	13	81	6	0	81	19	6	56	38
26°C		No fan			0.3m/s			0.7m/s			0.85m/s	
80%RH	44	56	0	31	69	0	19	75	6	12	69	19
28°C		No fan			0.7m/s			0.85m/s			1.2m/s	
60%RH	69	31	0	31	69	0	25	75	0	6	88	6
28°C		No fan			0.85m/s			1.2m/s			1.6m/s	
80%RH	75	25	0	31	69	0	25	63	12	6	94	0
30°C		No fan			0.85m/s			1.2m/s			1.6m/s	
60%RH	100	0	0	56	44	0	44	56	0	19	81	0
30°C		No fan			1.2m/s			1.6m/s			1.8m/s	
80%RH	100	0	0	69	31	0	50	50	0	50	44	6

2.3.3.2 Perceived air quality and freshness

PAQ and freshness votes decrease slightly above ET* of 29ºC (Figure 6). Air movement was found to significantly improve PAQ in all the test conditions, although statistically significant differences were found only in conditions 3 to 6. As with the thermal comfort votes, the improvement in PAQ caused by air movement increased at higher temperatures and humidity.

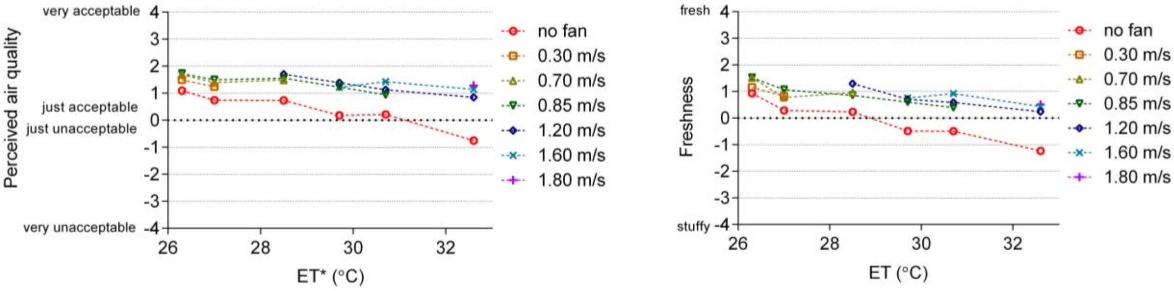


Figure 6 Perceived air quality and freshness votes with ET*

The trend of freshness votes was very similar to PAQ votes, however, the freshness votes were always about 0.5 scale unit lower than PAQ votes. Subjects perceived the air as stuffy when ET* was higher 29.7 ºC in still air. Air movement can highly improve perception of freshness, especially at higher ET*. The highest votes were found at 26ºC and 60%RH, similar for the rest condition except for at 30ºC and 80%RH. The subjects perceived air slightly fresh at this condition, although it was much better than the same environmental condition without air movement.

2.3.3.3 Acceptability of humidity and humidity sensation

The subjects' acceptability of humidity votes were significantly influenced by the humidity levels themselves (Figure 7). From test condition 1 to 5, there were no significant differences with and without air movement, and no differences among different air speeds, although mean votes with air movement were better than that without. Air movement significantly improved humidity acceptability at test condition 6, where an almost 2-scale-unit difference is seen compared to the still air condition. However, the difference is not statistically significant.

Similar to the acceptability of humidity votes, humidity levels had a significant effect on the sensation of humidity (Figure 7). However no significant differences were found with and without fan and between different air speeds at test condition 1 to 5. Mean votes were within -0.5 to 0.5 range from test condition 1 to 5. Test condition 6 was the only condition in which air movement significantly improved humidity sensation; air movement improved humidity sensation 1 scale unit, from -1.5 to -0.5 (the lower value means higher humidity sensation).

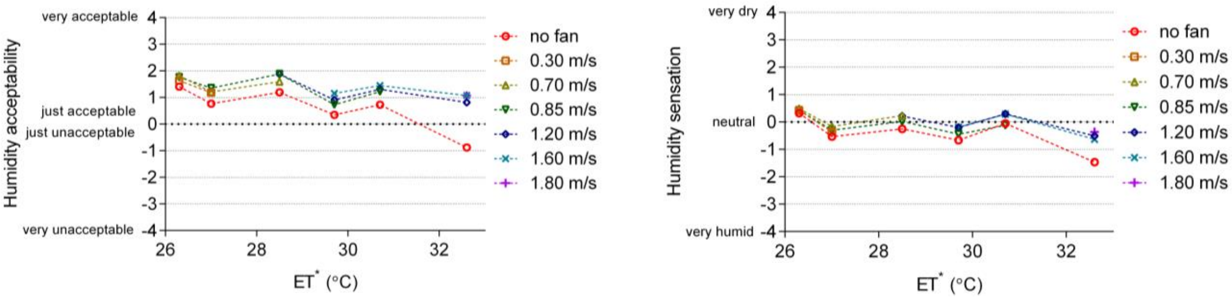


Figure 7 Humidity acceptability and sensation and ET*

2.3.3.4 Acceptability and preference of air movement

When fans were off, subjects' acceptability of air movement deceased as ET* increased (red line in Figure 8), meaning that the warmer the ambient condition is, the more that people would like to have air movement. When the fan was on, subjects were satisfied with air movement for all test conditions (Figure 8), and the acceptability levels are very similar among all the conditions. The differences between "no air movement" and "with air movement" were significant at test conditions 2 to 6.

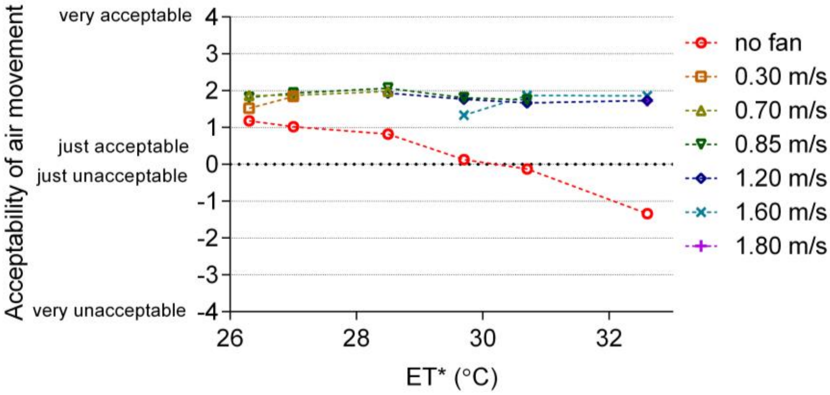


Figure 8 Acceptability of air movement votes and ET*

There were significant differences in terms of air movement preference. When fans were off, 50% of subjects wanted more air movement at 26ºC (60% and 80% RH); at 28ºC 60%RH and 80%RH, 75% and 81% respectively; at 30ºC (60% and 80% RH), 94%. When air movement was provided, the preference was largely depended on the air speeds. Air speed level 2 (0.7m/s) was found to have highest percentage "want no change" for condition 1 and 2, 88% and 75% respectively. For condition 3 (81%) and 4 (75%) the air speed with highest percentage "want no change" was 0.85m/s. At 30ºC and 60% RH, 81% of subjects wanted "no change" at 1.2m/s air speed, while for 30ºC and 80% RH, this was found at 1.6m/s (69% wanted no change). Below these air speeds at each test condition, more subjects wanted "more" air movement while they wanted "less" when air speeds were higher than these settings.

The relationship between preferred air speed and ET* (Figure 9) can be used as the bases of automatic control of the fan at different combinations of temperatures and humidity.

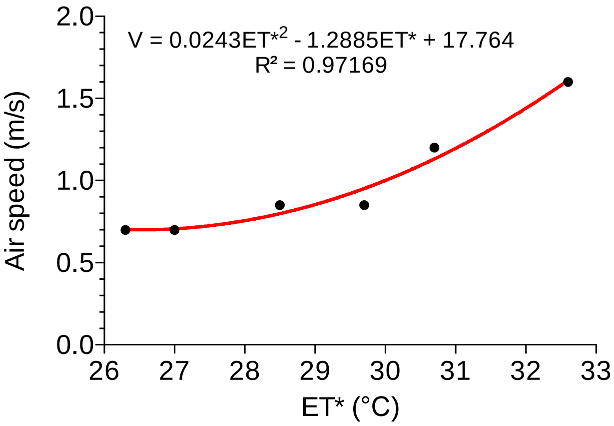


Figure 9 Preferred air movement and ET*

2.3.3.5 Noise and eye-dryness

No significant differences were found for noise acceptability for all test conditions and air speeds (Figure 10).

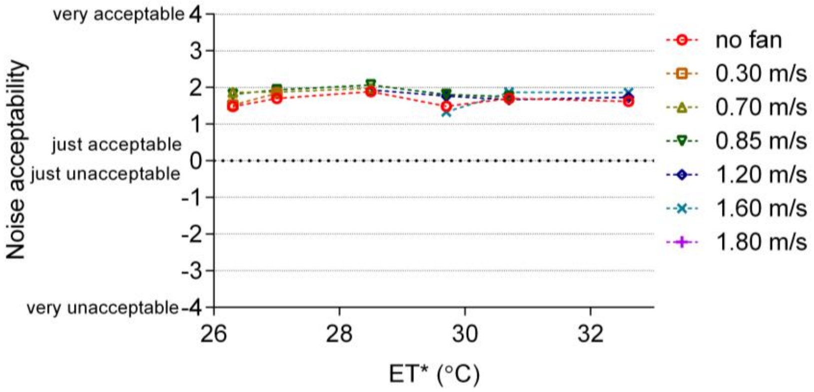


Figure 10 Noise acceptability and ET*

No significant differences were found in dry-eye discomfort for any test condition (Figure 11). At most test conditions, especially at 28 ºC and 80% RH, subjects voted higher dry-eye acceptability when there was air movement than with no air movement, (although the differences were not significant), indicating air speed did not cause dry-eye discomfort under the test conditions in the current study.

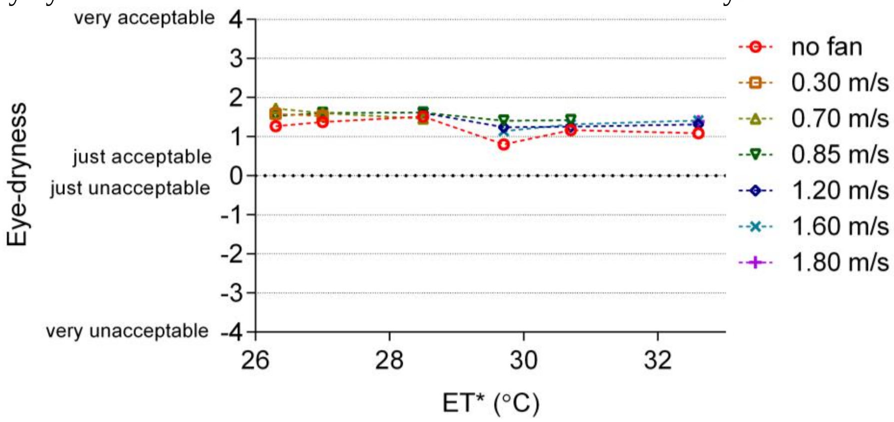


Figure 11 Eye-dryness and ET*

2.3.3.6 Percentage dissatisfied with thermal environments

In still air (no fan), more than 20% of subjects were dissatisfied (PD, percentage of dissatisfied) with thermal environment (Figure 12), PAQ, air movement and humidity when ET^* was higher than 27 ºC (26 ºC and 80% RH).

With air movement, PD was less than 10% for all conditions except at 30ºC, 60%RH with 0.85m/s air speed (20% PD). With higher speed (1.2 m/s, at the same condition, 30ºC, 60% RH), people were satisfied with the thermal environments and PD dropped to zero. Subjects were satisfied with air movement at all air speed levels. The humidity level strongly influenced the dissatisfaction level with PAQ and humidity, both with and without air movement.

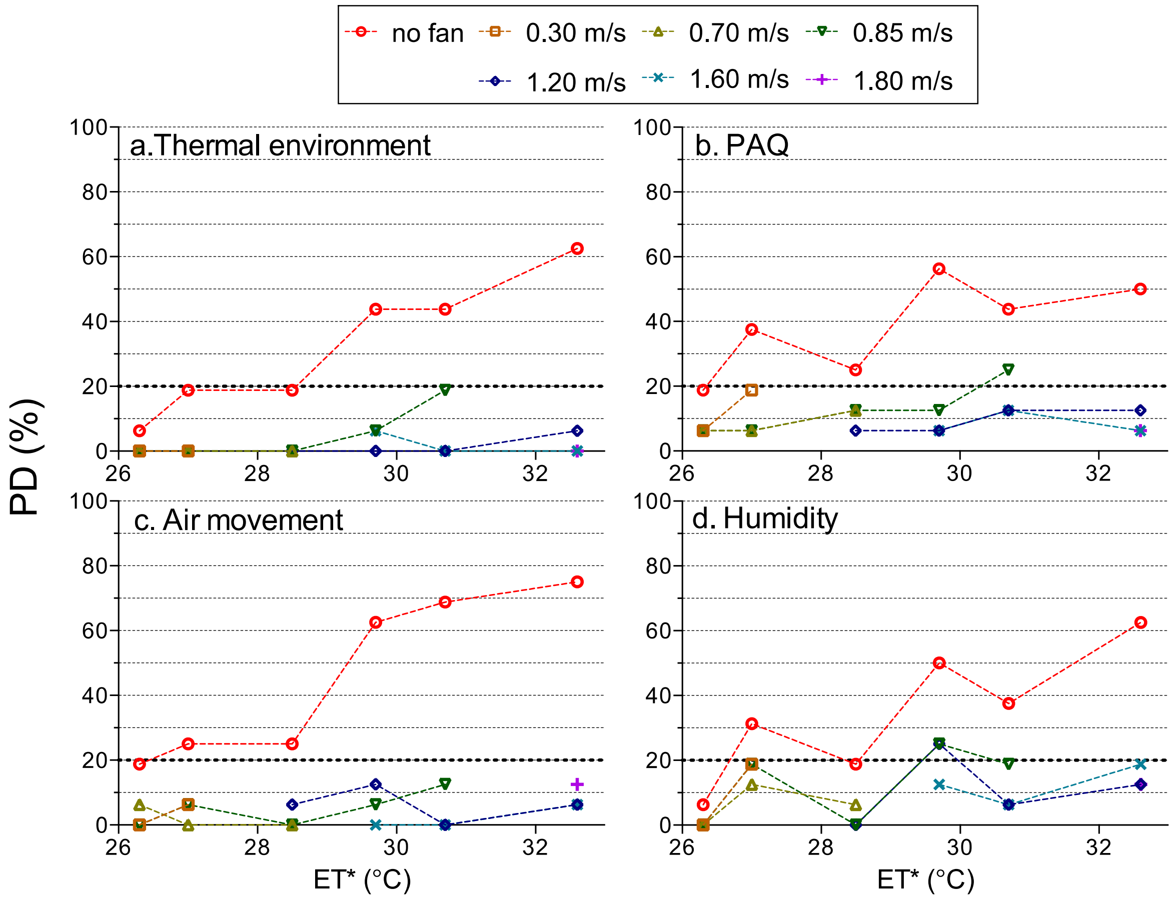


Figure 12 Percentage dissatisfied with thermal environment, humidity, PAQ and air movement

2.3.4 Discussion

2.3.4.1 Comparison with ASHRAE Standard 55-2010

The 80% acceptable limits found in the current study for environments with and without fan are shown against the ASHRAE comfort zone (Figure 13). Without fan, the upper limit is 26 ºC at 80% RH, and 28 ºC at 60% RH. With fan the limit can be extended to 30 ºC and 80% RH, which is well beyond the ASHRAE summer comfort zone.

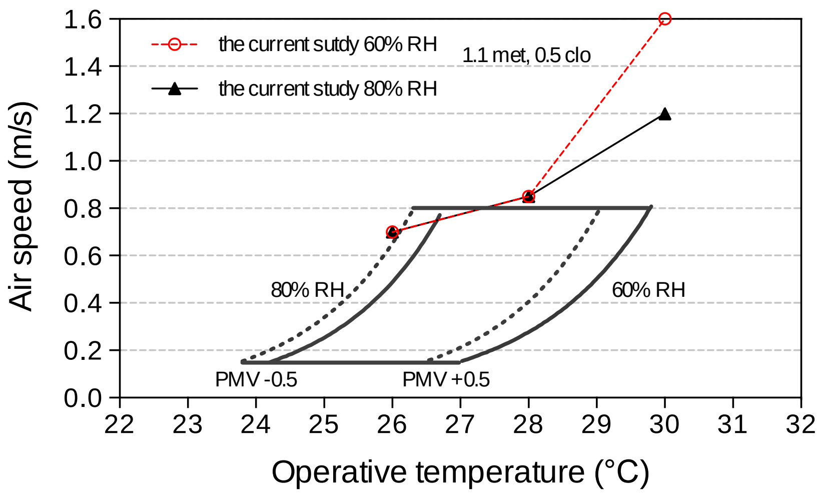


Figure 13 Comfort temperature and RH limits with ceiling fans against an ASHRAE comfort chart

When ET* is lower than 30ºC, the ASHRAE limit agrees with the results of the current study. When ET is higher than 30ºC, the preferred air speed (Figure 9) extends beyond the 0.8 m/s limit, without causing dissatisfaction with air movement, noise or eye-dryness.

2.3.4.2 Comparison with other studies

The preferred air speed and corrsponding upper comfort temperature and RH for the present study and previous studies are compared (Table 5). Our results agree well with most previous studies (Rohles 1983, McIntyre 1978, and Scheatzle 1989), but not with Ji et al. With similar air speeds, the upper comfort temperature limit found in Ji's study (Ji et al. 2003) was higher than the present study.

Table 5 Comparison with previous ceiling fan studies

study	Preferred velocity	Comfort limit
Rohles（1983）	1.00 m/s	29.5°C/50% RH
McIntyre（1978）	0.80 m/s	28.0°C/50% RH
Scheatzle（1989）	1.02 m/s	32.2°C/39% RH 31.1°C/50% RH 28.3°C/73% RH
Ji （2004）	0.80 m/s	29.7°C/60% RH
CBE study (2013)	1.30 m/s	30.0°C/30% RH
Current study (2013)	0.85 m/s 0.85m/s 1.20 m/s 1.60 m/s	28.0°C/60% RH 28.0°C/80% RH 30.0°C/60% RH 30.0°C/80% RH

We also did linear regressions of enthalpy of air against PAQ for with-fan and without-fan conditions respectively (Figure 14). As we can see from the chart, as enthalpy increases, PAQ decreased faster without fan than with fans. The differences of the slope of the two regression line are significant by ANCOVA test (P=0.0002). The no-fan line (red line in the figure) is similar to the findings by Fang et al and Toftum (1998). However, when the fan was on, the PAQ decreases much more slowly with increase in enthalpy (green line in the figure).

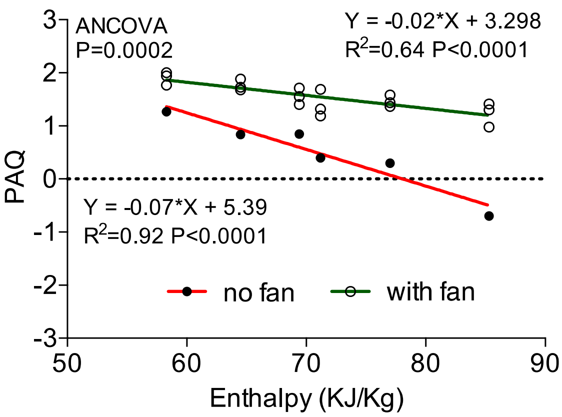


Figure 14 PAQ votes vs. enthalpy for with and without fan tests

2.3.5 Conclusion

1. The ceiling fan improved the subjects' thermal comfort and PAQ significantly, and increased their acceptance of environmental air movement and humidity significantly, without causing discomfort of eye-dryness or noise.
2. When the ceiling fan was off, subjects are only comfortable at 26°C/80% and 28°C/60%. However with air movement, subjects were comfortable up to the 30°C/80%RH environmental condition, which is well beyond the current ASHRAE limits to summer comfort temperature and RH.
3. The preferred air speed exceeded the upper limit set in the ASHRAE Standard.
4. PAQ decreased rapidly with enthalpy increase when there was no air movement. With air movement, the slope of the decrease is significantly flatter than in the no-fan condition.
5. The present study was performed with 30 minutes exposure to each condition. Further studies might test the effects of longer exposure times.

Acknowlegements

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2.4 Project 2: Human comfort and perceived air quality in hot and humid conditions with horizontal airflow (floor fans)

(this paper has now been published in Building and Environment)

Abstract
This study examined the effects of personally controlled air movement on human thermal comfort and perceived air quality (PAQ) in warm-humid environments. At temperatures 26, 28, and 30°C, and relative humidity (RH) 60% and 80%, sixteen human subjects were exposed to personally controlled air movement provided by floor fans in an environmental chamber. The subjects reported their thermal sensation, thermal comfort, and PAQ during the tests. Two breaks periods with elevated metabolic levels were used to simulate normal office activities. Results show that with personally controlled air movement, thermal comfort could be maintained up to 30°C and 60% RH, and acceptable PAQ could be maintained up to 30°C 80% RH, without discomfort from humidity, air movement or eye-dryness. Thermal comfort and PAQ were resumed within 5 minutes after the breaks. The 80% acceptable limit implicit in comfort standards could be extended to 30°C and 60% RH. The average energy consumed by the fans for maintaining comfort was lower than 10W per person, making air movement a very energy-efficient way to deliver comfort in warm-humid environments.

Keywords: Thermal comfort; Perceived air quality; Warm-humid; Air movement; Personal control; Low energy.

2.4.1 Introduction

Compressor cooling in buildings is already the main contributor to peak load in long tropical or sub-tropical summers, affecting both energy use and electrical grid safety, and this trend is going to accelerate in the coming decades with the cooling demand growth in South China, South-east and South Asia. In the face of the huge energy impacts that this increase is causing, one must examine alternative ways of achieving comfort in warm-humid environments.

In warm environments, air movement has the potential to conserve energy while maintaining occupants' comfort. Field studies in warm-humid climates have shown that occupants remained comfortable in naturally ventilated buildings with natural wind and fans (de Dear et al. 1993, Busch 1992, Kwok 1998, Nicol 2004, Yang et al. 2012, Candido et al. 2010, Zhang et al. 2011). In air-conditioned buildings, a reanalysis of ASHRAE field study database also shows that a majority of occupants preferred more air movement when their thermal sensations are slightly warm or warmer (Hoyt et al. 2009).

Recently, ASHRAE Standard 55 "Thermal environmental conditions for human occupancy" increased allowable air movement for comfort in warm environments (ANSI/ASHRAE/IES Standard 55-2010), providing more opportunities for air movement design for cooling (Arens et al. 2010). Air movement has long been shown to be effective at increasing convective and evaporative heat loss in warm environments Rohles et al. 1974, McIntyre 1978, Scheatzle et al. 1989, Tanabe et al. 1987). Laboratory studies have found that thermal comfort can be well maintained by personally controlled horizontal air movement in ambient temperatures as high as 27.8°C and 30°C (Fountain et al. 1994, Arens et al. 1998). Recent studies have shown that a 3W personal fan maintains a neutral thermal sensation up to 30°C and 50% RH (Zhang et al. 2010).

Of such studies, relatively few have combined high temperatures and high humidity. One of these, by Tanabe and Kimura (Tanabe and Kimura 1994) tested horizontal air movement provided by an array of box fans on sedentary subjects wearing 0.6clo, and found that comfort could be maintained at 31°C, 50% RH with 1.6 m/s airspeed, and at 29°C, 80% RH with 1.4 m/s air speed. Kubo et al. (1997) tested self-selected frontal airflow with subjects wearing 0.35 clo at 30°C and 80% RH. Subjects chose a cooler-than-neutral thermal sensation by selecting an average air speed of 1.27 m/s at 30°C and 80% RH. Even higher air speeds (up to 3 m/s) has been preferred by subjects in Thailand (Khedari et al 2000) and Hong Kong (Chow et al. 2010) at temperatures higher than 30°C and at RH values as high as 85%.

Studies have also shown that air movement significantly improves people's PAQ in warm temperature and moderate humidity conditions up to 30°C (Zhang et al. 2010, Arens et al. 2008, 2011), and in humid environments up to 28°C (Melikov and Kaczmarczyk 2012, Melikov et al. 2012, 2008), although the causal mechanism behind this is not well understood.

However, the cited research has mainly focused on overall thermal sensation, paying less attention to general acceptability of thermal environment, PAQ, humidity and air movement, and possible eye discomfort due to high air speeds. Previous studies used large fan-box or personalized ventilation systems; less has been done with regular room fans, which are easier and cheaper to implement in buildings. Another issue is human reaction to thermal transients, because air movement, unlike temperature and humidity, is almost never uniform across space, and the ability of air movement to restore comfort after periods of time spent in still air is important. These issues are of great importance because answers to these questions might impact the wide adoption of air movement devices.

The aims of the study were to: (1) examine the ability of personally controlled low-energy fans to maintain thermal comfort and PAQ in warm-humid environments; (2) examine the ability of air movement to restore thermal comfort and PAQ after a short burst of activity; (3) determine the threshold values for temperature and humidity under which acceptable comfort can be maintained with personally controlled air movement.

2.4.2 Methods

The experiments were carried out at the environmental chamber at the Center for the Built Environment (CBE), University of California, Berkeley in June 2012.

2.4.2.1 Facilities

Climate chamber

The CBE climate chamber measures 5.5m×5.5m×2.5m, controlling temperature to an accuracy of ±0.5°C, and RH ±3%. The chamber has windows on two sides, South and West. The windows are well shaded by fixed external shades. The windows temperature is controlled by a dedicated air system. The room air temperature is controlled and ventilated by 8 floor grill diffusers, and the air is exhausted through a ceiling return grill. The outdoor flow rate in this study was around 85-104 L/s. Since the maximum number of occupants was 5 (four subjects and one experimenter), the minimum outdoor air supply rate was between 17.0 and 20.8.L/s person, much higher than the current requirement for office buildings (4.3 L/s.person) (ANSI/ASHRAE/IES Standard 62.1-2010).

The chamber was set up to simulate a typical open plan office without partitions (Figure 15). Four workstations (WS) were set up so that four subjects could be tested at the same time. Each workstation was assigned a floor fan, a laptop and a mesh chair. The fans were placed in the middle of the room, blowing air toward the corners in order to minimize interaction between the airflows.

Chamber configuration

Air speed measurement

Figure 15 Layout of the test chamber and air speed measurement

The fans

The commercially available fan is very energy efficient and consumes only 2 to14 W for fan speed settings 1 to 7 (Table 1). Each fan was placed 1.5m away from the position of the subject. The subjects controlled the fan speeds with a remote controller. Mean air speeds (1.1m height at where the subjects sat) ranged from 0.4 m/s to 1.7 m/s from level 1 to level 7 (Table 6).

Table 6 Fan power and measured air speed at each setting

level	Power (W)	Air speed (m/s)				
		ws 1	ws 2	ws 3	ws 4	mean
0	1*	0.05	0.05	0.05	0.05	0.05
1	2	0.49	0.46	0.45	0.41	0.44
2	3	0.54	0.58	0.56	0.61	0.57
3	4	0.65	0.72	0.66	0.71	0.69
4	7	1.19	1.30	1.25	1.34	1.27
5	9	1.28	1.42	1.44	1.40	1.39
6	11	1.54	1.55	1.62	1.63	1.59
7	14	1.72	1.70	1.73	1.74	1.72

*Plug load

2.4.2.2 Physical measurements

Room temperature and RH were measured continuously with HOBO Temperature and RH data loggers attached to the back of each table during all the tests. The accuracy of temperature measurement was ± 0.35°C, and RH accuracy was ± 2.5%.

Air speed was measured with omnidirectional hotwire anemometers (Sensor Inc., with a response time of 2s and an accuracy of 0.02 m/s ± 1.5% of reading). Measurements were made before and after the experiment to characterize the air speed at each workstation, at the 0.1m, 0.6m, and 1.1m levels. Air speeds were determined for each of the available fan speeds. During the tests, the fan power was measured each minute with wireless power meters with 1W accuracy, to determine the fan speed levels (Table 1) from which the air speeds at the workstation could be derived.

Indoor CO2 level was measured in the middle of the test chamber, and outdoor CO2 was measured in the supply duct, both with Vaisala CARBOCAP® GMW115Transmitters, at an accuracy of ±2 % of range and ±2 % of reading.

Prior to the experiment, all these test instruments were calibrated against higher accuracy instruments.

2.4.2.3 Test conditions

Test conditions (Table 7) were selected to represent the typical temperature and humidity levels in free running buildings in warm-humid climates, based on the ASHRAE RP 884 database (deDear 1998) and the extensive field data from naturally ventilated buildings in Guangzhou (Zhang et al. 2010). The corresponding ET^*, SET (at still air and 0.1 m/s), PMV, and enthalpy values are also included in the table.

Table 7 Test conditions

Test conditions	T (°C)	RH (%)	ET^* (°C)	SET (°C)	PMV	Enthalpy(KJ/kg)
1	26	60	26.3	25.7	0.12	58.34
2	26	80	27.0	26.4	0.26	69.37
3	28	60	28.5	28.0	0.67	64.51
4	28	80	29.7	29.4	0.82	77.00
5	30	60	30.7	30.4	1.42	71.15
6	30	80	32.6	32.5	1.59	85.29

2.4.2.4 Subjects

Eight male and eight female subjects participated in the tests (see Table 8 for the subjects' anthropology data). The subjects were instructed to dress in typical summer clothes (0.5 clo) – T-shirt or short sleeve shirts, jeans or light pants, underwear, light socks, and sandals. Mesh chairs were used to minimize the additional insulation of the chair. The subjects were allowed to lean forward or backward as in real offices, but not allowed to stand up or move around during the tests.

Prior to tests all the subjects attended a training session to get familiar with the chamber, test procedure, control of fans, and survey questions.

Table 8 Subjects' anthropology data

Sex	Sample size	Age	Height (m)	Weight (Kg)	BMI#
Female	8	30.6±6.4*	1.65±0.3	56.4±4.2	20.7±1.2
Male	8	27.2±2.3	1.74±4.9	69.5±0.6	22.9±1.8
Female + Male	16	28.9±5.0	1.70±0.6	63.0±8.5	21.8±1.9

*Standard deviation; #Body Mass Index = weight (kg) / [height (m)]^2

2.4.2.5 Experimental procedure

Before each test, the chamber was preconditioned to the temperature and RH according to Table 1 and the floor fans were turned on at a low speed to create background air movement in the chamber.

Each test took 2 hours, including one adaptation period, three 30-minute test periods, and two short breaks (Figure 16). The subjects were instructed to adjust their fan speeds freely in the adaptation and sedentary periods, but not during break periods.

At the beginning of each test, the subjects sat for 15 minutes in the chamber to adapt to the environment, while adjusting fan speed to maintain personal comfort. They answered computerized surveys at the beginning and end of this period. The first sedentary test period lasted 30 minutes, during which the subjects worked on their computers. They answered surveys at the middle and end of the period.

The first break (low activity break) was 5 minutes, in which the subjects were asked to turn fans off, stand up and leave their workstations. They were instructed to walk around and stretch their arms and legs, and to take 12 vertical steps with a 22 cm height step stool. This was to simulate activity levels resembling those in offices when occupants are away from their desks (go to coffee machine, printer and etc.). After the exercise the subjects went back to their workstations, answered a survey, turned on their fan, and immediately took a second survey. Another 30 minutes elapsed until the second break, with three surveys.

The second break (high activity break) was 10 minutes. The subjects were asked to turn the fans off again and take four sets of 20 vertical steps to simulate going upstairs and downstairs. After this break the subjects resumed sedentary in their workstations for a final 30 minutes, answered an identical set of survey questions, and left the test chamber.

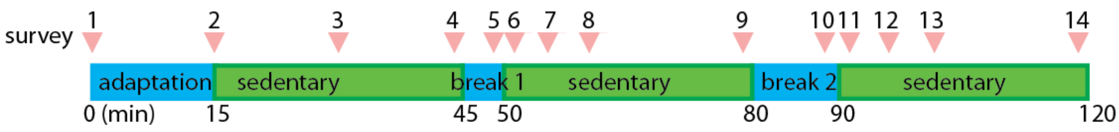
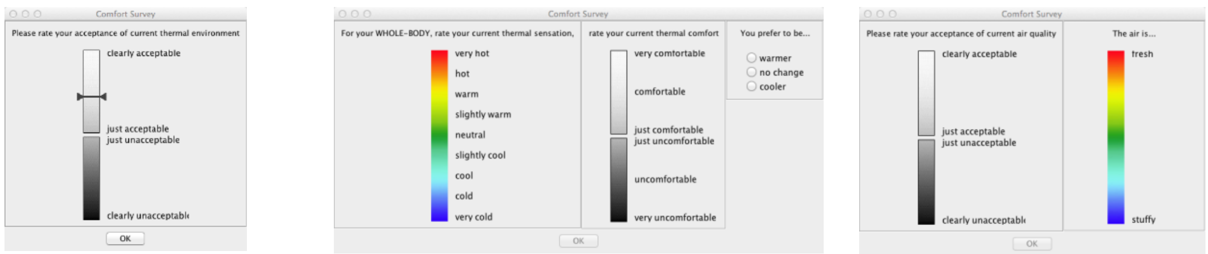


Figure 16 Experimental procedure

2.4.2.6 Questionnaires

The survey questionnaires include six parts: (1) overall thermal acceptability; (2) thermal sensation (TS) in a 9-point extended ASHRAE scale, thermal comfort (TC) and thermal preference; (3) perceived air quality (PAQ) and air freshness; (4) humidity acceptability and sensation; (5) air movement acceptability (AMA) and preference; and (6) eye-dryness discomfort. All the acceptability, TC, eye-dryness scale, and air freshness scales ranged from -4 to 4. A few examples of the survey questions are shown in the Fig. 3. The survey questions automatically appeared on the subjects' computer screen based on the schedules described above (Figure 17).



Thermal acceptability | Thermal sensation, thermal comfort and preference | Perceived air quality and air freshness

Figure 17 Sample survey screens

2.4.2.7 Statistical analysis

The statistical analysis was performed using Graphpad 6 Prism (Graphpad, San Diego, CA) with repeated-measure Friedman test with Dunn's post-hoc test. Statistical significance was tested at $p<0.05$.

2.4.3 Results

2.4.3.1 Indoor physical environment

The indoor temperature and RH were controlled close to the planned conditions (Table 9) in the tests. The indoor CO2 level was 200 ppm above that of the outdoors during sedentary periods, and went up to 300 ppm higher during elevated activity periods. This means there was adequate ventilation because CO2 is an indicator of inadequate ventilation.

Table 9 Planned and measured temperature and RH

Planned		Measured（mean over 4 WSs）	
T (°C)	RH (%)	T (°C)	RH (%)
26	60	25.97±0.17	61.9±0.8
26	80	25.86±0.27	78.3±0.8
28	60	27.99±0.07	61.5±0.6
28	80	27.86±0.13	78.7±0.7
30	60	29.84±0.07	61.9±0.6
30	80	30.01±0.16	77.4±0.7

2.4.3.2 Subjective responses - responses over time

Fig. 4 displays mean overall TS, TC and PAQ over time for the entire experiment. The subjects' mean TS in each sedentary period were between neutral to slightly warm in all test conditions except 30°C and 80% RH. The TS votes were slightly higher at the beginning of adaptation periods, and reached stable levels within 30 minutes (Figure 18). At the end of the first break, TS increased 0.5 scale units in condition 1, 4 and 5; and increased nearly 1 scale unit at the condition 2, 3 and 6. Turning on the fan after the break 1 significantly decreased TS to neutral levels. TS reached stable levels within 5 minutes in the second sedentary period. The second break increased TS from the last sedentary votes by 1 to 1.5 scale units, to the "warm" or "hot" side. Again, TS decreased immediately after turning on the fan, and reached previous levels within 5 minutes.

The subjects were thermally comfortable and acceptable with the air quality in all test conditions except at the beginning of adaptation periods and after the two breaks (Figure 18). Similar patterns were found for the TC and PAQ votes. The TC and PAQ votes were lower at the beginning of the tests, and improved by air movement over time in the first 45 minutes. The two high activities breaks decreased the TC and PAQ levels significantly; when exposed to air movement after the breaks, TC and PAQ were improved immediately and were completely restored within 5 minutes.

In all test conditions, no significant differences were found between the 3 last votes for TS, TC and PAQ in any of the three 30-minute sedentary sessions, indicating that the adaptation period and the two high activities breaks have no significant effect on the subjects' votes after they had been exposed to personally controlled air movement for 30 minutes.

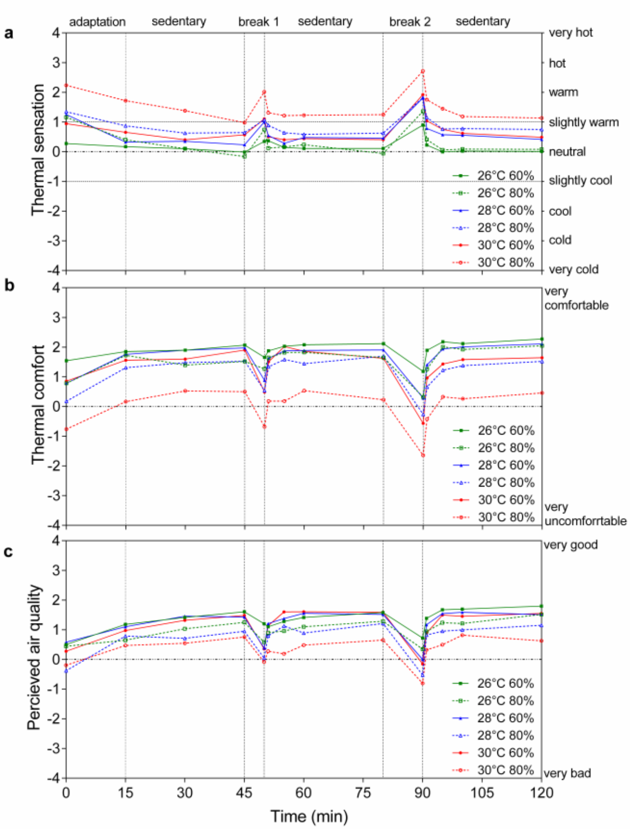


Figure 18 Mean overall thermal sensation, thermal comfort, and PAQ votes over time

2.4.3.3 Steady-state responses

As mentioned before, pre-conditions (adaptation and high activity breaks) had no significant effect on the subjects' votes once stability was reached, and therefore the last votes of each sedentary period were pooled and analyzed as steady state responses.

3.2.2.1 Thermal sensation, thermal comfort and thermal preference

Figure 19 shows the boxplot of TS votes at the six test conditions. The median TS votes were almost 0 at 26ºC 60% and 80% RH; the distribution is well within ±1, indicating that almost all the votes are neutral. For the test conditions 2, 3, and 4, more than 75% percent of the TS votes were in the neutral range, with median TS at 0.2, 0.52 and 0.49 respectively. The test condition 6 is the only condition with significantly higher votes than the rest of the test conditions ($P<0.001$), with about 50% of TS votes above 1.

The TC responses (Figure 19) shows that comfort was well maintained in test conditions 1 to 5, with nearly 100% of the subjects were comfortable in conditions 1 to 3, and more than 80% in conditions 4 and 5. In condition 6, only 60% of the subjects reported being comfortable.

The TP votes show that over 60% of the subjects wanted "no change" in test conditions 1 to 5, and 40% in test 6 (Figure 19). However, more than 20% of the subjects wanted to be "cooler" in test conditions 3, 4, 5, and nearly 60% of subjects in test 6.

Figure 19 shows significant effects of humidity on TS, TC and preference votes at higher temperatures (28ºC and 30ºC), which were associated with higher TS votes, lower TC votes and a larger percentage of the subjects wanting to be "cooler".

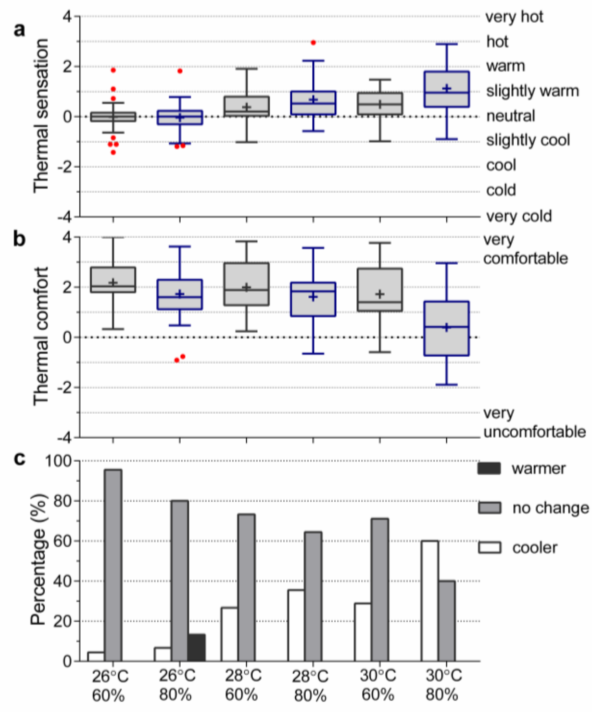


Figure 19 Thermal sensation, thermal comfort and thermal preference votes at each test condition. The figure shows median votes (lines), 25% to 75% quartiles (boxes) and ranges (whiskers). Mean votes are shown as crosses. Red dots are shown if extreme values are more than 1.5 times the interquartile range of the box.

2.4.3.4 PAQ and air freshness votes

The PAQ votes were similar among test conditions 1 to 5 (Figure 20). The acceptability for test condition 6 was significantly lower than the other conditions ($P<0.001$), but was still within the acceptable range. However, more than 20% of the subjects perceived the air to be stuffy in all the test conditions (Figure 20). At the test conditions 4 and 6, nearly 75% of the subjects perceived the air stuffy. Nonetheless, the subjects reported acceptable PAQ, which suggests that providing air movement may maintain acceptability of PAQ, but may not make the air feel fresh. Post-hoc analysis shows that freshness votes were lower at 80% RH than 60% RH at 28 ºC and 30 ºC ($P<0.001$), and lower at 28 ºC 80% RH than 30 ºC 60% RH ($P<0.001$). The subjects may have felt the air stuffy, despite the high fresh air supply rate in the chamber, because they were tested in a chamber with fixed windows. Also, the outdoor conditions during the test period were cool and dry rather than hot and humid. Further study in a hot-humid outdoor climate may be needed to determine this effect.

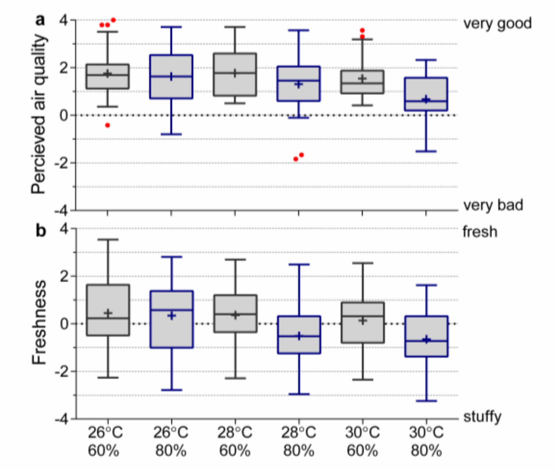


Figure 20 PAQ and air freshness votes. See Figure 17 definition of symbols.

2.4.3.5 Fan usage, air movement acceptability and preference

During all the tests the fan power for each WS was continuously measured. The measured fan power was continuously converted to fan speed levels. The air speed levels in the last 5 minutes during each sedentary session were selected to represent steady-state speed levels. The measured air speeds at the 1.1m heights were used in the current analysis. Figure 21 shows the air speed selected by the subjects under 6 test conditions (Figure 21), and the corresponding fan power (Figure 21). The selected mean air speed was 1.3m/s at 30°C and 80% RH, 1 m/s at 30°C/60% RH, 0.7 m/s at 28°C/60% RH and 80% RH, and 0.4 m/s and 0.3 m/s at 26°C/80% RH and 26°C/60% RH respectively. The mean energy consumption based on the selected speed was 3W to 10W. There were big individual differences among the subjects. Some preferred high air speed even when they felt cool, while some preferred no air movement even when they felt hot and uncomfortable.

Figure 22 shows that the distribution of the subjects' air movement acceptability votes was on the acceptable side, with nearly 75% votes at test condition 1 to 5 higher than 1 on the scale. There were no significant differences between test conditions 1 to 5. Only the test condition 6 was significantly lower than other test conditions ($p<0.0001$), however the majority of the votes were still on the acceptable side (89%).

Most of the subjects (more than 60%) indicated "no change" as their preference for air movement, in all the test conditions except in the test condition 6 (50%) (Figure 22). More subjects preferred "more" air movement rather than "less" in all test conditions except in test condition 2. Further analysis shows that only 10% of the subjects who wanted more air movement actually used the full power of their fan. For those subjects who didn't choose full power, the reason may be that the control of fan speeds was not step-less, thus some subjects chose a lower air speed than needed because the next speed level was considered too high for them. The 10% of subjects who wanted more air movement even while using the full power of their fans (all at 30°C and 80% RH) did not have sufficient air speed for this test condition. This information could be useful for the design and implementation of floor fans on adding more air speed control options and increasing the maximum air speed for environments with higher temperature and humidity.

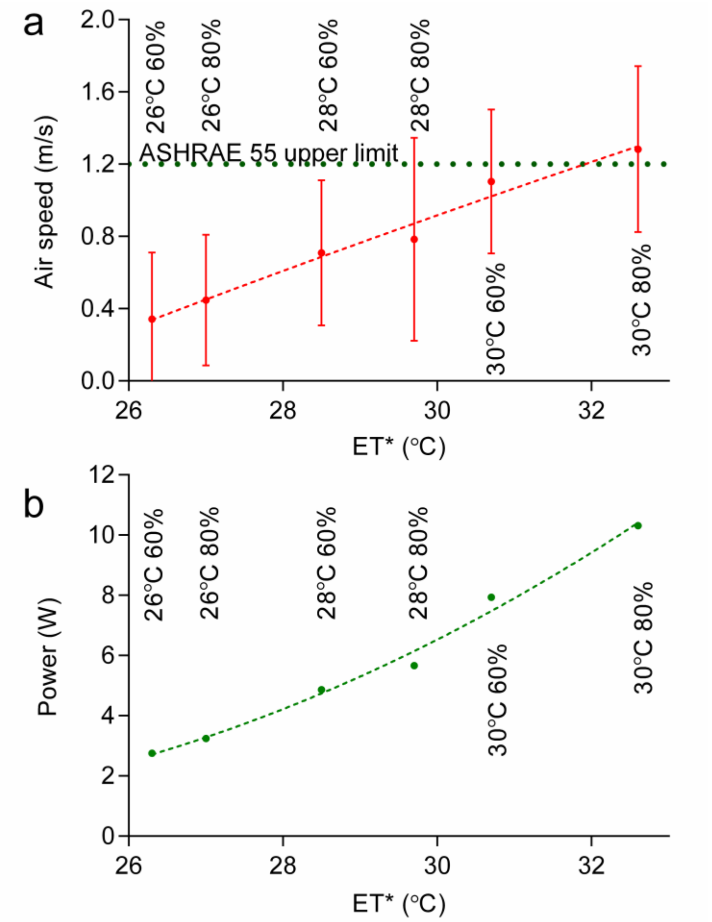


Figure 21 Air speeds selected by the subjects, and corresponding fan power

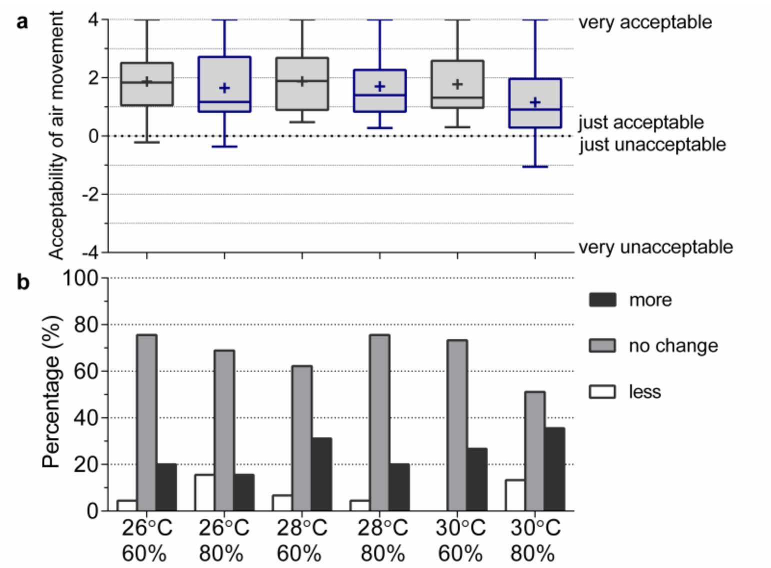


Figure 22 Acceptability and preference of air movement. See Figure 17 definition of symbols

2.4.3.6 Humidity acceptability and humidity sensation

Humidity acceptability votes were comparable among the test conditions 1 to 5, while in test condition 6 they were significantly lower ($P<0.001$), with nearly 40% of the subjects reporting unacceptable (Figure 23). The median humidity sensation votes were similar for the test conditions 1 to 5, with the subjects' humidity sensation votes evenly distributed around neutral. At 30 ºC and 80% RH, 75% of the subjects sensed the air as humid (Figure 23); this may explain the low acceptability of humidity at this test condition.

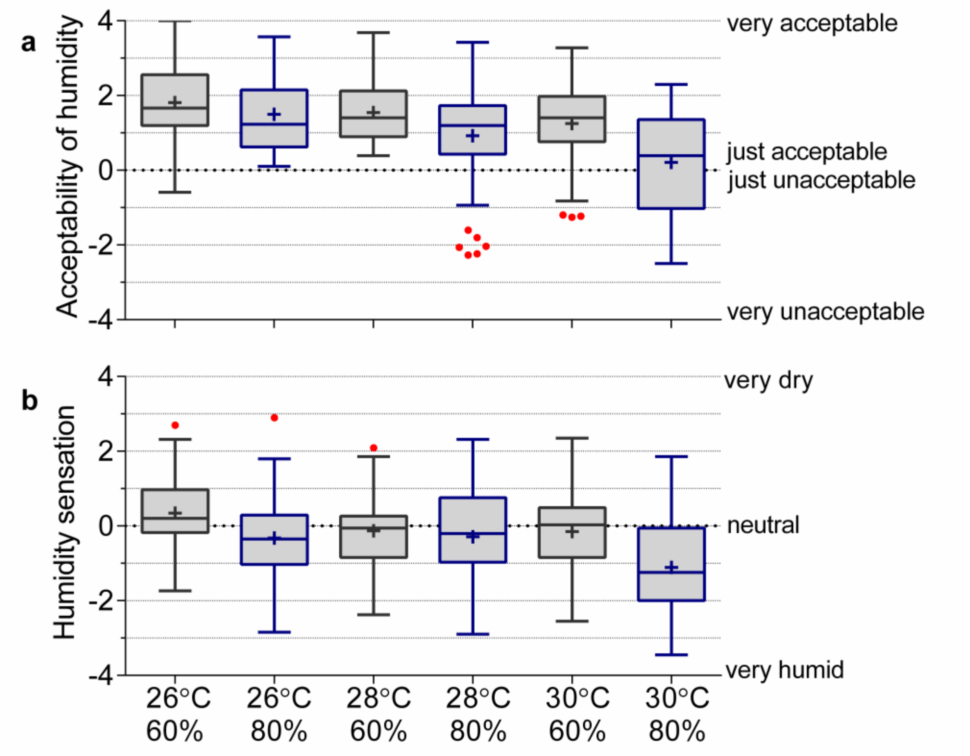


Figure 23 Acceptability of humidity and humidity sensation. See Figure 17 definition of symbols.

2.4.3.7 Eye-dryness discomfort

The distributions of Eye-dryness acceptability were mainly on the acceptable side at all tests conditions. No significant differences were found (P = 0.2895) among all the test conditions despite the high air speed chosen by the subjects, indicating that the elevated air speed controlled by the subjects did not cause eye-dryness discomfort in these warm-humid environments (Figure 24).

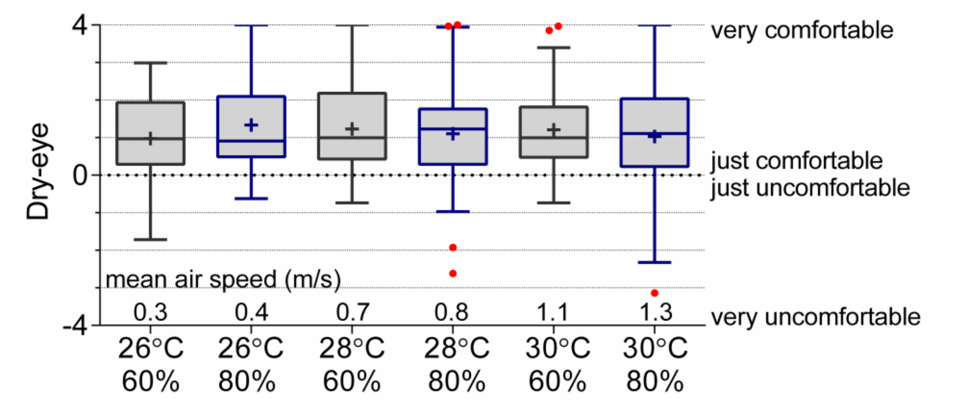


Figure 24 Eye-dryness discomfort. See Figure 17 definition of symbols.

2.4.3.8 The acceptability threshold

Figure 25 shows the percentage acceptable (PA) with overall thermal environment, air quality, air movement and humidity. The PA only decreased at 30ºC and 80% RH (32.7ºC ET*). The PA of air quality and air movement was above 80% for all test conditions. The PA of overall thermal environment and humidity was higher than 80% in all conditions except 30ºC and 80% RH, suggesting an upper limit at 30ºC and 60% RH for cooling with such horizontal fans.

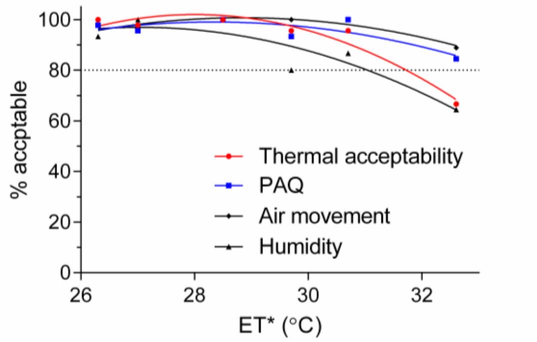


Figure 25 Percentage acceptable with overall thermal environment, PAQ, air movement and humidity

2.4.4 Discussion

2.4.4.1 Comparison with the ASHRAE comfort zone with elevated air movement

ASHRAE Standard 55 2010 specifies the air movement required to compensate for elevated temperatures above the standard's summer comfort envelope, for both with and without occupant control over air movement. With occupant control, ASHRAE 55 specifies a comfort zone with elevated air movement up to 1.2 m/s. To compare our results with this comfort zone, we calculated equal SET curves for 60% and 80% RH respectively based on the method provided by ASHRAE 55 (Figure 26). The air speeds chosen by the subjects for this study's temperature-humidity combinations are plotted on the same chart for comparison. It can be seen that, at 26°C and 28°C, the study's results are well within the ASHRAE comfort zone, however at 30°C/60RH, most subjects chose higher air speed than 1.2m/s. At 30°C /80% RH, most subjects again chose higher air speeds but this time some of them were not able to be comfortable. These results suggest that at temperatures near 28 and 30°C air speeds of 1.8 m/s are practical, but may not be sufficient for the combination of 30°C /80% RH.

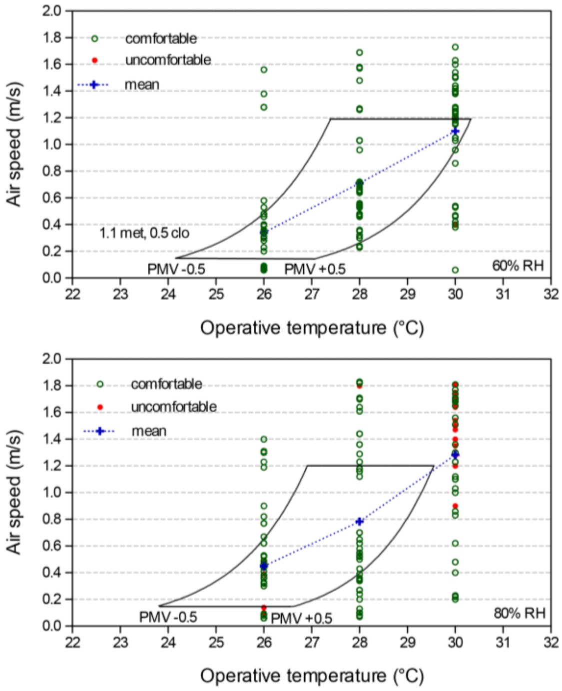


Fig. 12.

Figure 26 Comfortable temperature, RH and selected air speeds against the ASHRAE comfort zone at 60% and 80% RH with elevated air movement. The calculations were made with 0.5 clo and 1.1 met

2.4.4.2 Comparison with previous studies

The present results confirm previously findings that personally controlled air movement can maintain thermal comfort in warm-humid environments (Tanabe and Kimura 1994, Kubo et al. 1997). The mean TS votes in the current study were mostly within the neutral to slightly-warm range at 28ºC and 30ºC (see Fig.5a), while in study (Tanabe and Kimura 1994, Kubo et al. 1997), the subjects' TS were mostly on neutral to slightly cool range. The reasons may be that we used commercially available fans instead of fan-boxes producing air speed over a larger area of the body, and their subjects preferred slightly higher air speeds than our fans were capable of providing.

Arens et al. (2011) described the effect of air movement on restoring thermal comfort and PAQ at 28 ºC and 50% RH after breaks with high levels of activity. Our study extended their findings to higher humidity at 28ºC and 80% RH, and higher temperature at 30ºC and 60% RH, showing that air movement can restore comfort immediately after turning on the fan after a 12-step break, and that thermal comfort and PAQ were improved immediately after a 80-step break, and were restored within 5 minutes. Because air movement can improve comfort so quickly, it can be combined with more slowly acting systems, such as radiant cooling systems, to improve comfort and to save energy by reducing the number pre-cooling hours.

Another important finding of the current study is that PAQ was well maintained by elevated air movement at temperature-humidity combinations up to 30ºC and 80% RH, with less than 20% of the subjects dissatisfied with PAQ. This finding confirmed previous findings of Zhang et al. (2010), Arens et al. (2008, 2011) and Melikov et. al. (Melikov and Kaczmarczyk 2012, Melikov et al. 2012, 2008) and extended the positive effect of air movement on PAQ to higher temperatures and humidity.

The present study shows that the high air speeds chosen by the subjects didn't cause eye dryness discomfort in warm-humid environments. This confirmed previous findings by Melikov et al. (Melikov et al. 2011), which suggested that in warm-humid environment the water evaporation due to elevated air speed was low, and that giving subjects personal control over air speed might act to reduce eye dryness discomfort.

The mean power required to maintain a subject's comfort with the floor fan was only 10W at 30ºC 60% RH, a very energy-efficient way for providing comfort in buildings. In air-conditioned buildings, less cooling is required with the raised temperature and humidity set points permitted by elevated air movement. Air movement saves further energy by increasing the number of days in which natural ventilation or economizer cycles can be comfortably employed (Hoyt et al. 2009, Schiavon and Melikov 2008).

Human performance was not addressed in the current study. It is suggested by Wyon that the thermal state of the body determines arousal thus also performance (Wyon 1993), therefore warm temperatures may reduce performance by inducing warm sensations (Seppänen et al. 2006, Jensen et al. 2009). Since air movement can offset warm temperatures and maintains occupants' thermal state close to neutral, it may maintain performance close to that at neutral temperatures. This is supported by study (Zhang et al. 2010) and (Cui et al. 2013), in which isothermal airflow was found to maintain performance at temperatures up to 30ºC. It is worth to note that parameters other than temperature and air movement may also affect performance, such as humidity, air movement characteristics, ventilation rates, pollution levels, further studies should be done to address this.

Several limitations of the current study should be addressed. Only 16 subjects participated the current experiment. And the subjects were not acclimated to hot-humid climate. The chamber was ventilated with large amount of outdoor air, which is not always the case in real buildings. Further studies with more subjects who are acclimated to a hot-humid climate should be done to validate the findings of the current study, and the effects of different ventilation rates should be studied as well.

2.4.5 Conclusions

In conclusion,

1. Thermal comfort can be maintained up to 30ºC and 60% RH, and PAQ maintained up to 30ºC and 80% RH with personally controlled air movement, without causing discomfort from humidity, air movement or eye dryness;
2. Thermal comfort and PAQ was restored immediately after turning on the fans after the low-activity breaks, and comfort and PAQ were improved immediately after turning on the fans after the high activity breaks and restored within 5 minutes;
3. The 80% acceptable limit can be extended to 30ºC and 60% RH with personally controlled air movement;
4. The power required for maintaining comfort with the floor fan was less than 10W per person, which is a very efficient way to reduce energy consumption in hot-humid climate.

Acknowledgements

The work was supported by National Natural Science Foundation of China (Project No. 50978102, 50838003), California Air Resource Board (Project No. 10308), Center for the Built Environment, UC Berkeley, Program for New Century Excellent Talents in University, and National Key Technology R&D Program of China (Project No. 2011BAJ01B01). The authors would like to thank Professor Shin-ichi Tanabe for kindly helping us on purchasing and shipping the floor fans.

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2.5 Project 3: Human comfort and perceived air quality with an oscillating ceiling fan

(this paper was published at CLIMA 2013 in Prague, June)

Abstract

Moving air cools the human body. In warm environments, room fans can provide comfort using substantially less energy than air-conditioning. The savings are greater if the fans make it possible to successfully condition the building with natural ventilation or evaporative cooling systems, instead of chillers. Although there are many laboratory studies of comfort using desk fans and personalized fans, tests for ceiling fans are rare, mainly in early studies from the 1980s. This study examines the cooling effect of a low-wattage ceiling fan on occupants when air comes from different directions with different speeds. We conducted 96 human subject tests in an environmental chamber. Sixteen college students each experienced 6 air movement conditions: two different air speeds and three different air directions between fan and subject: from front, side, or right above the head (total eleven configurations). The difference in thermal comfort and thermal sensation generated by fixed and oscillating fans was also investigated. The temperature and humidity conditions for the tests were 28 °C and 50% RH.

It was found that the majority of subject (70%) perceive the thermal environment without fans comfortable. This number rise to 100% for some configuration with fans. Our subject found that the oscillating air movement had no effect in terms of improved thermal comfort or thermal sensation, but it greatly improves their air quality perception. The subjects did not report any dry eyes discomfort for any of the eleven configurations.

Keywords - Integrated ceiling fan; Air movement; Thermal comfort; Thermal sensation; Oscillating airflow.

2.5.1 Introduction

Compressor-based air-conditioning is the second largest consumer of electricity in US commercial buildings, exceeded only by electrical lighting (141 billion kWh vs. 393 billion kWh), (US Energy Information Administration 2009). In office buildings compressor-based cooling constitutes roughly 15% of total electricity consumption.

The amount of energy used to condition commercial buildings is increased by the tendency of building operators to maintain buildings at too-low ambient temperatures during warm seasons (Mendell and Mirer 2009). The overcooling was found to significantly reduce occupant comfort and even caused health symptoms (Moezzi and Goins 2010). There are a number of reasons for summer overcooling, but an important one is the insufficient air movement around occupants in conventional sealed office buildings (Hoyt et al. 2009, Zhang et al. 2007). Sealed designs with low air movement consistently show lower occupant satisfaction than offices with operable windows (Fisk et al. 1993, Brager and Baker 2009). Although buildings with operable windows tend to have warmer interior temperatures than sealed buildings, the modest increase in indoor air movement that they provide gives them comfort ratings superior to those of sealed buildings.

This suggests one can reduce cooling energy demand by allowing a building to float within an expanded indoor temperature range while maintaining the occupants' thermal comfort by providing air movement using fans. Fans of very low wattage (as low as 3W) have been shown to yield the equivalent of 3K (6ºF) offset of air temperature within an individual workstation (Zhang et al. 2010). Buildings employing such fan cooling promise substantial savings in their HVAC energy, more than 30% below that of conventionally conditioned buildings (Zhang et al. 2011, Kimura 1993, Hoyt et al. 2009). The energy savings may be even greater if the warmer setpoint temperatures enable the primary cooling source to be switched from a compressor-based system to one of the more efficient and lower-power approaches, such as natural ventilation or evaporative cooling. Room fans may be readily applied in both new and retrofit designs since they can be easily installed and the savings can be achieved by only changing HVAC system setpoints.

Since the1960's, the use of air movement to maintain thermal comfort in warm conditions was impeded by strict limits to air movements in thermal comfort standards (e.g. ASHRAE 55-1992, 2004 and ISO 7730). This has led to almost no innovation in designing interiors to use air movement, and only modest innovation in industrial products that move air. However, in recent years, the air movement requirements in these standards have been revised to permit higher indoor air speeds, following the results of extensive field and laboratory studies (ASHRAE 55-2010, Arens et al. 2009). The revisions pave the way for the above mentioned 30% reduction in HVAC, and also for significant reduction in their peak power demand. They also enable more individual environmental control and the good levels of occupant comfort already observed in naturally ventilated buildings with fans in warm and humid climates (Zhang et al. 2011, de Dear et al. 1991, Busch 1992, Kwok 1998, Zhang et al. 2010, Cândido et al. 2010).

The ASHRAE database of comfort field studies (Hoyt et al. 2009, Zhang et al. 2007, de Dear 1998, Toftum 2004) shows that more occupants prefer more air movement, while very few prefer less, when environments are in the range of slightly cool to slightly warm. The challenge becomes how to implement indoor air movement devices within the interior space, so that they are: highly energy efficient, comfortable and acceptable to occupants, visually attractive to building management and designers, and straightforward to design. For this purpose we integrated a head of an oscillating floor fan into an acoustical ceiling panel, and performed this study to characterize the thermal responses of occupants under different subject-fan positions, different air velocities, and fixed vs. oscillating fan settings..

2.5.2 Method

The experiments were carried out at the Center for the Built Environment (CBE), University of California at Berkeley, between August and September 2012.

2.5.2.1 Chamber setup and the ceiling fan

We set up 4 workstations in the chamber (Figure 27) with two fans. One fan was set in a way to provide airflow toward heads and faces of the two subjects' (we call it "front" in this paper). The other fan provided airflow towards sides of heads of the two subjects' (we call it "side" in this paper).

For this study two prototypes of oscillating ceiling fan were made (circled in red in Figure 27). The fan motor and propeller are commercially available, while the structure to integrate the fan into the ceiling panels was custom fabricated. These fans were selected due to their excellent energy efficiency, with an energy consumption that ranges from 2 W to 15 W, and because they are extremely quiet.

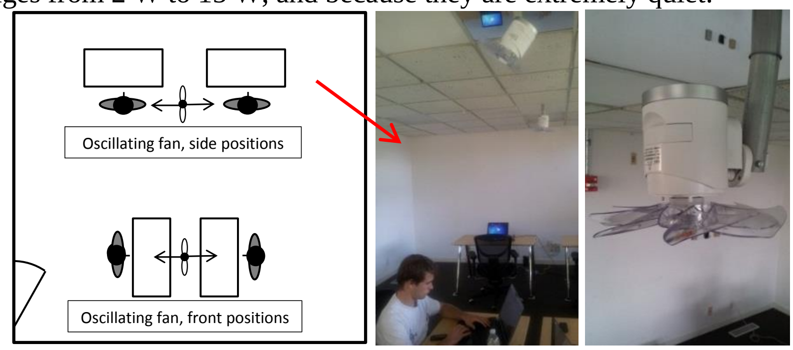


Figure 27 Chamber set up and ceiling fan prototype

The fan power can be set among 7 levels. For this study we selected level 2 and level 3. The fan oscillation period is 28 seconds.

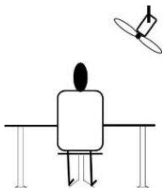
2.5.2.2 Subjects and test conditions

Human subjects were tested to evaluate comfort for warm conditions (28±0.3°C). The relative humidity of the chamber was kept at 50% ±1%.

Four workstations were installed in the Controlled Environmental Chamber at UC Berkeley so that four subjects could be tested at the same time. The chamber size is 5.5 x 5.5 m (18 x 18 ft), with windows on two sides, south and west. The windows are well shaded by fixed external shades. The windows temperature was controlled by a dedicated air system. The room air temperature is controlled by 8 floor grille diffusers, and the air is exhausted through a 0.6 x 0.6 m (2 x 2 ft) ceiling return grille.

Several configuration were studied, the subjects experienced two different air velocity and three different air directions, plus the oscillation feature. A schematic representation of all the different configurations and their configuration codes used in the analysis is presented in Table 10.

Table 10 Test configurations

Subject position	Fan mode	Power level [W]	Configuration code
 Front	Fixed	2	2 Fixed Front
		3	3 Fixed Front
	Oscillating	2	2 Oscillating Front
		3	3 Oscillating Front
 Side	Fixed	2	2 Fixed Side
		3	3 Fixed Side
	Oscillating	2	2 Oscillating Side
		3	3 Oscillating Side
 Below	Fixed	2	2 Fixed Below
		3	3 Fixed Below

Sixteen subjects (8 females and 8 males) participated in each of the ten test conditions, plus one test without fans, for a total of 112 tests. The tests without fan (in this paper called no-fan) provided reference conditions for comparison with the tests with the fans.

Subjects were asked to wear summer clothing (0.5 clo), and their clothing were checked before every test to guarantee the right clo value.

2.5.2.3 Schedule for the tests

Each test took one hour and forty-five minutes. At the beginning of the test, the subjects sat for 15 minutes in a room outside the chamber (≈ 21 °C), to stabilize their metabolic levels. After these 15 minutes the subjects moved into the environmental chamber and sat at the workstations with the fans on. The remaining part of the test was divided into three parts by two five-minute breaks. The first part, 20 minute long, was used to let subjects' body adapt to the temperature. The second and the third part are 30 minutes long. The results from part 2 and 3 are used for the analysis. During the breaks the subjects were asked to stand up, and in the middle of each break period, they took 12 vertical steps on a 22cm tall step stool. This was to simulate activity levels when occupants are away from their desks in a real office. After the second break we asked the 2 subjects experiencing "front" airflow to switch workstations with the 2 subjects experiencing "side" airflow, to test different fan oscillating directions (Figure 27).

The survey questions automatically appeared on subjects' computer screens based on pre-set schedules (Figure 28).

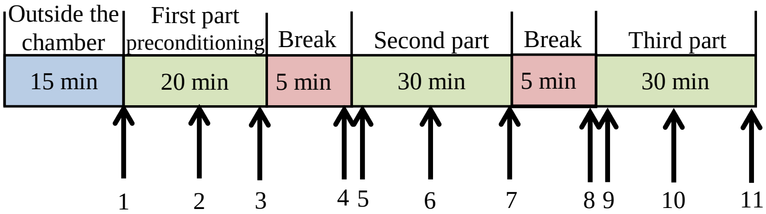


Figure 28 Test schedule. Arrows indicate times when the survey are administrated

2.5.2.4 Survey questions

In addition to temperature satisfaction, thermal sensation, preferred thermal sensation, and thermal comfort (for the whole body and several body parts separately), the survey questionnaire also included questions related with the use of fans: air movement acceptability, air movement preference, and dry-eyes discomfort. We also included two questions about air quality acceptability and air freshness, to investigate the effect of air movement on perceived air quality.

Two survey question scales are continuous (Figure 29 shows two examples).

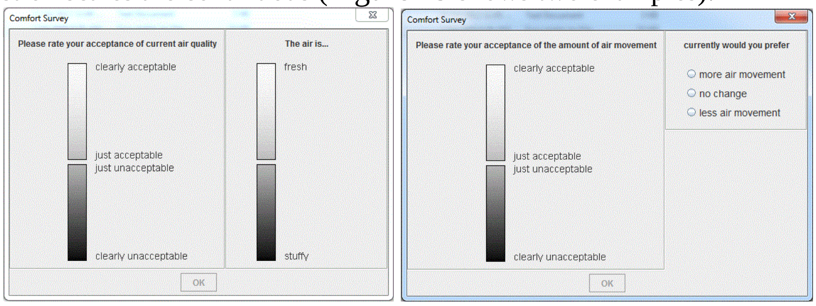


Figure 29 Two examples of survey questions

2.5.2.5 Measurements

Room air temperature and humidity were measured at 1.1 m height.

Air velocity was measured for all the configurations in 27 points in the area where the subjects were supposed to be sat at 3 heights. In this paper only the air velocity values at three heights (1.1 m, 0.6 m, and 0.1m), and at the location 20 cm from the center of a desk, are reported (see Table 11). These three points has been selected as the most representative of the airflow experienced by the subjects for the six different configurations with the fixed fan. For the configurations with the oscillating fan, the maximum air velocities for the three points are comparable with value measured with the fixed fan, but there is a transitory phenomenon due to the fan oscillation. In Figure 30 it is reported an example of measured data for the configuration "3 Oscillating Front". Velocity sensors at 0.6m and 0.1 m were shade from the air movement by the desk, so over time the air velocity was always low.

Table 11 Measured air velocities and standard deviation

Configuration code	Mean value [m/s]/SD[m/s] 1.1 m	0.6 m	0.1 m	Energy (Watts)
2 Fix Front	0.68/0.16	0.51/0.11	0.24/0.06	2
3 Fix Front	0.88/0.13	0.62/0.12	0.28/0.07	3
2 Fix Side	0.70/0.21	0.76/0.11	0.34/0.09	2
3 Fix Side	0.81/0.29	0.79/0.48	0.35/0.19	3
2 Fix Below	0.66/0.20	0.54/0.25	0.24/0.12	2
3 Fix Below	0.88/0.49	0.69/0.27	0.31/0.11	3

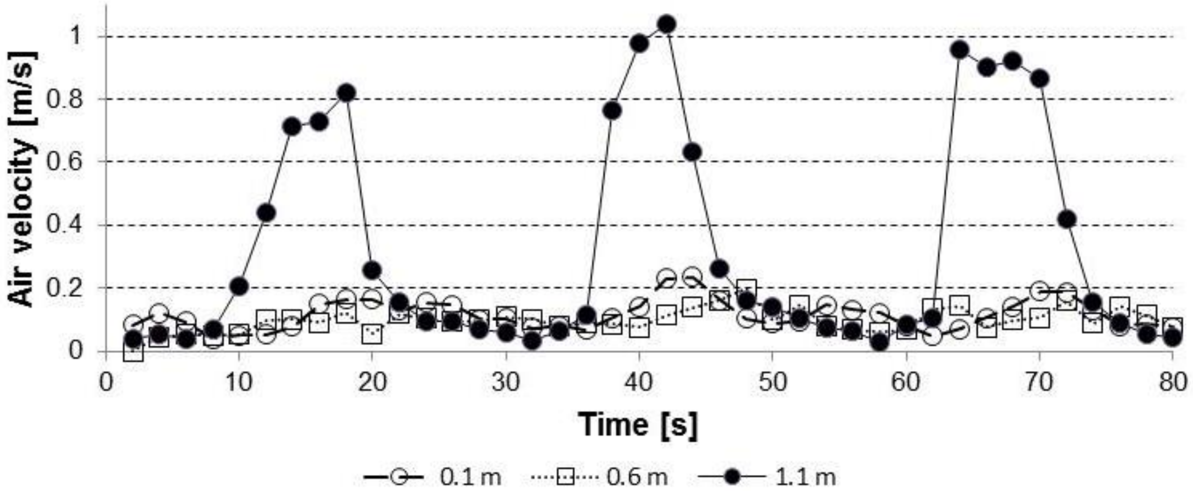


Figure 30 Measured air velocity for configuration "3 Oscillating Front"

Air speed was measured with omnidirectional hotwire anemometers, with a response time of 2s and an accuracy of 0.02m/s±1.5% of reading

2.5.3 Results

This paper focuses on stable conditions, analyzing subjects' responses to the number seven and eleven surveys (Figure 28), and leaving the results of the other surveys for future analysis.

2.5.3.1 Whole-body thermal sensation and thermal comfort

All the results for the configurations with fans were compared against the reference 'no-fan' condition. Statistical analysis was performed with a non-parametric method called a permutation test, using the program R (R Development Core Team 2011). In the graphs the symbol "*" represents a statistically significant difference ($p<0.05$).

In Figure 31 the whole body thermal sensations are presented for the eleven configurations. It can be seen that are no differences between "no-fan" and the "oscillating fan"-configurations. The differences are statistically significant between "no-fan" and all the six configurations with "fixed-fan"s.

Similar results were obtained for the whole body thermal comfort (Figure 32). From these two charts is clear that the amount of air movement generated with the oscillating fan was not enough to modify subjects' thermal sensation, or to affect their thermal comfort.

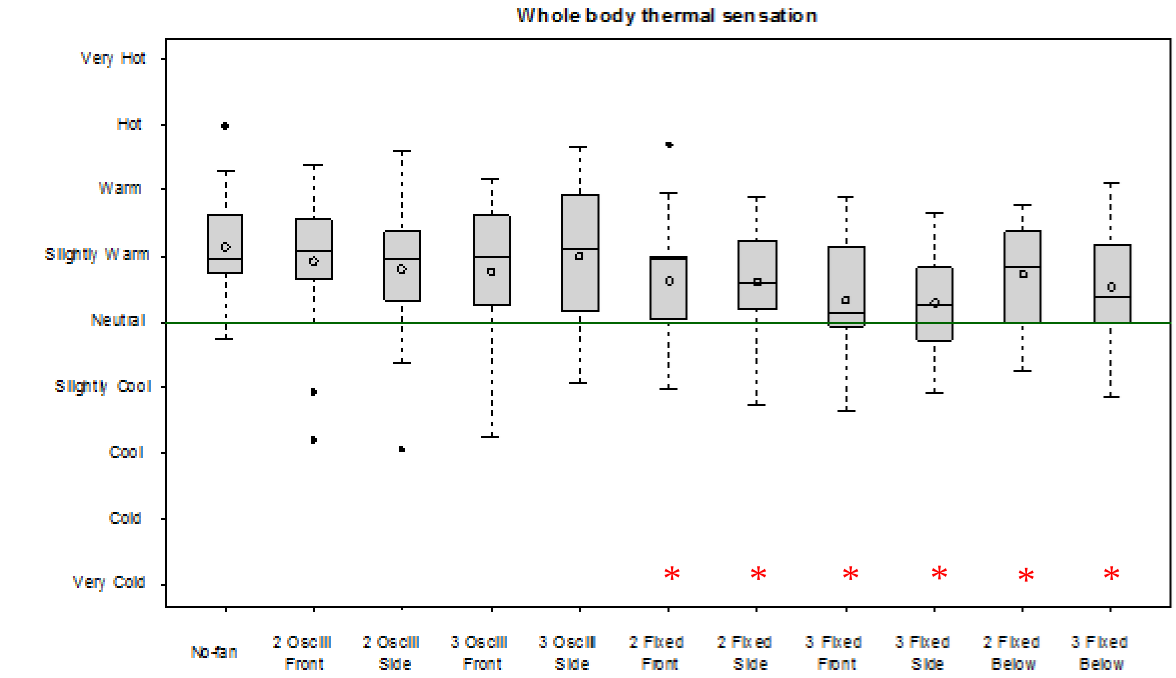


Figure 31 Whole body thermal sensation

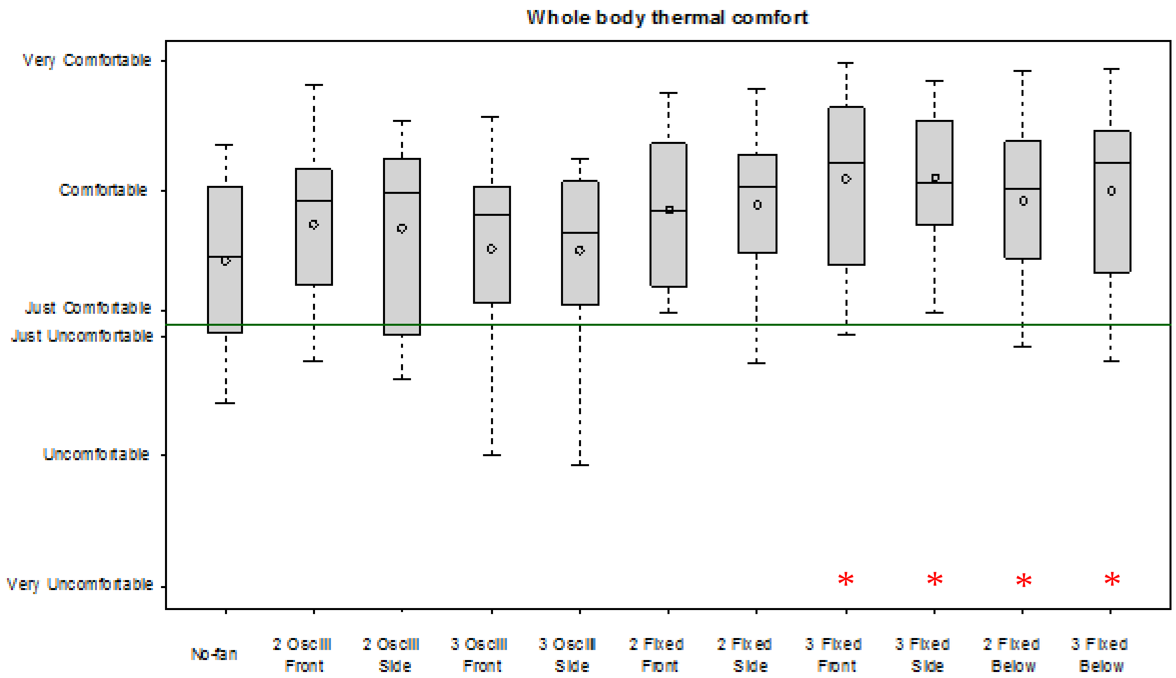


Figure 32 Whole body thermal comfort

2.5.3.2 Thermal acceptability

During the tests the subjects were asked to rate the acceptability of the thermal environment. The results are presented in **Error! Reference source not found.**. Although some configurations of the oscillating fan showed an improvement over the no-fan reference case in terms of thermal acceptability, the result was not statistically significant. This may be related to the nature of the air movement generated by the oscillating fan, but also perhaps to the relatively small sample size. It is possible that the number of subject used for this work (16) was not enough to statistically show improvements caused by the oscillating fan.

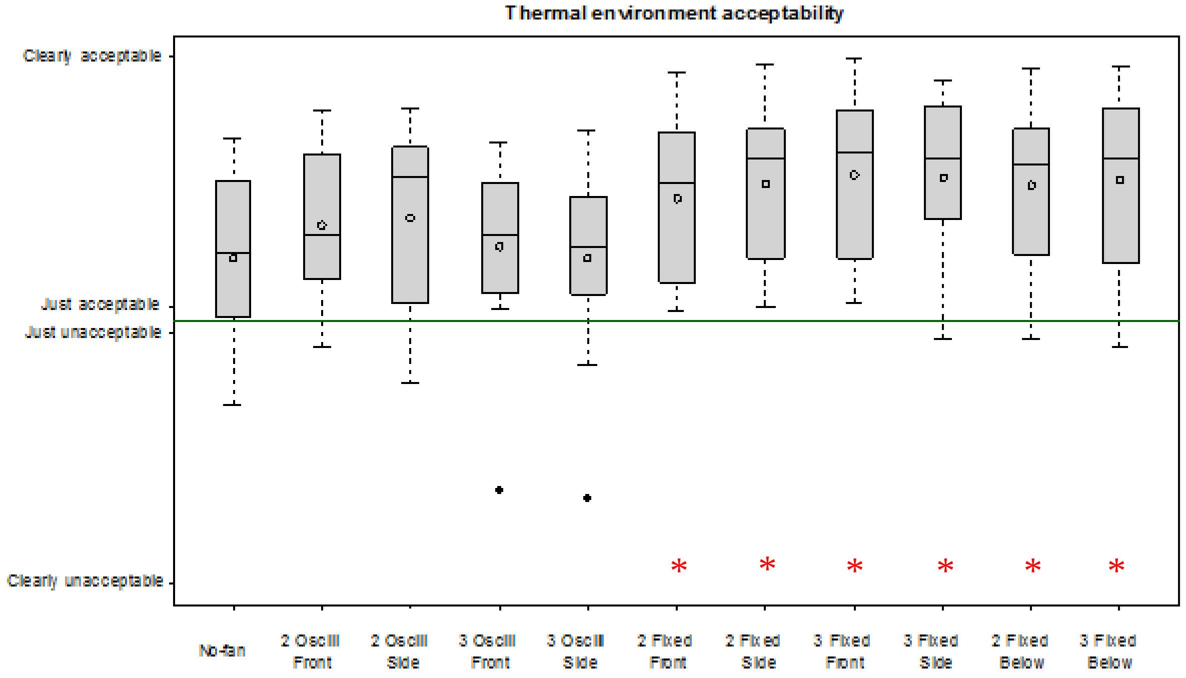


Figure 33 Thermal acceptability

2.5.3.3 Whole body preferred thermal sensation

In line with a previous work (Pasut et al. 2013) the preferred thermal sensation was not affected by the thermal sensation. Although there are some differences in terms of width of the distributions, the statistical analysis did not evidence any significant difference.

Figure 34 Whole body preferred thermal sensation

2.5.3.4 Face thermal sensation

The differences in face thermal sensation were statistically significant only for the configurations “3 Fixed Front” and “3 Fixed Side”. Differences for the other fixed-fan configurations had a p-value around 0.15. This result is not enough to say with confidence that there is a significant difference (p = 0.05), but can be an indication that with a bigger sample size the difference would have been statistically significant.

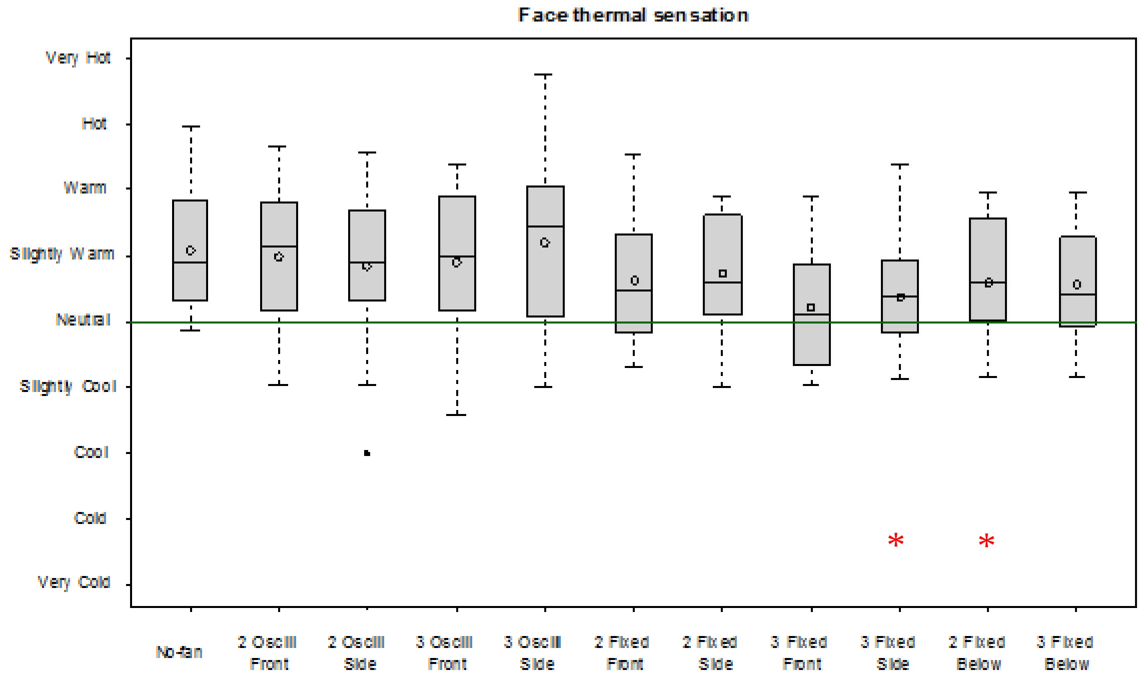


Figure 35 Face thermal sensation

2.5.3.5 Perceived air freshness and air quality acceptability

Similarly to what Melikov and Kacmarczyk 2008 did to investigate the perceived air quality, we used two questions in the survey: "Please rate your acceptance of current air quality" and "The air is…", with a scale for this last that range from fresh to stuffy (see **Error! Reference source not found.**). The acceptability scale was a continuous scale (see Figure 29) (EN 15251 2007), which ranged from "clearly acceptable" (+1) to "clearly unacceptable" (-1), and was split in the middle with two labels "just acceptable" and "just unacceptable".

In Figure 36 and Figure 37the results for the "perceived air freshness" and "air quality acceptability" are reported respectively. The results for the two questions are very similar. Almost every configuration presents a statistically significant difference compared to the configuration "No-fan".

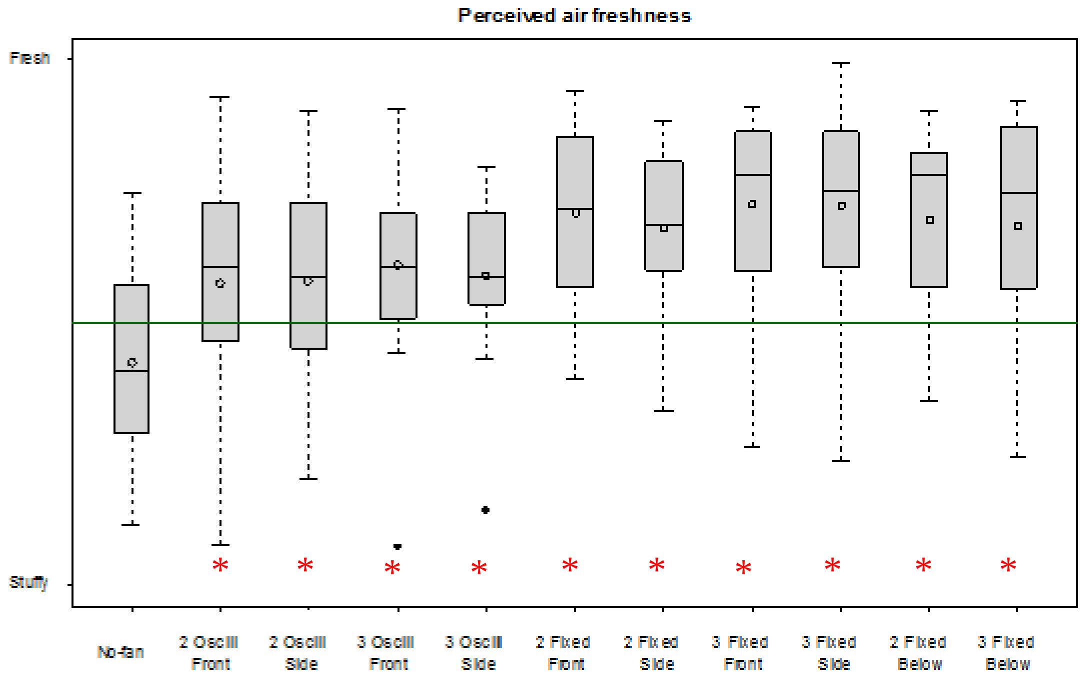


Figure 36 Perceived air freshness

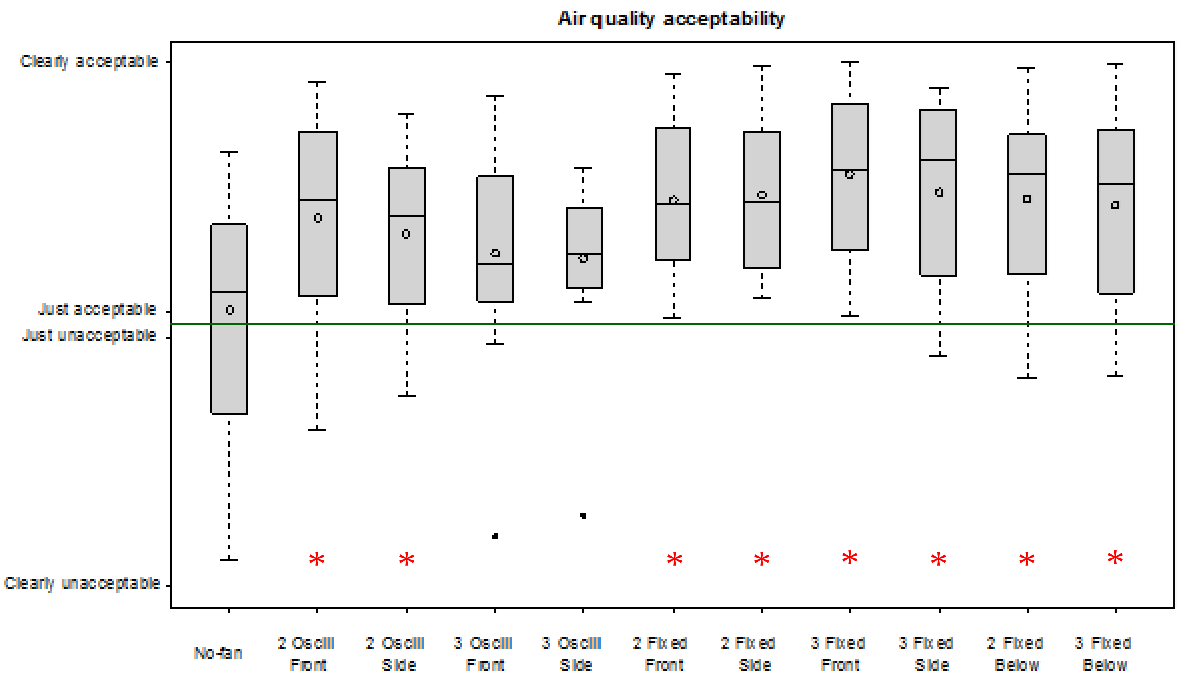


Figure 37 Air quality acceptability

2.5.3.6 Air Movement acceptability and preference

Subjects were asked about the acceptability of the air movement, and their preferences. They had to choose among three options: 1) less air movement; 2) no change; 3) more air movement. Results are presented in Table 12.

As expected, for the configuration without fans, the majority of the subjects (94%) wanted more air movement. For the oscillating fan, more people wanted more air movement. For the fixed fan, more people wanted no change, and just few wanted less air movement.

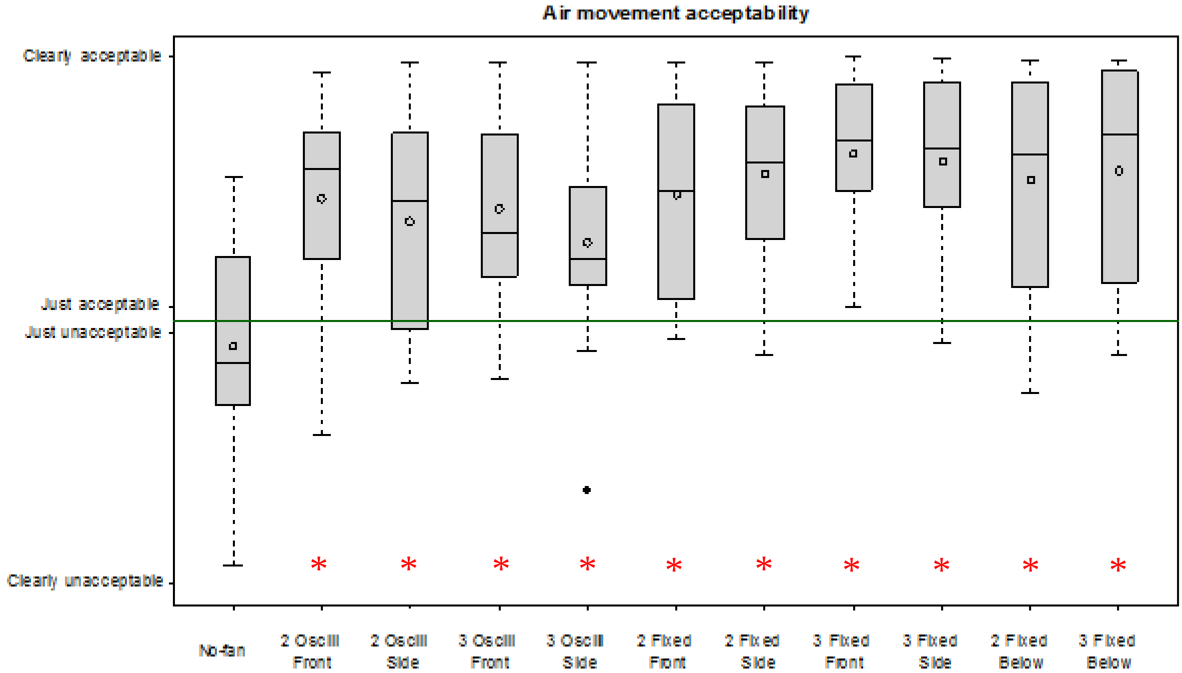


Figure 38 Air movement acceptability

Table 12 Air movement preferences

Fan Configuration	Less air movement [%]	No change [%]	More air movement [%]
no-fan	0	6	94
2 Oscillating Front	0	56	44
2 Oscillating Side	0	44	56
3 Oscillating Front	0	44	56
3 Oscillating Side	0	38	63
2 Fixed Front	13	50	38
3 Fixed Front	0	75	25
2 Fixed Side	0	56	44
3 Fixed Side	19	56	25
2 Above	6	50	44
3 Above	13	56	31

2.5.3.7 Dry-Eye Discomfort

Subjects were asked a question about dry-eye discomfort: " Do you currently feel any discomfort due to dry eyes?". The percentage of subjects who experienced dry-eye discomfort was minimal, ranging from 0% to 10% (Figure 39). The majority of the subjects who reported dry-eye discomfort placed their vote very close to the "just uncomfortable" line. The statistical analysis proved that there is no significant difference among the eleven configurations.

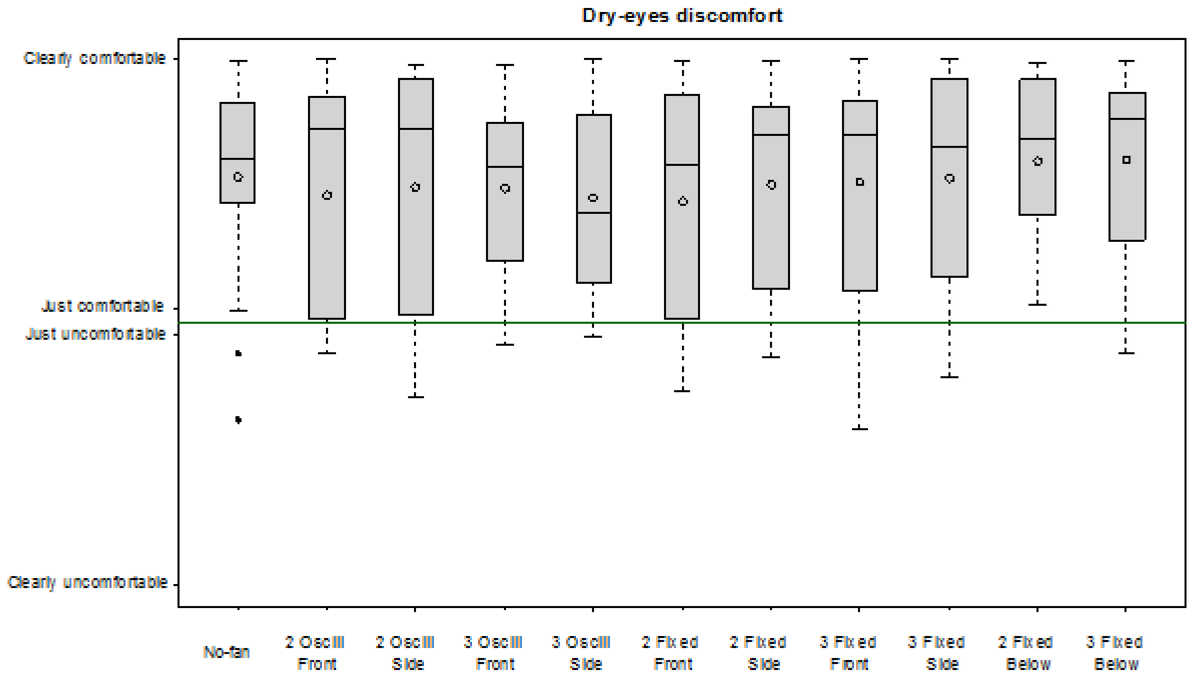


Figure 39 Dry-eyes discomfort.

2.5.4 Discussion

Oscillating mode: Compared to the no-fan reference configuration, neither the whole-body thermal sensation nor the whole-body thermal comfort was significantly affected by the oscillating fan. This negative result seems to be a result of the long oscillation period, especially the 15-second interval during which the airspeed at the occupant is effectively zero. This interval is long compared to the 10 seconds when the occupant received appreciable air movement. This type of intermittent air flow has not been addressed before. Previous studies of the comfort and sensation effects of fluctuating airflow were done by Fanger and Pedersen 1977, Tanabe and Kimura 1994, Arens et al. 1998, and Hua at al.2012.

Fanger and Pedersen 1977 studied draft discomfort under turbulence and found the maximum discomfort at 0.5 Hz. Tanabe and Kimura 1994 tested different patterns of air movement, concluding that fluctuating air movement has a stronger effect on subjects' thermal sensation than constant air movement. As with Fanger, the authors reported an increased number of subjects feeling draft under fluctuating air movement (reaching peaks of 2 m/s) compared to those exposed to constant air.

Arens at al. 1998 points in another direction. In this study, the constant air speed provided more effective cooling compared to fluctuating air speed. The authors pointed out that the result was related to the specific fan used for the constant air movement. The constant speed mode of the fan tested in Arens' work was not actually constant, producing a power spectrum peaking between 0.7 and 1 Hz, compared to 0.2-0.4Hz for the fluctuating mode. Previous work published by Ring at al. 1993 shows that cold receptors in the skin have a peak sensitivity around 1 or 2 Hz, which matches the power spectrum peak of the 'constant' fan. The constant power spectrum more closely resembled outdoor rural conditions, while the fluctuating one resembled a built-up urban spectrum.

The study of Hua (Hua et al. 2012) studied a simulated natural wind (SNW) and a constant mechanical wind (CMW). on subjects' thermal comfort and thermal sensation, under two different environmental conditions, 28°C and 30°C. They found no statistically significant difference between thermal sensation vote of the SNW and CMW for either environmental condition, and a very small improvement in terms of thermal comfort vote only at 30°C.

The airflows used in the above-mentioned studies fluctuated around an average air velocity and, differently from our study, never reached zero. The subjects in the above-mentioned studies were always exposed to a certain minimum amount of air movement. This may explain the different conclusions of our work compared to the literature.

Fixed mode: In contrast with the results for the oscillating fan, the configurations with the fixed fan showed an improved thermal comfort and a cooler whole body thermal sensation.

The remarkably low energy consumption of the new generation of DC fans emphasizes the role that these devices can play in saving energy in buildings. The results of this study show that a fan consuming 2-3 W can maintain comfort for an occupant in a warm indoor environment. Fans can improve resilience to future climate change in both existing and new buildings. They can be deployed to assist existing air conditioning (AC) systems or, in climates characterized by moderate temperatures, to help avoid the installation of new AC devices (Lomas et al. 2012).

In terms of perceived air quality, almost all the fan configurations performed better than the still-air reference configuration. This result is in line with the literature (Melikov and Kaczmarczyk 2008, Arens et al 2008, Melikov et al. 2005). However, the literature ascribes the improvements to whole-body thermal sensation, face thermal sensation, or cooling effect in the respiratory tract (Fang et al. 1998, Berglund and Cain 1989, Toftum 1998). In the case of the oscillating fan in this study, there is a notable improvement in terms of air freshness and air quality acceptability, but no difference in thermal sensation compared to the base case. This suggests that a small amount of intermittent air movement, though insufficient to affect the subjects' thermal sensation, may improve their perception of the air quality. There can be two explanations to this phenomenon:

a) The amount of air movement is enough to disrupt the thermal plume that envelops the face. Plume disruption occurs at approximately 0.3m/s airspeed (Melikov and Kaczmarczyk 2008). Disruption dilutes the bio-effluents and other pollutants picked up in nose and eyes, which the subjects may be able to perceive;

b) There may be some psychological sense of the outdoors and fresh air when feeling air movement. More research needs to be done to examine this effect.

A separate statistical analysis was done to verify whether there is any difference between front and side configurations. No statistical difference was found.

In terms of "air movement preference" the configuration "3 fixed front" performed the best (biggest "no change" population, 75%). Under this condition, for an average air velocity around 0.9 m/s, nobody asked for less air movement.

The combinations of air velocities, air directions, temperature and humidity tested in this study did not cause any appreciable dry-eye discomfort. Fans blowing air from the ceiling at about 0.9 m/s toward the front, the side, or directly above the subjects' heads did not cause dry-eye discomfort.

2.5.5 Conclusions

1) The oscillating fan tested in this study had no statistically significant effect on subjects' thermal comfort or thermal sensation at 28ºC. Further study should examine shorter recurrence intervals. There is a heightened perception of dynamic change in airspeed cooling, and at some shorter time interval this perception should outweigh the reduction of mean airspeed caused by a fan in oscillating mode.
2) Under the tested conditions, a fixed-fan that directs air over a human body at a velocity between 0.8 and 0.9 m/s has a positive and statistically significant effect on users' thermal comfort and thermal sensation.
3) Air quality acceptability is improved by the air movement, even if the amount of air movement is not enough to improve subjects' thermal comfort and thermal sensation.
4) For the ceiling fans studied in this work, an air velocity of 0.9 m/s directed on subjects' face did not cause any dry-eyes discomfort.

Acknowledgement

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2.6 Project 4: Comparison with whole-body heat balance models under airflow (desk fans)

(this paper is under review by Building and Environments)

Abstract
In ASHRAE Standard 55-2010, the comfort effects of elevated air movement are evaluated using the SET index as computed by the Gagge 2-Node model of whole-body heat balance. Air movement in reality has many forms, which might create heat flows and thermal sensations that cannot be accurately predicted by a simple whole-body model. This paper addresses two of such potential inaccuracies: 1) indoor airflows may affect only a portion of the body surface (e.g., above desktop), and the affected body surface might be variably nude (e.g. face) or clothed, 2) the turbulence intensity (TI) in some typical airstreams (e.g., those created by fans) might have a different impact on heat transfer than the TI implicit in 2-Node's single convective heat transfer coefficient. For both these issues, can a whole-body index like SET represent such a wide range of possible exposures to airflow?

Measurements of thermal sensation were obtained from human subjects using face-level fans in warm environments. Previous laboratory studies of a range of airstream sources were also analyzed. The effects of turbulence intensity were examined with manikin tests.

The results show that indices derived from the 2-Node model of whole-body heat balance are effective at predicting thermal sensation under most non-uniform air movement. In contrast, the PMV index underestimates cooling in warm conditions. Turbulence increases the cooling effect of air movement, but by amounts that might be neglected for most design purposes.

Keywords
SET, 2-Node, Comfort model, Air movement, Thermal sensation, Turbulence intensity

2.6.1 Introduction

Air movement has a significant cooling effect, increasing the acceptable range of indoor temperatures (Toftum 2004, Fountain and Arens 1993, Zhou et al. 2006, Atnsley 2008). ASHRAE Standard 55-2010 uses the model PMV to determine comfortable temperatures under still air, and uses the SET (standard effective temperature) index as the basis for extending this still-air comfort zone under elevated air speeds (Arens et al. 2009).

The SET index is derived from Gagge's 2-Node model, which was introduced in 1970 (Gagge et al. 1972). The model considers a human as two concentric thermal compartments representing the skin and core of the body, producing a minute-by-minute simulation of the status of the human thermoregulatory system (Gerglund and Stolwijk 1978, Gagge et al. 1986). The model predicts skin temperature, skin wettedness, and thermal status for any combination of environmental and personal variables, including those outside the neutral range, and can be used to find the loci of environmental conditions that produce equal levels of heat loss. Therefore it appears reasonable to use SET as an index to evaluate cooling effect of elevated air movement.

However the environmental surroundings of a simplified model like 2-Node are assumed to be uniform. It is a 'whole-body' model, in which the entire body surface is represented by one average heat transfer coefficient, unlike a 'multi-segmented model', in which body segments are treated individually, and which are necessarily more complex. Recognizing the whole-body nature of SET, ASHRAE Standard 55 specifies that 'average air speed' be used as input to the model, which for sedentary occupants is defined as an average of airspeed measurements at 0.1, 0.6, and 1.1m above the floor.

There are many ways that air movement may be distributed across the body, uniform or non-uniform. The airflow from fans typically reaches only parts of the body surface. The airspeed across these exposed parts is higher than the average airspeed, and the physical and psychological effects may be sensitive to this difference.

In addition, whole-body models use an average clothing resistance value for the whole body surface (Arens et al. 1986). But the airflow from fans passes over both clothed (e.g., trunk) or unclothed (e.g., face) portions of the body. While the heat loss from clothed and nude surfaces might be linearly related to clothing resistance, the psychological sensitivity may not be.

Finally, a given airspeed's transient flow characteristics (intensity and scale of turbulence) are likely to be different from the fixed level of turbulence assumed in the 2-Node model. Nishi and Gagge (Nishi and Gagge 1970) experimentally developed the model's forced-convection equation by having subjects walk through still air at a fixed speed. Turbulence was not measured in their experiment, and is not an input variable to 2-Node. In reality, however, turbulence from different air movement sources will differ, and will affect heat transfer. Mayer's research with a manikin head (Mayer 1984) showed an increase in heat transfer with increasing turbulence intensity (TI) when TI is above 40%. It would be desirable to know whether the differences in turbulence intensity found in typical air movement sources like ceiling or desk fans significantly affect body cooling, and whether 2-Node predictions using only one implied turbulence level accurately predict these differences.

This paper examines each of the above issues as follows:

1) In a study in which fans provided non-uniform frontal air flow to the upper body, subjects' actual thermal sensation votes (TSV) could be compared to SET values calculated for the experiment's test environmental conditions. The calculations were done in two ways: using only the air speed around the face, and using the average air speed of three heights next to the subjects: 0.1m, 0.6m, and at face level (1.1m). If the calculated indices represent the subject's responses well even in under non-uniform flow and non-uniform clothing coverage, then the general use of a whole-body model is supported.

2) A number of published human subject experiments provide TSV results for other types of airflow sources and exposures of the body surface. These experiments involved airflow exposures to a variety of body parts that have differing thermal sensitivity (e.g. face vs. chest vs. back) (Arent et al. 2006). Differences in subjects' thermal sensations should appear, even at the same air velocity. The regression relationship of TSV against SET value is therefore likely to differ among various types and extent of exposure.

3) Finally, we determined the turbulence intensity that had occurred in the Nishi/Gagge experiment (Nishi and Gagge 1970), since this TI is inherent in the 2-Node convection algorithm. We repeated the conditions of the Nishi and Gagge experiment and directly measured the TI. The heat loss from this TI was then measured with a manikin, and compared with heat losses from a range of TI values occurring from fans and other indoor sources. A relationship describing the difference in heat loss and thermal comfort could be developed from this.

2.6.2 Methods

2.6.2.1 Test of cooling under non-uniform air flow

A subjective experiment was conducted in a climate chamber in Tsinghua University in Beijing. 30 subjects took part in the experiment, experiencing warm environments with fan-generated frontal air flows to the face and upper body. They wore summer clothing of 0.57 clo. The temperature ranged from 28°C to 34°C with relative humidity 40%-50%. At each temperature, air speed ranged from 0.6 m/s to 2 m/s. All the fans were placed in front of the subjects at a horizontal distance of 0.6m and a vertical distance of 0.6m from the desk (Figure 40). The experiments were designed orthogonally with different temperatures and air speeds. Each experiment lasted about 2 hours at a fixed temperature. At the beginning, the subjects were given no air flow for 45 min, and then they voted their thermal sensation. After that, the fans provided air flows with four randomly sequenced speeds in turn, with each air flow lasting for 15 min, for a total duration of 60 min. Subjects' TSV were collected at the end of each 15-min period, using the ASHRAE seven-point thermal sensation scale (-3 cold, -2 cool, -1 slightly cool, 0 neutral, +1 slightly warm, +2 warm, +3 hot). Using the environmental parameters of each experiment, SET values for different conditions were calculated using the SET model and compared with the subjects' thermal sensation votes. The SET calculations were done using air speed measured in front of the face, 1.1m above the floor and 5 cm from the nose. Then they were repeated using the average speed of the three heights (0.1, 0.6, and 1.1m) to represent the whole-body air speed. Further details about this experiment are described in (Huang et al. 2013).

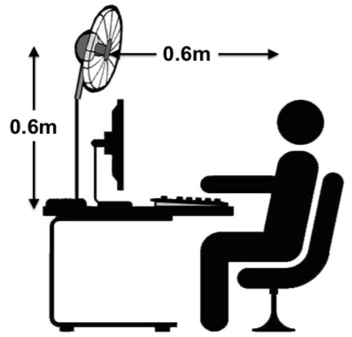

Figure 40 The relative position with the subject and fan

2.6.2.2 Studies of other air flow sources and exposure types

Error! Reference source not found.shows published studies from which subjects' thermal sensation values could be obtained, and SET calculated. A variety of different air-movement devices are represented in the studies. We have categorized them as: ceiling fan, desk fan, tower fan, wind box, and nozzle. Subjects' exposures were to air flow on their head, back, and face/chest. SET could be calculated using the reported test conditions. The results are aggregated and compared with those of the fan study described above.

Table 13 Studies of air movement using different air movement devices

Researcher	Location	RH (%)	Local control	Air movement supply device	Body part directly exposed to the air movement
Mclntyre (1978)	UK	50	Yes	Ceiling fan	Head
Zhai (2013)	USA	60/80	No	Ceiling fan	Head
Fountain (1994)	USA	50	Yes	Desk fan	Face and chest
Atthajariyakul (2008)	Thailand	45-80	No	Desk fan	Face and chest
Chow (2010)	Hong Kong	50	No	Tower fan	Back
Tanabe (1994)	Japan	50	Yes	Wind box	Back
Kubo (1997)	Japan	50	Yes	Wind box	Front
Gong (2006)	Singapore	40-55	No	Nozzle	Face
Yang (2010)	Singapore	-	No	Nozzle	Head
Zhang (2010)	USA	50	Yes	Nozzle	Head

2.6.2.3 Test of the impact of turbulence intensity (TI)

A thermal manikin (Figure 41) was used to measure the convective heat transfer coefficients for air flows of different turbulence intensity, following the method described in (deDear et al. 1997). Analysis was done for the manikin's head alone, without hair. The manikin consists of 20 thermal segments, electrically heated at the surface to simulate metabolic heat output, and surface temperature is measured for the entire segment surface. Heat losses and surface temperatures were recorded in 30s intervals. Mean values were calculated for 1-hour measurements after the manikin reached steady state for each variant of the experimental conditions. Each test was repeated three times. The manikin heater was controlled under "comfort mode" to maintain a realistic skin temperature for the conditions.

The same fan used in the human subject thermal comfort study described above (a 0.6m diameter fan) was adjusted to produce three different mean air speeds: 0.6m/s, 0.8m/s, and 1m/s. Air flows with different turbulence intensities ranging from 20% to 42% were produced by increasing the distance from fan to manikin. An omnidirectional anemometer was used to measure the air speed, sampling 10 values every second. The anemometers (Swema) have a rated frequency response of 20 Hz. TI was calculated using the standard deviation of the sampled speeds divided by the mean speed. The air flow from the fan mostly reached the face and chest.

Table 14 shows the air flow for the tested conditions. The average TI produced by the frontal fan for all the listed conditions is around 30%.

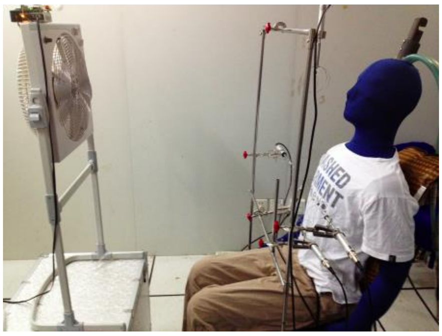

Figure 41 Convective heat transfer coefficient measurement with manikin 'Newton'

Table 14 Experimental airspeeds and turbulence intensities, measured 4.5 cm in front of the face

Average air velocity (m/s)	Turbulence intensity (%)	Distance from fan to manikin (m)
0.63	20	0.6
0.62	28	0.8
0.60	32	1
0.58	40	1.5
0.82	21	0.7
0.82	28	0.9
0.79	30	1
0.81	38	1.5
1.03	22	0.8
1.02	27	1
1.00	31	1.1
0.98	42	1.6

The Nishi and Gagge (1970) experiment had measured time-averaged local heat transfer coefficients and air speeds for walking people at 10 locations around the body (front head, chest, back, upper arm, lower arm, hand, thigh, and lower leg). The sensors (sublimating naphthalene balls) were attached to the body at a fixed offset of 4.5 cm from the skin. The local TI values were not quantified.

In order to determine the TI that would have existed during the Nishi/Gagge tests, we repeated their test protocol, measuring airspeed and TI at the same 10 body locations with our omnidirectional anemometers positioned 4.5 cm away from the skin (Figure 42). The subject walked at the speed of 1m/s in still air. The setup is shown in Figure 3.

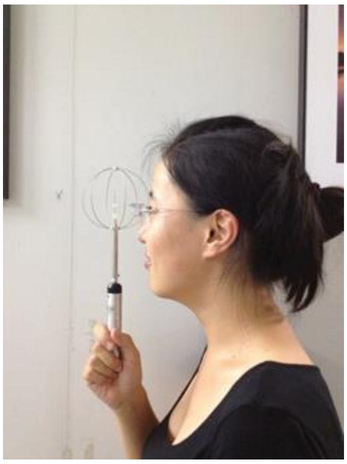

Figure 42 Turbulence intensity test for walking people

2.6.3 Results

2.6.3.1 Use of SET to represent spatially non-uniform air flow cooling

SET values were calculated for each set of environmental conditions in the human subject tests. SET was obtained from 2-Node as embodied in the ASHRAE Thermal Comfort Tool (Fountain and Huizenga 1997). Two air speeds were used as input: the airspeed in front of the face, and the average of the speed at three heights, 0.1, 0.6, and 1.1m. In each case the measurements had been taken 5cm in front of the body.

We did the regression of SET using the whole-body air speed against actual TSV (Figure 43). The SET value and TSV are linearly and closely related. It suggests that SET is a practical index for predicting human thermal sensation in warm environments, even under the non-uniform air flow conditions of this study.

The regression between SET and TSV using the air speed in front of the face also shows them to be linearly and closely related. The slope is higher and the intercept is lower for the whole-body SET (TSV=0.3106SETwhole-body – 8.1165, R2=0.93) than for the slope and intercept for SET using the air speed in front of the face (TSV=0.2846SETface – 7.1041, R2=0.94). This is because the latter uses a greater air speed to calculate the SET, overestimating the cooling effect and producing a lower slope and SET value. The comparison shows that it is fine to use either facial or whole-body-average air speed to calculate SET in order to predict thermal sensation, as long as the corresponding regression equation is used.

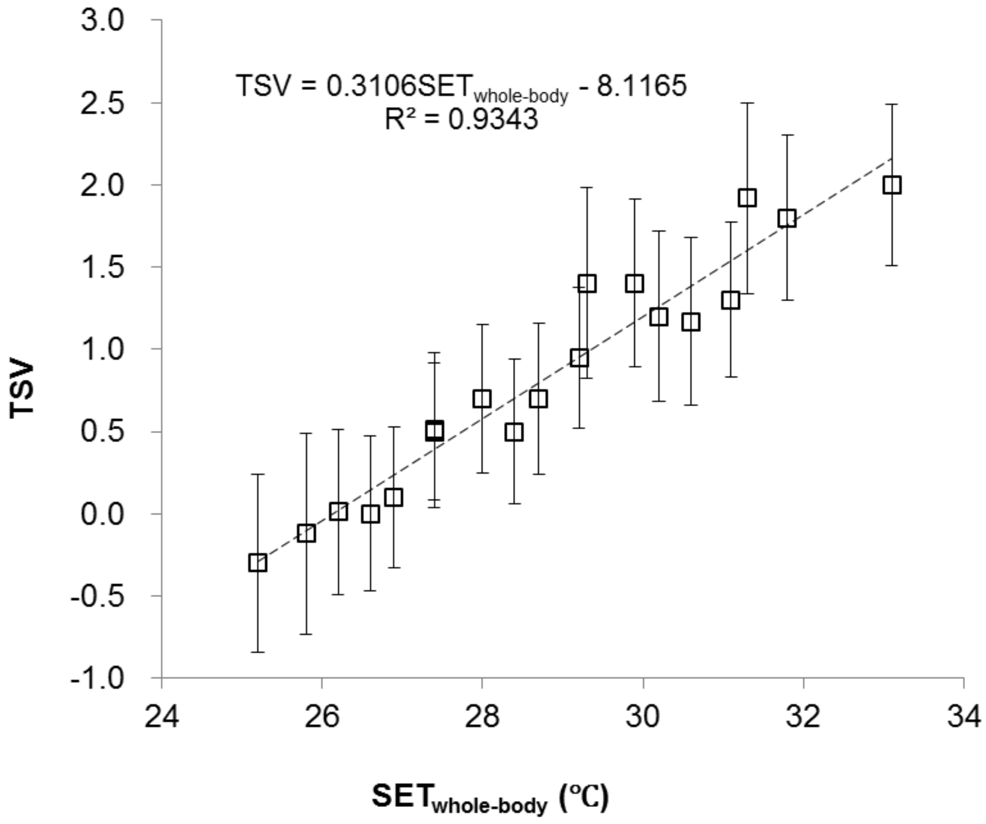

Figure 43 Relationship between SET using the whole-body air speed and TSV

Figure 44 compares the results from this study with studies from the literature in which air temperature and air speed were tested orthogonally. These studies are listed in Table 1. The regression for the device category 'desk fan' is based on the test results from this study and from study (Atthajariyakul and Lertsatittanakorn 2008). It indicates that thermal sensation differs by body part when exposed to the same air speed, which can be seen in the variation of TSV under the same SET. When the SET value was high-- ie, the air speed was low--the difference among the different modes of exposure was not so significant.

Subjects in experiments with their heads exposed to ceiling fans (Zhai 2013) and jets (Yang et al. 2010) had relatively warmer thermal sensation than subjects with chest and whole-body exposure. This may be because the top of the head exposes a smaller area to the air flow in ceiling fans and jets. The presence of hair may also be a factor. Comparing the ceiling fan and ceiling jet, the jet produced a warmer thermal sensation, again due to the smaller body area impacted. The face appears to be more sensitive to the cooling effect of air movement (Zhang 2003). For the experiment in which subjects' whole back was exposed to the air flow from a large-area wind box (Tanabe and Kimura 1994), people had strong cool sensations because the exposed body area was larger than the other exposures.

Table 15 shows the differences in TSV-versus-SET regression coefficients for all these exposure conditions. Statistical analysis shows significant pairwise differences between the regression lines. The coefficient for whole-back cooling (0.37) is larger than the coefficient for the desk fan (0.33), ceiling fan (0.28), and ceiling jet (0.26), indicating that cooling effectiveness decreases in this order. In ASHRAE Standard 55, the cooling effect of air movement is calculated with SET, without reference to the type of exposure or wind source. From the analysis above it is seen that variation does exist between different exposures to air movement. However, for the most common airspeed sources (ceiling fan and desk fan), the variation of thermal sensation for a given SET is small (see the open diamonds and triangles in Figure 44). Only when SET is as low as 22°C, a temperature too cool for elevated airspeeds, does the variation between the ceiling fan and desk fan reach 0.5 in the thermal sensation scale (Figure 44).

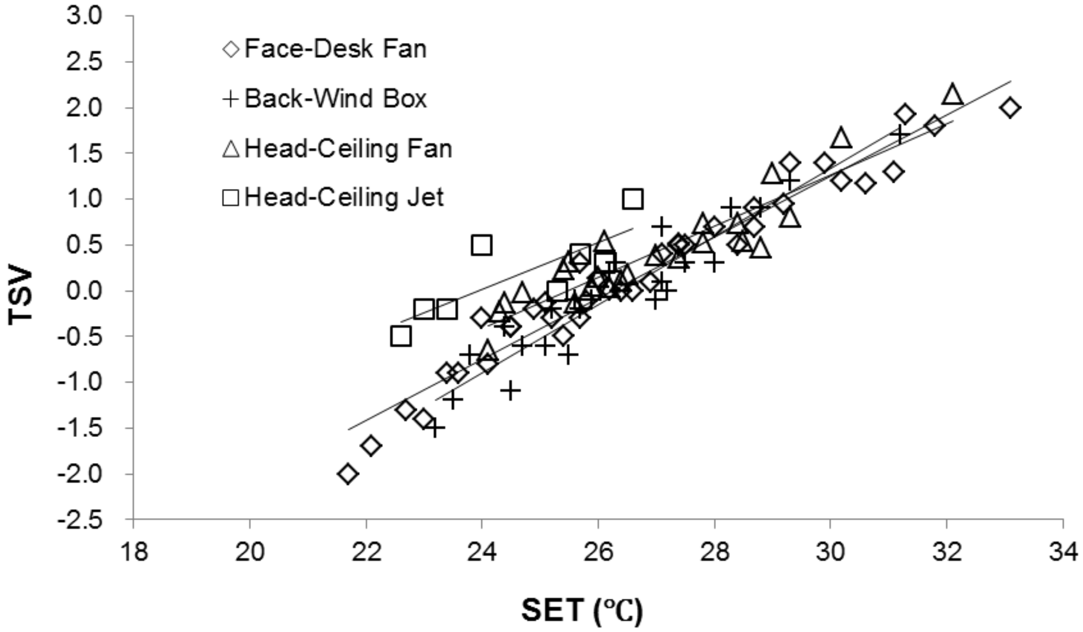

Figure 44 Relationship between TSV and SET in different experiments

Table 15 Linear regression equations of TSV and SET

Exposed body part-Air flow facility	Linear regression equation	R^2
Back - Wind Box (Tanabe and Kimura 1994)	TSV=0.37SET-9.82	0.89
Face - Desk Fan (current and Atthajariyakul Lertsatittanakorn 2008)	TSV=0.33SET-8.74	0.95
Head - Ceiling Fan (Zhai 2013)	TSV=0.28SET-7.15	0.85
Head - Ceiling Jet (Yang 2010)	TSV=0.26SET-6.12	0.66

2.6.3.2 The effect of turbulence intensity on convective heat transfer coefficient

The total dry heat transfer of the head was measured with the manikin as described in 2.3. The radiation heat transfer coefficient for the head region was obtained from previous research (3.9 W/m^2 per K) (deDear et al. 1997). Subtracting this radiative coefficient from the total dry heat transfer coefficient gives the convective heat transfer coefficient h_c for the head. Each of the dots (Figure 45) represents the mean value of three repeated measurements at the indicated airspeed and TI. For a given mean airspeed, the convective heat transfer coefficient increases with turbulence intensity, and the contribution from a given increment of additional turbulence intensity increases with airspeed.

Mayer (1987) found that the convective heat transfer coefficient depends on air speed only at lower TI; and is a function of the product of TI and air speed at higher TI. His approach requires a piecewise calculation. Others (Kondjoyan and Boisson 1997, Kondjoyan and Daudin 1995, Verboven et al. 2000, Zhou 2005) have generally related the convective heat transfer coefficient to mean air speed and TI as follows:

$$h_c = A \times v^{0.5} + B \times TI \times v$$

where A and B are constants.

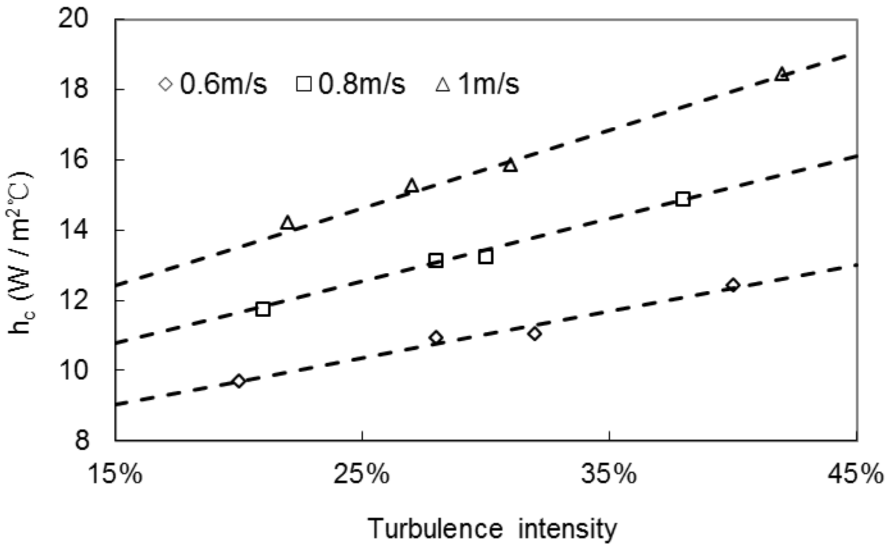

Figure 45 Bivariate linear regression of air velocity and turbulence intensity

The coefficients for the equation can be obtained from the Figure 6 data by bivariate linear regression.

$$h_c = 9.08v^{0.5} + 22.16TI \times v$$

These coefficients produce a good fit to Mayer's measured data, within 20%.

Using the linear relationship between SET and TSV given above for the desk fan (Table 15), TSV was predicted from SET as calculated with the new (airspeed+TI) heat transfer coefficient. Table 4 shows the drop in TSV for a wide range of TI levels (10, 30, 60%). The theoretical base case for each drop is TI = 0. For TI levels of 10%, 30%, and 60%, the TSV drops are about 0.1, 0.3, and 0.4 TSV scale units. One must interpolate between these drops to account for the 2-Node model's base case TI of 24%. This value, the area-weighted average of TI at the 10 measurement locations in Nishi/Gagge's experiment, was obtained in our test of a subject walking at 1 m/s (shown in Figure 3).

In our walking TI test, the TI for head was found to be 19%, for chest 16%, and for forearm 29%. These would be the local TI values implicit in 2-Node. Comparing 2-Node's head value of 19% with the TI that occurs at the head in the airstream of a frontal fan (average value around 30%), the SET prediction of 2-Node should underestimate the cooling by about 0.1 TSV scale units.

Table 16 Effect of TI level on TSV, compared to a zero TI base case

Air temperature	Air velocity	Predicted drop of TSV (TI=10%)	Predicted drop of TSV (TI=30%)	Predicted drop of TSV (TI=60%)
28	0.6	0.13	0.31	0.49
28	1	0.14	0.34	0.51
28	1.5	0.16	0.35	0.50
30	0.6	0.13	0.30	0.47
30	1	0.14	0.31	0.46
30	1.5	0.14	0.31	0.44
30	2	0.15	0.31	0.43
32	0.6	0.13	0.30	0.46
32	1	0.13	0.30	0.43
32	1.5	0.13	0.29	0.41
32	2	0.14	0.28	0.39
34	0.6	0.12	0.29	0.44
34	1	0.13	0.28	0.40
34	1.5	0.13	0.27	0.37
34	2	0.13	0.26	0.35

For comparison, measured indoor TI values range from 10-60%, most typically between 20-40% (Melikov et al. 1988). In the current fan test, TI ranged from 20-40% (Table 14).

2.6.4 Discussion

2.6.4.1 Use of the PMV model for predicting thermal sensation under air movement

The PMV model is a whole-body model used in Standard 55 for determining the still-air comfort zone. It is worth comparing its performance to that of the SET approach. PMV values were calculated for all the conditions in the human subject experiments described in Table 3. Unlike SET, the cooling effect of air flow on sweat evaporation is not taken into account in the PMV model. The solid symbols (Figure 46) represent results in which the skin wettedness is 0.06, meaning the skin has not begun to sweat. In this condition, heat transfer by evaporation at skin surface is small, and PMV is seen to predict thermal sensation fairly well, with the PMV values near the TSV values. However when skin wettedness is higher than 0.06, represented by the hollow marks in the figure, the evaporative heat transfer is larger and strongly affected by air speed. This leads to the result that PMV overestimates subjects' actual thermal sensation votes, and underestimates the cooling effect of air speed.

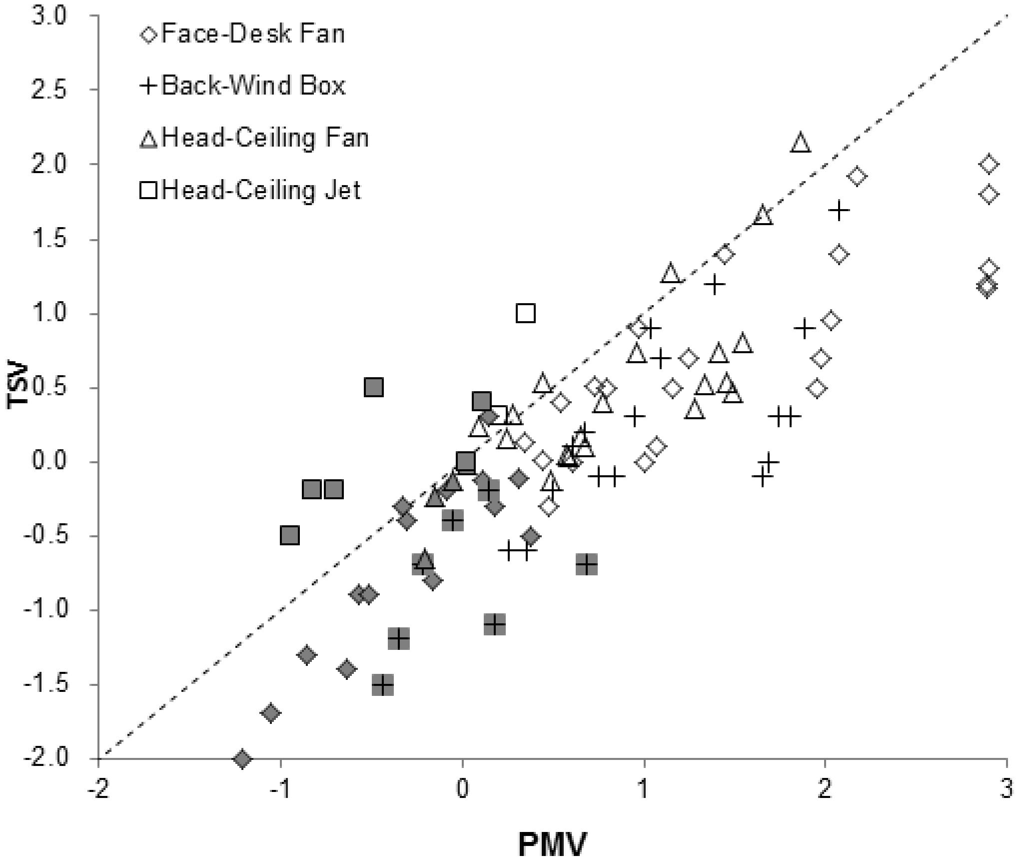

Figure 46 Relationship between TSV and PMV

2.6.5 Conclusions

Although it is based on whole-body heat balance, the SET index can be used to predict thermal sensation for air movement that is not uniformly distributed across the body. The SET value and thermal sensation vote TSV are linearly well-related for a variety of non-uniform airflow distributions.

Fan airstreams impacting different body parts require different regression coefficients for predicting TSV from SET. Representative coefficients are provided form an analysis of the existing literature on fan studies. However, the differences in cooling between substantially different types of fans are small. This can be seen in comparing the effects of ceiling fans (fan or jet) with desk fans.

The PMV model does not take into account the cooling effect of air flow on sweat evaporation, causing PMV to overestimate subjects' thermal sensation in warm conditions when the fraction of skin wettedness is above 0.06. This cannot be easily corrected by simple regression.

The 2-Node model implicitly involves a TI of 19% for the head and 24% for the whole-body average. In the current study, TI in the front of the face created by fans is around 30%. The differences between 19% and 30% in predicting in heat transfer coefficients and thermal sensation is small (within 0.1 TSV scale unit). For most design purposes, it is not necessary to consider TI differences.

Acknowledgements
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3 Thermal comfort in naturally ventilated and mixed mode buildings with ceiling fans – field studies

Laboratory studies tend to focus on fundamental issues, such as determining the temperature and humidity limits for comfort with different types of air movement. Although thermal comfort researchers have increasingly made their laboratory test experiences as realistic as possible, real buildings and occupancies contain features that cannot be tested in laboratory studies. For example, laboratory studies normally take place over relatively short periods of time, and though the subjects may be involved in normal work tasks, it is not their long-term work environment. In a real office building, when people at work are exposed to air movement all day, and for many days, do they find that such an environment is acceptable?

This chapter describes field studies of naturally ventilated buildings that included ceiling fans. ***Project 5*** describes field studies that we conducted in a project funded by the California Energy Commission PIER program (Arens et al. 2014). In these studies, we examined thermal comfort in one naturally ventilated and two mixed-mode office buildings. The primary goal of the CEC study was to determine the levels of comfort that can be maintained in office buildings equipped with operable windows, and to compare their occupants' satisfaction levels with those of conventional air-conditioned buildings. Because of this project's focus on ceiling fans, we also monitored fan use and added questions about occupants' responses to the long-term exposure to the ceiling fans. These results are summarized in this report.

A school project is described in ***Project 6***. CBE industry partners Taylor Engineering LLC (an engineering design firm) and Big Ass Fans (a fan manufacturing company) had completed an innovative project employing large ceiling fans in Oakland school district classrooms. The school had been occupied 2 years at the time we studied it. Our team surveyed the teachers' opinions about ceiling fan use in their classrooms, and issues related with automatic fan speed control, which is important because automatically controlled ceiling fans have started to appear in designs. Ceiling fans can also be used in the heating season to efficiently destratify warm air that collects near the ceiling to warm the occupied zone. We tested the effectiveness of the ceiling fans in reducing air temperature stratification.

3.1 Project 5: field studies of naturally ventilated or mixed mode office buildings with ceiling fans

3.1.1 Introduction

In many NV buildings, operable windows are the most prominent source of air movement and personal cooling control, but they are logically supplemented by ceiling and desk fans. Such air movement devices now use very small amounts of electricity, and give occupants air motion and personal control that are not dependent on outside wind conditions or on the occupants' distance from, or access to, an operable window.

3.1.2 Method

Three buildings were tested. One entirely NV building was studied intensively over time. It is the office of an architecture and energy consulting firm in Alameda, CA. The office is on the second floor of a 2-story building and contains 13 occupants (Figure 47). Occupants were repeatedly surveyed over a course of a full year (Oct. 2011 to Oct. 2012) about their perceptions, satisfaction, and thermal preferences using a "right now" survey that obtains point-in-time responses to the environment (see Appendix A for the questionnaire). The surveys were conducted 3 times/day for 2 weeks each month or for 2 weeks every two months when the weather was mild. About 2000 individual survey responses were received. Hourly window opening and fan operation was monitored through timelapse photography and fan-power logging, and the space's temperature, relative humidity, and CO_2 profiles were measured in 5 minute intervals. The timestamps of these physical measurements survey responses were matched to find the conditions under which the responses were given.

Figure 47 Exterior and the interior of the study building (ceiling fans are highlighted by a red circle)

In addition to the repeated “right-now” survey and detailed environmental monitoring in this building, two other office buildings (mixed mode) with operable windows and ceiling fans were studied using the CBE general survey.

The CBE Occupant Satisfaction Survey was administered once in each building to obtain people’s long-term experience of the building’s indoor environmental quality (IEQ) and its effect on their work performance (Huizenga et al. 2006). The main IEQ categories are thermal comfort, indoor air movement, perceived air quality, lighting, acoustics, space, and quality of furnishings and upkeep. Modules containing questions about window and fan usage were added to the general survey. The core questions of the CBE survey have now been administered in over 650 buildings with over 65,000 individual responses resulting in a large database of responses that can be used as a performance benchmark of IEQ in the entire building stock. The scores from an individual building survey can be compared against the accumulated scores in the CBE occupant satisfaction database, using either the entire database or filtered subsets thereof.

General survey building 1: The DPR Construction office building is located in Phoenix, AZ. It is certified as net zero energy by the International Living Future Institute (ILFI) through its Living Building Challenge (Figure 48). The building has 13 ceiling fans, 87 operable windows, an 87-foot long zinc-clad solar chimney and a photovoltaic array rated at 79 kW-dc. It is operated as a mixed-mode building with the mechanical cooling switching on at temperatures above 27°C (82°F). This is a very unusual interior cooling setpoint for an office building. 37 out of 45 employees answered the survey (82% response rate).

Source: DPR Website

Figure 48 DPR Net Zero Building, Phoenix. The circle points to a ceiling fan.

CBE general survey building 2: The Molecular Engineering and Sciences Building at the University of Washington has office areas that are mixed-mode-controlled, and laboratory areas which are fully air-conditioned (Figure 49). The setpoints deadbands are 68 – 72F in the MM offices and 68 – 70F in the laboratories. Occupants often move between the MM offices and the air-conditioned laboratories. The laboratories tend to be slightly cooler than the offices. Because the work activities are different in the offices and laboratories the survey results include an unusual amount of transient perception. 41 out of 185 employees answered the survey (22% response rate).

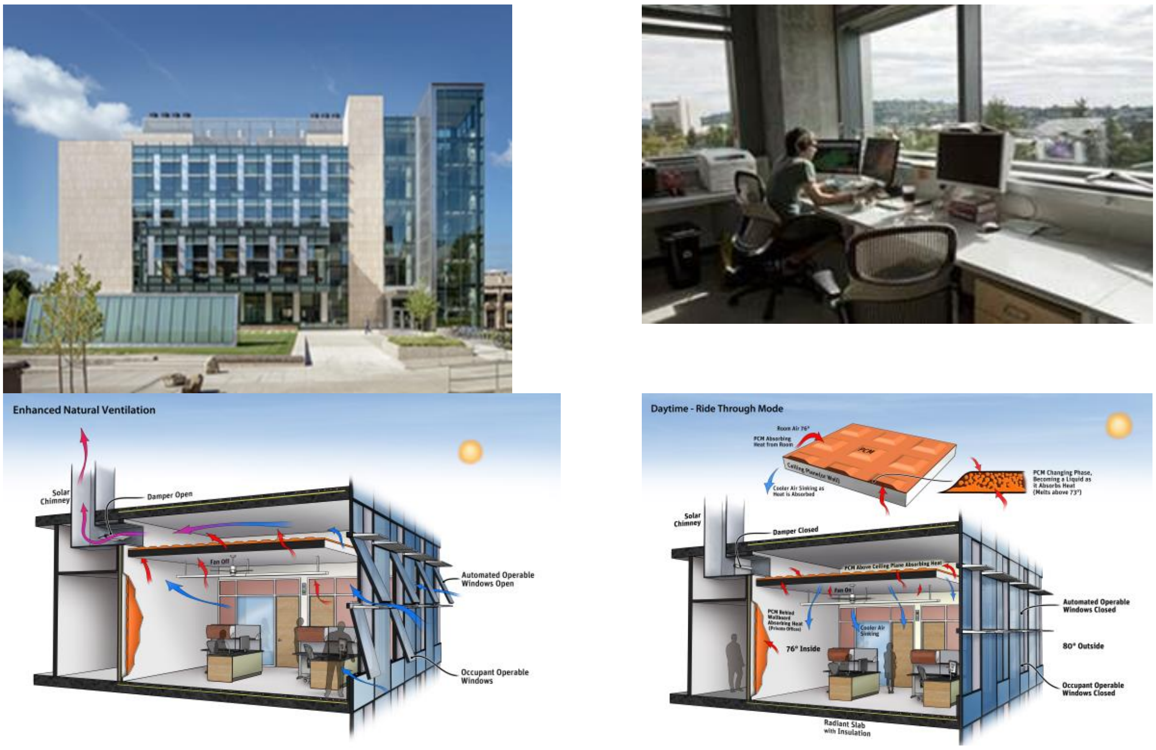

Figure 49 University of Washington Building and its natural ventilation strategies

3.1.3 Results

3.1.3.1 The year-long thermal comfort monitoring in the NV building in Alameda, CA

Thermal comfort analysis

Figure 50 shows the relationship between thermal sensation and indoor air temperature in the summer season from June through October. From cold to neutral temperatures, thermal sensation increases with temperature as expected. It is interesting that above 22°C the rate of sensation increase is lower. Perhaps convective cooling from air movement is offsetting the rise in thermal sensation at higher air temperature since windows and fans are being used in this indoor temperature range (20 – 21°C for windows, 22 – 23°C for fans) (Figure 51).

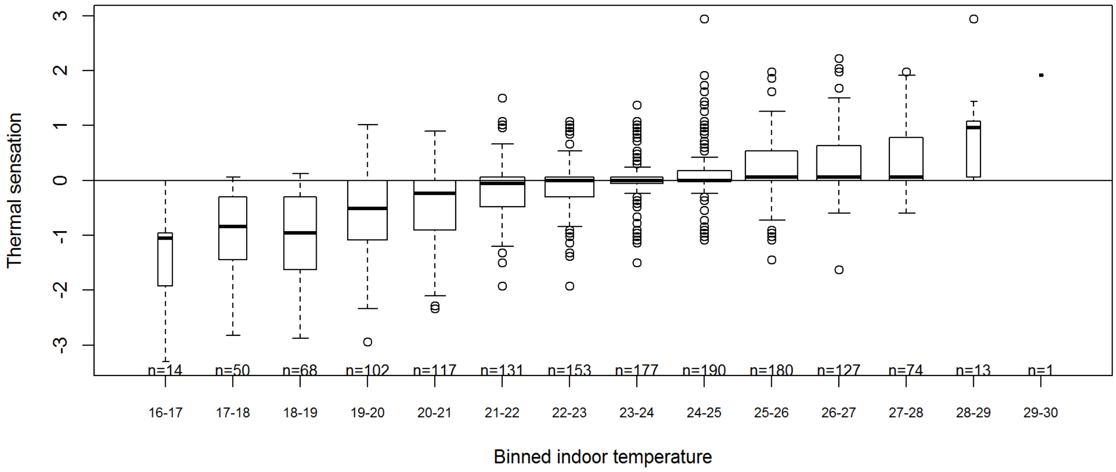

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Figure 50 Thermal sensation vs. indoor air temperature

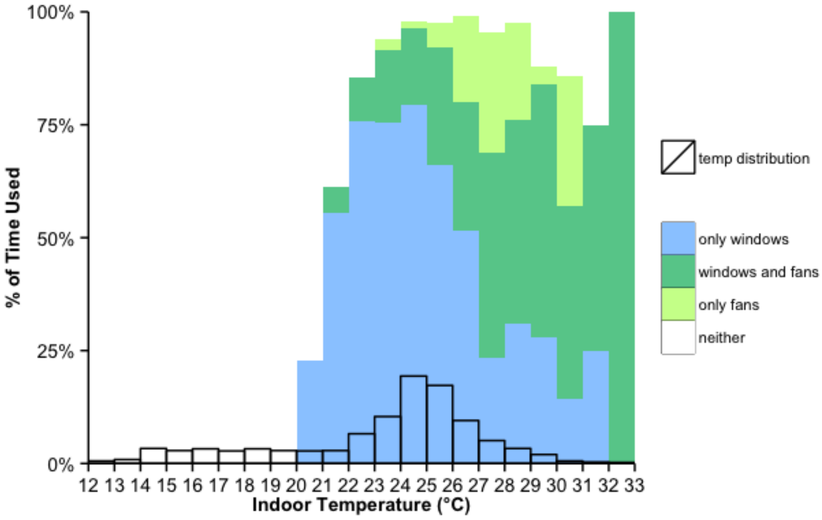
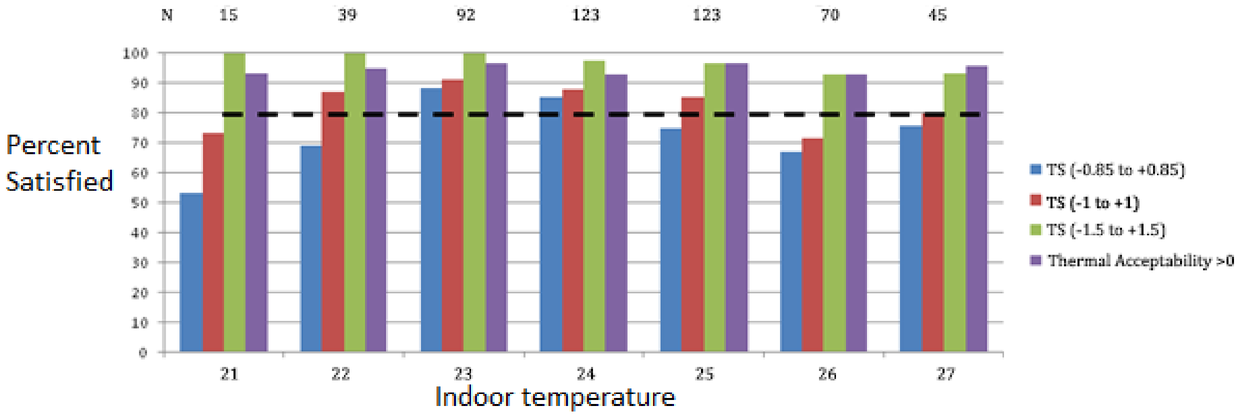


Figure 51 Windows and fans vs. indoor air temperature

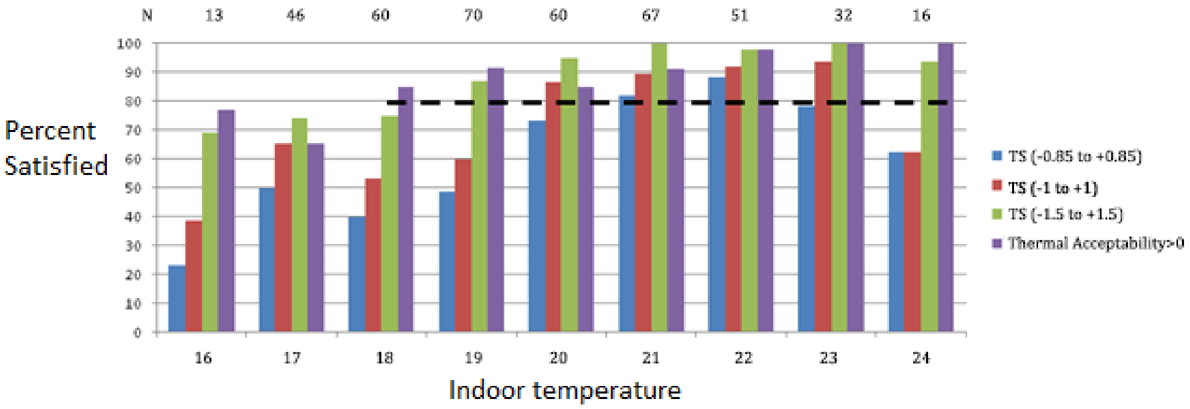
Defining the comfort zone is an ongoing issue in standards committees. There is great interpersonal variability in an occupant's neutral temperatures and in the ranges of temperature above and below these neutrals that occupants find acceptable. The ASHRAE 55 comfort zone for air conditioned buildings is based on a calculated prediction of a population's mean thermal sensation vote (PMV). The allowable PMV range in Standard 55 is ±0.5 representing 80% satisfaction. The Adaptive Model comfort zone for NV buildings also represents 80% satisfied, but has a width that would be predicted by a PMV of ±0.85 (de Dear and Brager 1998b). Both of these comfort zones were developed using an empirical percent dissatisfied curve developed from laboratory studies that is open to question (Arens et al 2010).

Field surveys have increasingly asked a question about thermal acceptability to supplement the thermal sensation question. The thermal acceptability metric can be used directly to determine the range of thermal sensation that is acceptable for occupants. Figure 52 shows the percentage of people satisfied in a particular indoor temperature bin, as defined by three thermal sensation ranges: ±0.85, ±1, and ±1.5. The threshold is defined as the temperature at which 80% occupants are satisfied. The ±1.5 range matches most closely with the thermal acceptability threshold.

This study's results can be compared with the thresholds for the ASHRAE 884 database (Zhang et al. 2011). The summer threshold (temperature at which 80% occupants vote "acceptable") for the Alameda building is 21-27°C (Figure 52) while for the ASHRAE database (for NV buildings) is 22-30°C. The winter threshold range for the Alameda building is 18-24°C while it is 19-27°C for the ASHRAE database. Perhaps the lower hot season threshold ranges from the current study are due to the mild climate in Alameda.



a. Summer (June-October)



b. Winter (December-February)

Only bins with at least 5 votes are shown.

Figure 52 Thermal sensation and thermal acceptability

Figure 53 explicitly compares the percent of people dissatisfied at each thermal sensation range (blue) with the standard PMV-PPD curve (dotted). Compared to the standard PMV-PPD curve, the one based on this field study data is broader and indicates that the 20% dissatisfaction threshold does not occur until thermal sensations of ±1.5 or even ±2.

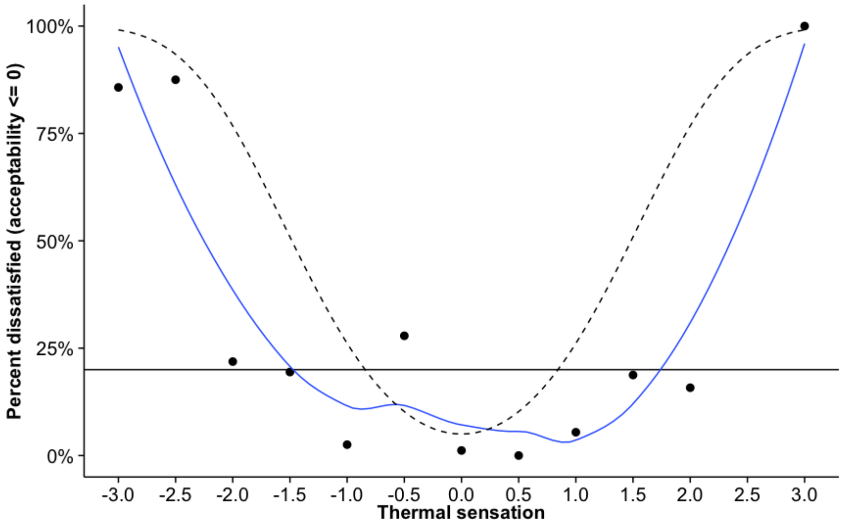


Figure 53 Thermal sensation and percent of people dissatisfied (PPD)

The noise satisfaction analysis

The satisfaction is separated for conditions when all the windows were closed or open at the time of the survey. "Windows open" means that at least one window was open in the building. As usual the occupants had very few complaints. You can see that 99% of the right-now survey results are satisfied, and the satisfaction levels are similar with window closed or open (Figure 54, positive mean satisfied and negative mean dissatisfied, see questions in item 4 in Appendix A).

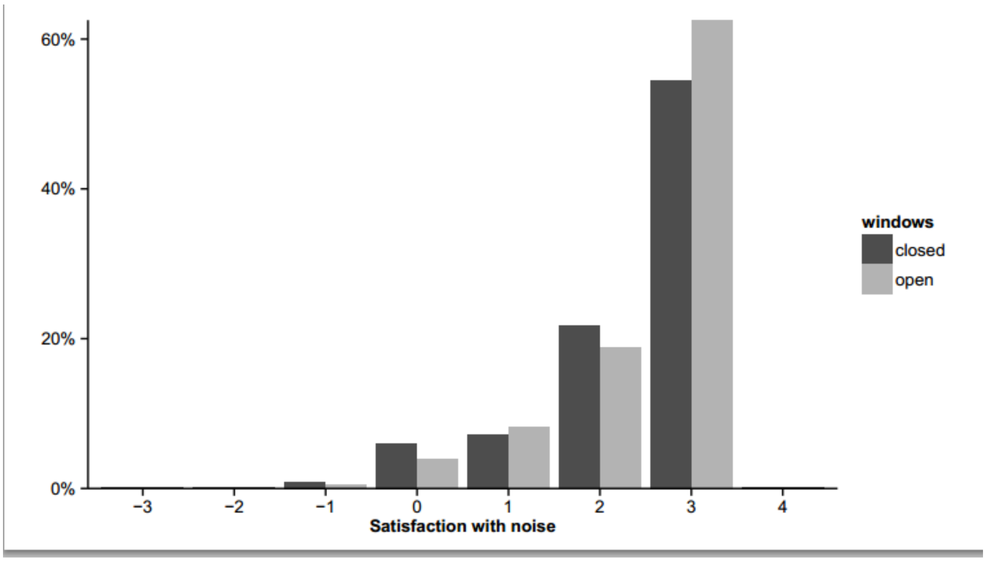


Figure 54 Noise satisfaction for window closed or open during the time of the survey

For the 1% of the dissatisfied votes, the sources of dissatisfaction show no difference between windows being open and closed for the 5 categories listed in question 4 in Appendix A (Figure 55). The ceiling fan is not the major source of noise dissatisfaction.

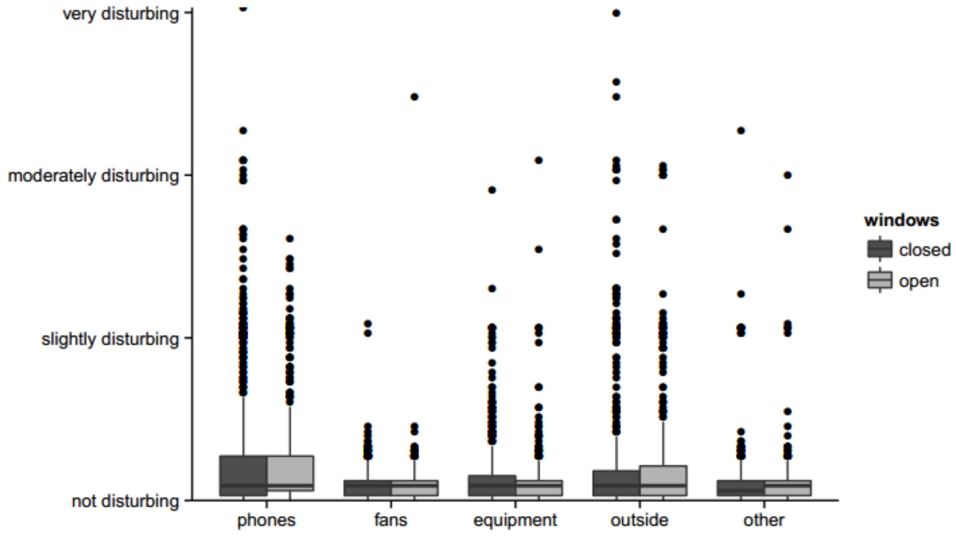


Figure 55 Distribution of noise dissatisfaction

3.1.3.2 DPR survey results

Overall satisfaction

The overall satisfaction with the building is high: 97% of occupants were satisfied. This positions the building in the top 5% of the entire CBE Survey database. In addition, 92% of the occupants were satisfied with the fact that the building has natural ventilation features.

Temperature satisfaction

The mean score for temperature satisfaction is 0.97 on a scale that goes from -3 to 3. This is much higher than the entire survey database (mean score equals -0.13), LEED buildings (0.42), and mixed-mode buildings (0.62). The building's temperature satisfaction is ranked in the 91st percentile of the entire database (Figure 59 below). This is a dramatic finding since the air conditioning does not switch on until the indoor temperature is above 28°C (82°F). Clearly the operable windows and ceiling fans allow people to be comfortable.

Counting neutral votes as satisfied, 81% of the occupants were satisfied with the temperature and 72% were satisfied with the ability to control temperature (Figure 56, Figure 57). 75% occupants felt that thermal comfort in their workspace enhanced their ability to get their job done.

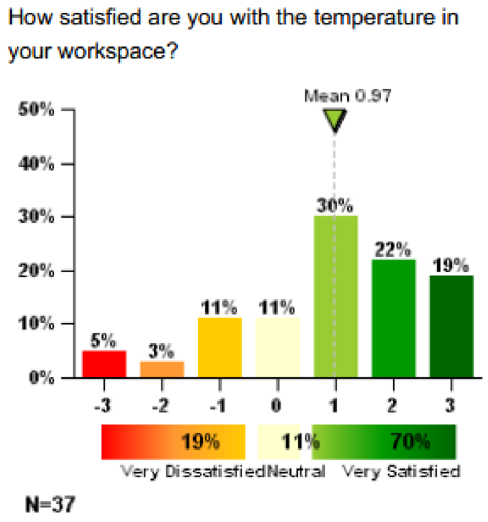


Figure 56 Satisfaction with temperature

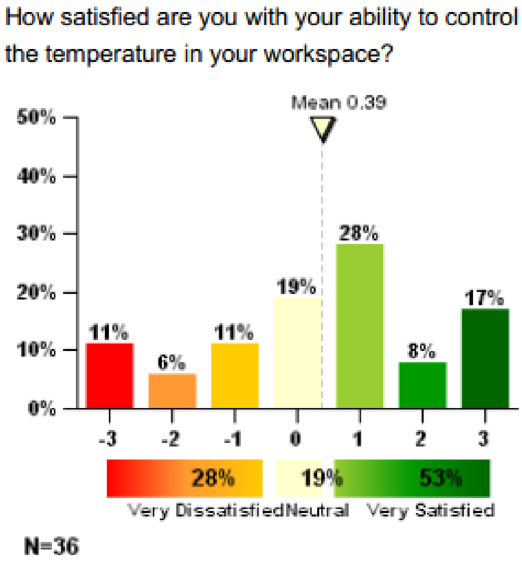


Figure 57 Satisfaction temperature control

Air Movement and ceiling fans

75% of occupants were satisfied with the amount of air movement in the workplace and 67% felt that it enhanced their ability to get their job done. 83% of occupants were satisfied with the ceiling fans in their workspaces. 70% of occupants indicated that fans provide relief from being too warm and that the air movement made them comfortable. When asked what might be the main concerns regarding using ceiling fans, the main concern was that air movement might be disruptive (blowing papers).

Noise analysis

Indoor noise by people is by far has been found to be the most important source of dissatisfaction for both air-conditioned and naturally ventilated buildings (Goins et al. 2012). The current finding from DPR survey showed similar results. The people who were dissatisfied with the noise, again the major source is from people talking indoors (Figure 58).

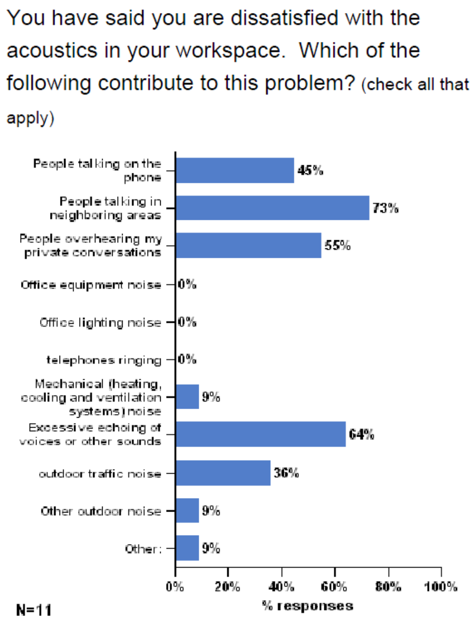


Figure 58 sources of noise dissatisfaction in DPR

3.1.3.3 University of Washington survey results

Overall satisfaction

The overall satisfaction with the building is high: 92% of occupants were satisfied. This positions the building in the top 8% of the entire database.

Temperature satisfaction

The mean score for temperature satisfaction is 0.38 on a scale that goes from -3 to 3, higher than the entire database (-0.13), but lower than the LEED buildings (0.42), and mixed-mode buildings (0.62). It is ranked in the 70st percentile of the entire database (Figure 59).

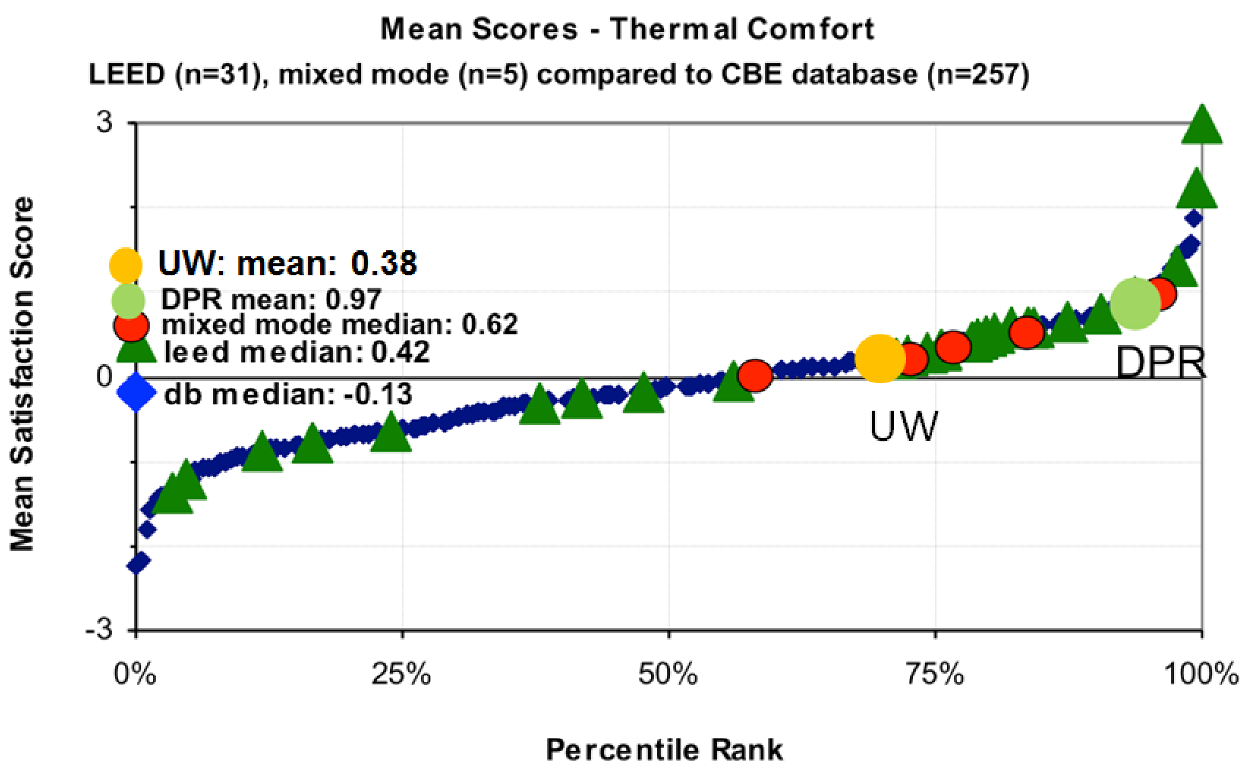
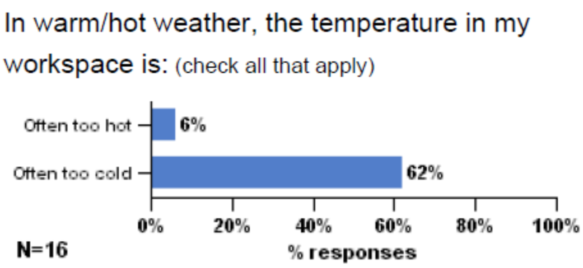


Figure 59 Temperature satisfaction percentile ranking, UW

Overcooling seems to be the reason for the temperature dissatisfaction in both summer and winter (Figure 60). 62% and 88% people who are dissatisfied with the temperature said that the workspace is often too cold in warm/hot and cool/cold weather respectively. For comparison, only 6% and 0% said that the workspace is often too warm. Although the building has ceiling fans installed, the air temperature setpoint too low for them to be used in summer. The setpoints are 68 – 72F year-round for offices and 68 – 70F for the laboratories.

70% occupants felt that thermal comfort in their workspace enhanced their ability to get their job done.



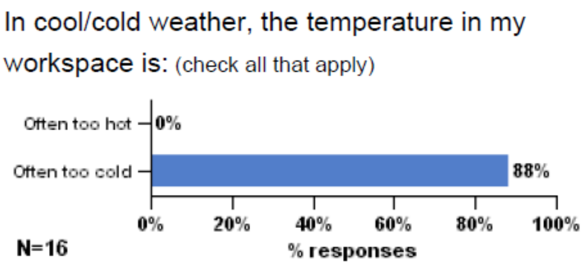


Figure 60: Reasons of temperature dissatisfaction

Air Movement and ceiling fans

93% of occupants were satisfied with the amount of air movement in the workplace and 93% felt that it enhanced their ability to get their job done.

100% of occupants are satisfied with ceiling fans in their workspaces. When asked reasons that someone might be dissatisfied with ceiling fans, the highest reason was that they do not have access to a ceiling fan. Only 18% said that the air movement might be too disruptive (paper blow etc.). As for added question whether ceiling fans are visually distracting, only 5% (the second lowest percentage among all the listed possible reasons) agreed.

From the survey responses from DPR and UW, it appears that UW is operated at lower ambient air temperatures and ceiling fans are not operated as much as in DPR.

Noise analysis

For people who were dissatisfied with the noise in UW, again the major source is from people talking indoors (Figure 61).

You have said you are dissatisfied with the acoustics in your workspace. Which of the following contribute to this problem? (check all that apply)

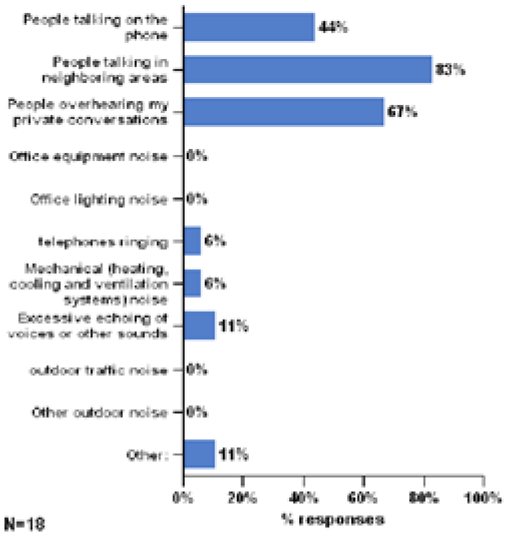


Figure 61 sources of noise dissatisfaction in UW

3.1.4 Discussion

In the MM building of the University of Washington, although ceiling fans are installed, the setpoint for cooling is low in the offices, 72F year-round. The overcooled operation does not allow the fans to save energy. When using fans or other local thermal comfort systems, the ambient temperature deadband needs to be extended in order to save energy in the HVAC system. Overconditioning with the HVAC system is a major issue in today's buildings, obviating the energy benefits of personally-controlled comfort systems such as fans.

This study also points to another issue about using fans in designs: engineers may not design or operate the HVAC system properly to match the capabilities of fans.

3.1.5 Conclusion

The occupants in the Alameda building are comfortable over a very broad range of temperatures, from 16-30°C. We measured many indoor conditions that were outside of the ASHRAE 55 80% satisfaction zone, but in which occupants' satisfactions are higher than 80%. In warm environments, people made active use of windows and fans, which by themselves maintained a high level of thermal satisfaction among the occupants. The occupants are engineers and architects involved in energy-efficient design. It might be argued that these occupants might be more tolerant than typical occupants. However, it should be noted that these findings match those of an earlier study in the Berkeley Civic Center, a large NV building of municipal government office workers (Brager et al 2004).

The thermal comfort ranking of the DPR mixed-mode buildings is very high: 92nd percentile based on the CBE occupant satisfaction database. The thermal comfort rankings in the UW building is lower than the DPR building but still above the database average: 70th percentile. Part of the reason for the lower thermal comfort satisfaction in UW is due to overcooling. With ceiling fans, the setpoints for summer could be set higher, which would enable more energy savings, provide better comfort. In DPR and UW buildings, 75% and 70% occupants felt that thermal comfort in their workspace enhanced their ability to get their job done respectively.

Indoor noise by people is by far the most important source of dissatisfaction for both naturally ventilated and mixed mode buildings. This is consistent with previous finding by CBE that in all types of buildings (air- conditioned, NV, MM buildings), indoor noise is the major dissatisfaction source.

3.2 Project 6: Oakland school classrooms with ceiling fans

3.2.1 Introduction

The Oakland Downtown Educational Complex is a large naturally ventilated school that uses an evaporative wind tower and ceiling fans for summer cooling. Ceiling fans manufactured by the manufacturer Big Ass Fans (BAF) are installed in seventeen classrooms, its multi-purpose room, and its library (Figure 62). The fan diameter is 12 feet. Each classroom has one fan installed. 15 teachers use the 17 classrooms.

To automatically control ceiling fan speed based on ambient air temperatures has just started to appear in designs. Taylor engineering developed such control algorithms, and these algorithms are incorporated into the fan operations in the Oakland school.

Using ceiling fans at lower speed can reduce room air stratification in winter by moving the warm air near the ceiling downwards to occupied zone, but the fan speeds has to be slow enough to not causing convective cooling of the occupants. How much the air movement is needed to reduce the air temperature stratification in the heating season has not been evaluated in literature.

This team worked together with the design firm (Taylor Engineering) and the fan manufacturer (BAF), both are the CBE industry partners, did the study in the school. The objective is to compare the field study results with the design intents, and to provide information for future designs.

3.2.2 Method

Surveys: We surveyed all 15 teachers to get their opinions regarding the ceiling fans in their classrooms. The survey questions include two parts. Questions 1 though 4 are for comfort, and questions 5 – 9 are dedicated to fan control issues.

Ceiling Fan Thermal Comfort Questions

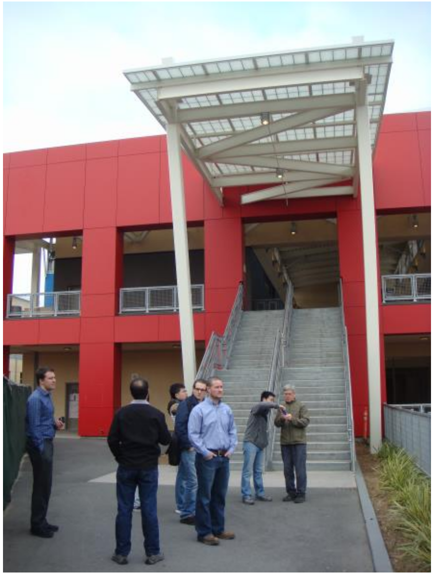
1. Does the fan provide comfort in warm weather?
2. How often do you use the fan when you feel warm?
3. Do you feel that the fan is helpful in heating to mix warm air from ceiling to the floor level?
4. Do you like having the ceiling fan in your classroom?

Ceiling Fan Control Questions

5. Do you allow the fan to operate in automatic mode heating?

6. Do you allow the fan to operate in automatic mode cooling?
7. Do you use the fan in manual mode? Why?
8. Do you ever disable the fan off? Why?
9. What would you like to improve about the fans or the control of the fans?

Temperature measurement for heating mode: The team visited two classrooms and measured the velocity and temperature distributions in March 26 and 27 2013 during the school's Spring break. Hot wire anemometers (with an accuracy of 0.05% of full scale reading) and temperature sensors were mounted on a pole to take airspeed and temperature measurements at six heights, 0.1m, 0.6m, 1.1m, 1.4m, 1.7m, and 2.0m. Figure 63 shows measurement setup in classroom 1, and Figure 64 shows test positions in classroom 2.


Entrance of the school


Cooling tower


BSF ceiling fans in multi-purpose room

Figure 62 Oakland school district

Figure 63 Airspeed measurement and flow visualization in classroom 1in an open space

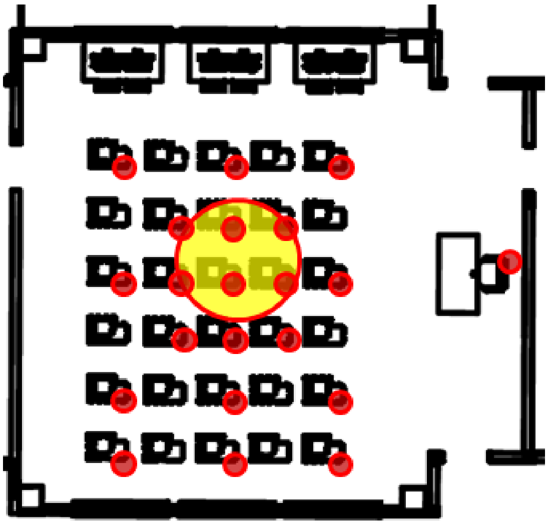


Figure 64 Measurement locations in classroom 2 with tables

3.2.3 Results

3.2.3.1 Survey results

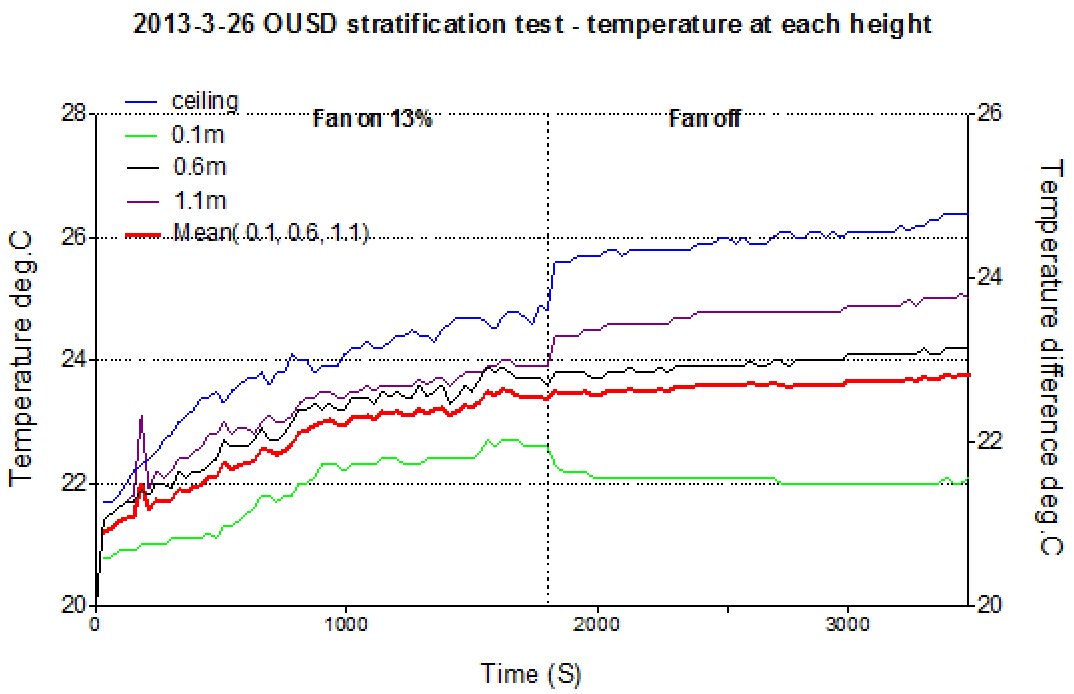
Nine teachers were satisfied with the ceiling fans. They felt that ceiling fans provides cooling and heating properly. Among the other 6 teachers, one said that the fan sometimes provided comfort, but sometimes not. For the 5 teachers who were not satisfied, one pointed out that the fan makes her dizzy, and she felt that students were initially distracted watching the fan turn. The reason for another teacher's dissatisfaction seems to be related with the fan control, which had not been operating properly, or was incorrectly operated. Another teacher said that she preferred to open the windows and door when warm, to get fresh air, rather than use the ceiling fan.

Most of the dissatisfactions was related with automatic control of the fans. One teacher felt that the control was too hard to handle. The most dissatisfied teacher said that the fan turned on when not needed, and did not turn on when needed. She also pointed out that after two years in the new building, the teachers had not received any training on how to use the fans or to override the controls. Even for those teachers who like the ceiling fans in their classrooms, several pointed that they do not know how to manually control the fans and that caused them inconvenience. Paper blowing was noted by only one teacher. He pointed out that when it happened, he did not know how to turn down the fan speed.

3.2.3.2 Temperature destratification test results

At moderate supply air temperatures (80 – 85F), turning on the fan at a low level (13% of maximum) reduced the room temperature stratification between 1.1m and 0.1m from 2F to 0.4 – 0.6F (Figure 65).

However, when the supply air temperature was higher (97-101F),the low level could only push the warmer air down to the 0.6m height, not to the 0.1m height. We see in Figure 62 that the air temperature at 0.6m increased when the fan came on but the air temperature at 0.1m height was not affected. Therefore, the low speed was not able to reduce the air temperature stratification between 1.1m and 0.1m. In fact, the air temperature stratification at this height range was increased (see “13% fan-on” period in Figure 66) from 6F to 7F. When the fan speed was increased to 20% of the entire range, then the stratification was reduced to 4F (Figure 66).



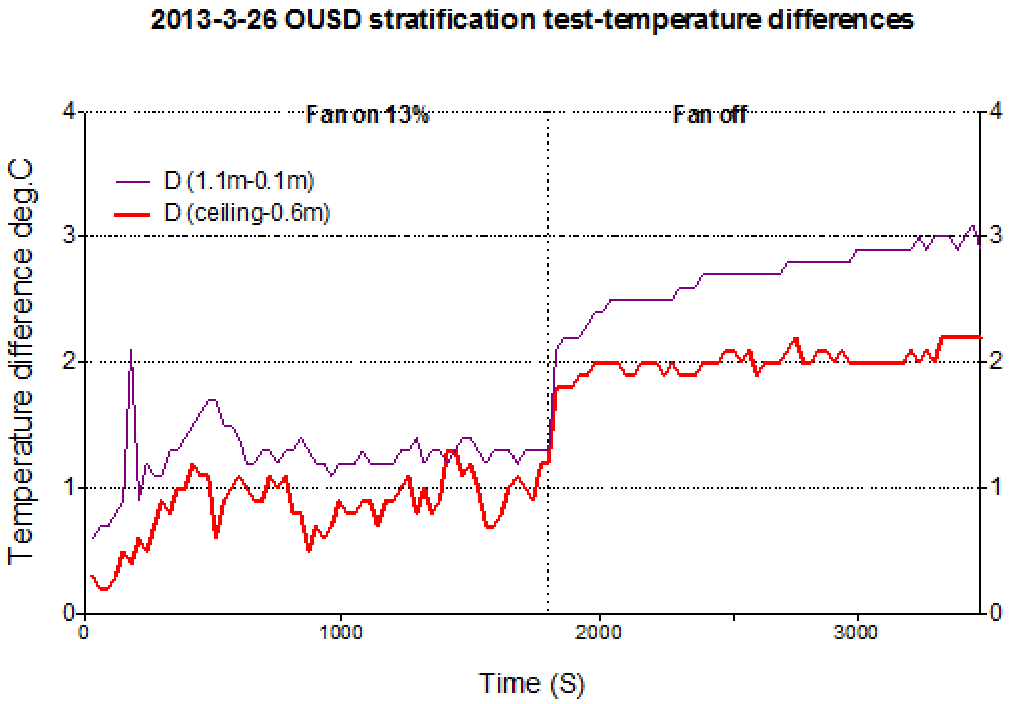
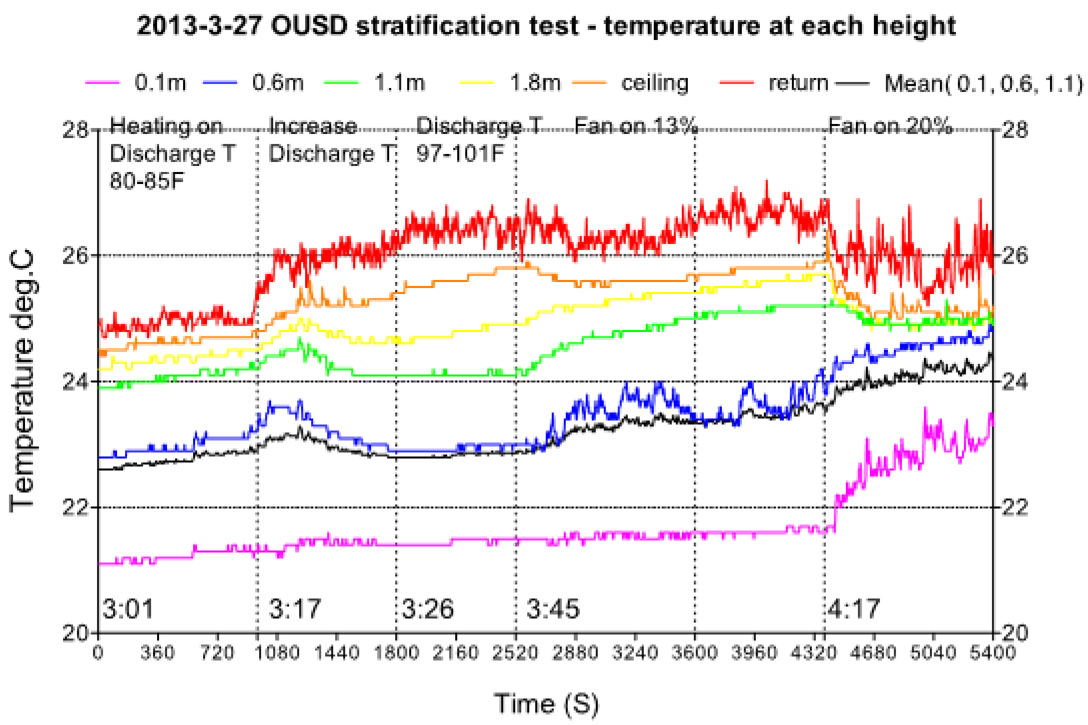


Figure 65 Air temperature stratification reduction when fan was on with low speed



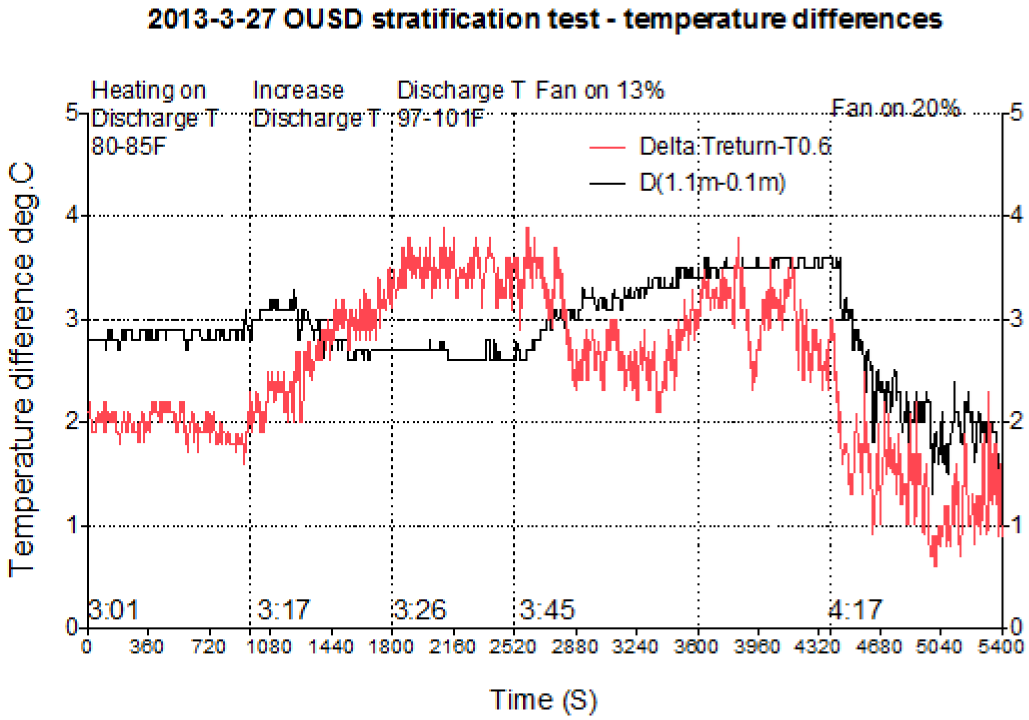


Figure 66 Air temperature stratifications at different fan levels

3.2.4 Discussion and conclusion

Most dissatisfaction with ceiling fans came from teachers feeling that they are not able to control their fans. Due to various reasons, the designers' recommended training never happened. This study shows that when using ceiling fans with automatic controls, training is very critical.

Ceiling fans reduce stratification in the heating season, but when the air temperature close to the ceiling is high (or the supply air temperature is high), the fan speed need to be higher to be effective (Figure 66). Unfortunately, as the velocity increases, the convective heat loss from the occupants also increases, which is not desirable in the heating season. There is a fine relationship between the fan speed and air temperature stratification. We only tested a limited number of supply air temperatures and fan speeds in this study. The strategy of increasing air speed to reduce air temperature stratification in the heating season needs further examination.

4 Testing air speeds for ceiling-integrated fans

4.1 Project 7: Fan velocity performance evaluation

4.1.1 Introduction

Currently, ceiling fans are evaluated by CFM/Watt (EPA Energy Star 2002, ANSI/AMCA 230-99 2000), which is valuable for rating the relative energy efficiency of fans but provides nearly no information on the effectiveness of a fan at providing comfort within a space. A method of testing that evaluates the comfort performance of ceiling fans does not exist. The information provided by ceiling fan manufacturers is generally not helpful to designers in spacing and sizing ceiling fans in rooms. ASHRAE Standard 90.1 (Energy Standards for Buildings Except Low-Rise Residential Buildings) has an agenda to develop such an evaluation method.

A table or desk surface changes the air flow from ceiling fans. It blocks the air flowing to floor level and increases the horizontal air flow near the table level. Since offices normally have tables, it is necessary to examine the influence of tables on air flow distribution in the occupied zone.

4.1.2 Objectives

1. To find the similarities in air velocity distribution generated by ceiling fans
2. To test the effect of table position on the airflow
3. To developing a standard testing method for ceiling fans based on their capabilities on providing comfort,.

4.1.3 Method

4.1.3.1 Testing facilities

The climatic controlled chamber at CBE was set up to serve as a standard space for measuring airflow from ceiling fans. Four ceiling fans are installed in the ceiling of the chamber for testing (Figure 67). The two small Toshiba fans provide vertical and oscillating flows.

Figure 67 Four ceiling fans installed in the ceiling for testing

Three heights (0.1m, 0.6m, and 1.1m) are the standard heights for measuring the environment of seated people, and four heights (0.1m, 0.6m, 1.1m, 1.7m) are standard measurement heights for standing people (ASHRAE Standard 55). In our test, an array of hot wire anemometers (with an accuracy of 0.05% of full scale reading) were mounted on a pole to take air velocity measurements at six heights (first at 0.1m, .06m, 1.1m, 1.4m, 1.7m and 2.0m to represent standing posture, then at 0.1, 0.3, 0.5, 0.7, 0.9 and 1.1m for sitting). Moving the pole allows the airflow distribution generated by the fans to be characterized. The heights 0.5m and 0.7m are near the table height. Since the velocity changes significantly near the table when the air flow from the ceiling fan intersects the table surface, we chose to measure these two heights, which represent velocity at 0.6m by averaging the two, instead of measuring at one height at 0.6m.

4.1.3.2 Testing without furniture

Three types of commercially available ceiling fans were acquired for the comparative testing. Efforts were made to find the best fans available on the market for the tests (Table 17). The fans selected all employ highly efficient DC motors, magnetically levitated bearings, and have great air moving ability. Two of the three fans won the EnergyStar most efficient fans award in 2013.

Table 17 Specs of three fans

No.	Brand	Model	Diameter(mm)	Fan speed levels	Power consumption
1	Aeratron	E503	1260	6	4 to 18 W
2	BIG ASS FANS	Haiku	1524	6	2 to 16W
3	Toshiba	SIENT	300	7	2 to 14W

The diameters of fans were different from one and another because we want to be able to generalize results. The air flow measurement points started at the center of the blade outwards in 10 cm intervals. Air velocity was measured at each of the points with the fan on low and high speed (Figure 68). Fans when tested were at least 1.5- 2.1m (5 to 7 feet) from any wall.

a. Three fans b. Measurement 0.1-2m c. Measurement 0.1-1.1m

Figure 68 Air speed testing without furniture

4.1.3.3 Testing with furniture

Tests were made only for Fan1 (Table 17) with air speeds at speed level 5 at 0.1, 0.3, 0.5, 0.7, 0.9 and 1.1m heights. The table measures 0.6m (width) x 1.2m (length) x 0.75m (height). The middle of the table and center of the fan are used to represent the fan-furniture relationships. The distance is normalized using the radius of the fan. Four relative positions were measured: 0R (center of the fan), 0.5R, 1R and 2R. Figure 69 is the schematic description.

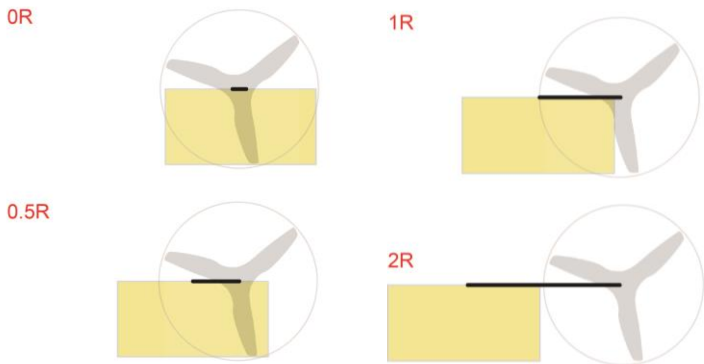


Figure 69 Testing with tables at different relative positions

4.1.4 Results

4.1.4.1 Air speeds from 0.1m to 2m

First, we examined whether the air flow is symmetrical on both sides of a fan using Fan 1 (Figure 70). It is clear that the distribution is more or less symmetrical from the center of the fan. Thus we can measure air flow on only one side of the fan. Since all the three fans are different in dimension, in order to compare the air speed distributions of different fans, we use the distance from the center of the fans divided by the radius of the fans (D/R in Figure 69).

Between 1.1m to 2m heights, air speed profiles are similar. They are lower at the center, and peak at the 0.5R. At the center, the velocity is higher at 2m height and lower at 1.1m height. At 0.5R, the velocities for all the heights measured between 1.1m and 2m are similar, reaching 3 m/s. At 1R, the velocity decreased, less than 0.5m/s (2m height) to 2m/s (1.1m height).

At 0.6m height, the peak appears at 0.65R, however the air speeds are more evenly distributed. At 0.1m, there was not much differences between different measured points, all being around 1.0m/s.

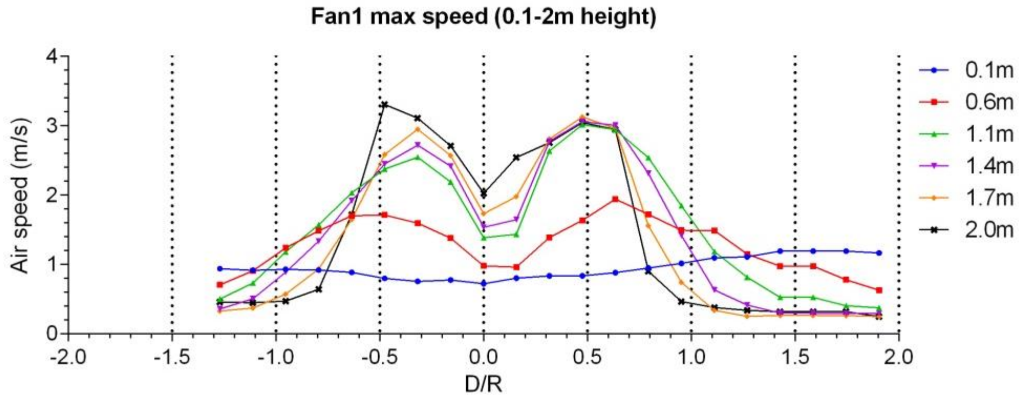


Figure 70 Air flow symmetrical examine with Fan 1 (maximum air speed)

Tests were made for Fan1, Fan2 and Fan3 at max fan speed levels. Similar patterns were found for the three fans (Figure 71). At 1.1m to 2m heights, the peak values of air speeds tend to happen at 0.25R to 0.5R for all the three fans. From these tests, it is clear that the fans provide vertical air movement only in the area under the fan blades. Beyond the blade radius the air speeds dropped almost completely, even at 0.6 m, the greatest distance from the fan where the flow is unaffected by the floor plane.

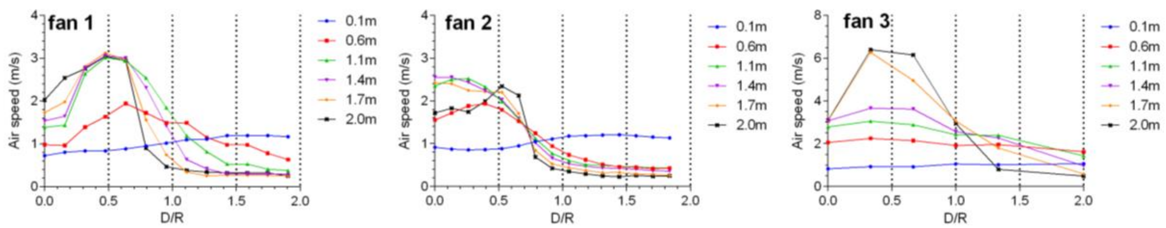


Figure 71 Air speeds distribution 0.1-2m (standing)

For the occupant sitting position, tests were made for Fan1 and Fan2 at medium and max air speeds (Figure 72). For Fan1, the peak happened at 0.5R at 0.9m and 1.1m for both high and medium speed levels. There is a trend of the positions of peak shifting away from the center of the fan as height decreases. For Fan 2, the peak is between 0.25R – 0.5R.

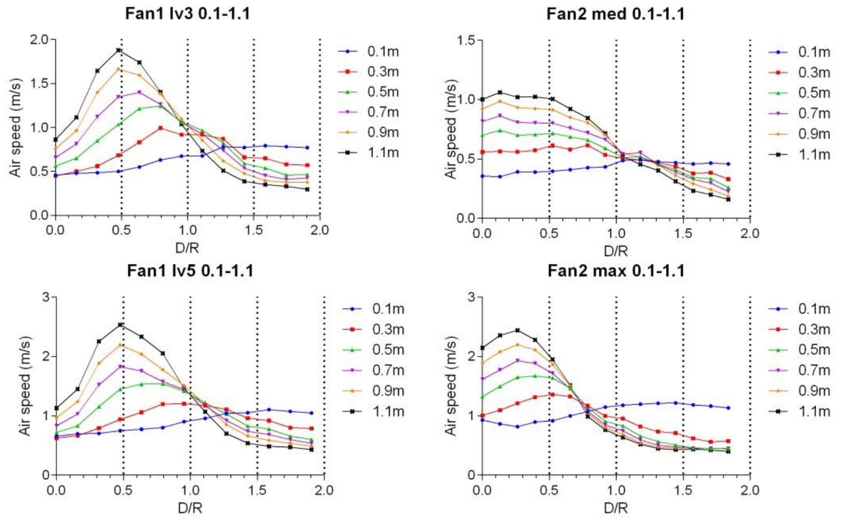


Figure 72 Air speeds distribution 0.1-1.1m (sitting)

4.1.4.2 With furniture

Figure 73 shows air speeds distributions for the 4 fan-table positions (Figure 69, 0R, 0.5R, 1R, 2R). At 0.5R, air speeds at 0.7m and 0.9m heights increased substantially because the airflow from the fan hits the surface of the table and spreads away. Obviously, the testing points lower than the surface of the table see decrements in air speeds, especially at 0.3m and 0.5m.

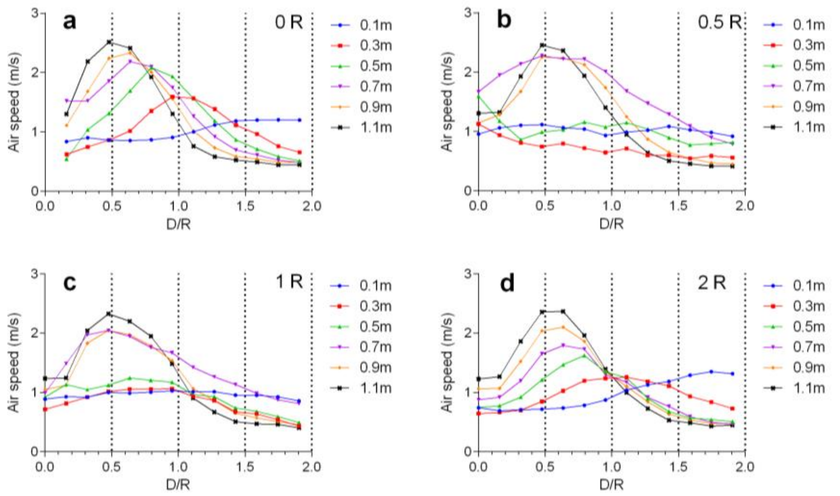


Figure 73 Fan1 at different furniture positions

However, the air speeds at 1.1m height, head level, show no difference for different table configurations (Figure 74), indicating that the table does not substantially affect the air speed at positions higher than the table. Mean air speeds of the four heights below 1.1m (0.1m, 0.5m, 0.7m, 1.1m, corresponding to the three heights specified in Standard 55--0.1m, 0.6m, 1.1m)) also show no significant differences. Thus in an open space, the effect of a table on airflow redistribution can be ignored since neither the air speed at 1.1m height or the mean air speed of four heights will be affected. However it is worth noting that in real buildings, partitions may play a role affecting air distribution, interacting with table surfaces. They will make airflow more complicated, which will require more tests in the future.

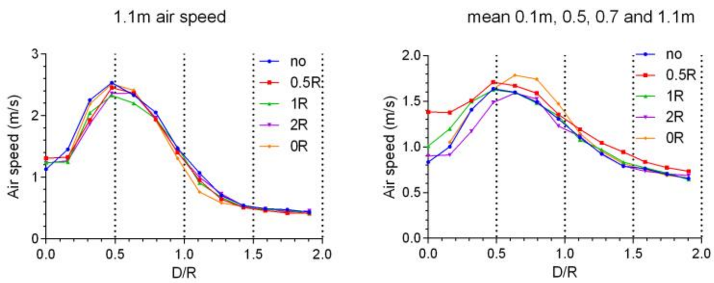


Figure 74 Air speed for different table-fan configurations

4.1.5 Conclusions

1) Similar patterns were found in air speed distributions for fans with different design and dimensions, with the peak air speeds happening around half of the fan radius from the center.

2) In open spaces, the existence of a table will not affect the air speeds at positions above the surface of the table (1.1 and 0.9m), but will increase air speed at the same height of the table (0.7m), and decrease the speed below it (0.5m and less). The table also does not affect the average air speed between 0.1m and 1.1m. Therefore, there is a good possibility that for ceiling fan velocity evaluation, it may not be necessary to include the tables and desks. This encouraging result may not apply when taller partitions are present in the space. This type of furniture configuration needs further study.

References

ENERGY STAR® Testing Facility Guidance Manual: Building a Testing Facility and Performing the Solid State Test Method for ENERGY STAR Qualified Ceiling Fans. 2002, prepared by The US Environmental Protection Agency

Laboratory methods of testing air circulator fans for rating. 2000. ANSI/AMCA 230-99, An American National Standard Approved by ANSI

4.2 Project 8: Ceiling-integrated fan mockups

One disadvantage of ceiling fans is that they only deliver high air movement within the area below the fan blades. An oscillating fan covers a larger floor area with air movement (Figure 75 shows the concept). It also creates transient air flow which has been found to benefit comfort (Zhao R. et al. 2004, Zhao X. et al.2006).

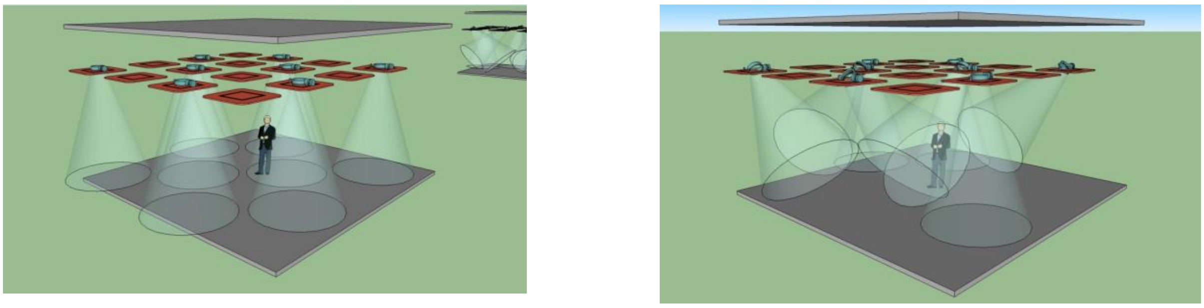


Figure 75 Concept of larger floor are coverage of oscillating fan vs. fixed fan

(images from Armstrong World Industries)

4.2.1 Dyson desk fan mocked up in ceiling

A Dyson desk fan was mocked up into the ceiling (Figure 76). The bladeless 33cm diameter induction ring was mounted at in a 2 x 2 ft ceiling panel. This way the fan can be easily moved to different locations by moving the panel. The Dyson fan oscillates through an angle of 90 degrees. Without oscillating, the area of air movement at desk level is roughly 2 times the diameter of the induction ring. With the oscillating, the dimension along the oscillation direction is about 8 times the diameter of the induction ring. Note: a Dyson fan oscillates along one direction, not two directions. Therefore, the swept area is a narrow oval shape.

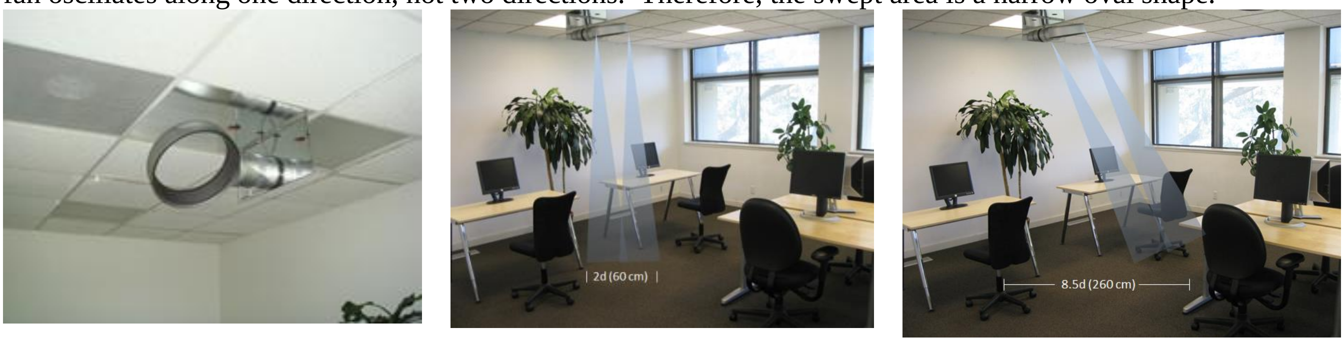


Figure 76 Dyson floor fan mock in ceilings (CBE environmental chamber)

4.2.2 Toshiba floor fan mocked up in ceilings

The Toshiba floor fan (see Table 1) oscillates in two axes. It sweeps through 60 degrees in the horizontal direction and 20 degrees on the vertical direction. When mocked up in the ceiling it is capable of sweeping a wide oval area (Figure 77).

Figure 77 Toshiba floor fan mocked up in the ceiling (CBE environmental chamber

4.2.3 Setup for measuring airspeed profiles for the two mock-up oscillating ceiling fans

A measurement cart with airspeed sensors was used to measure the airspeed profiles (Figure 78). The sensors are set at the ASHRAE standard heights of 0.1m, 0.6m, 1.1m.

Figure 78 Airspeed measurement setup

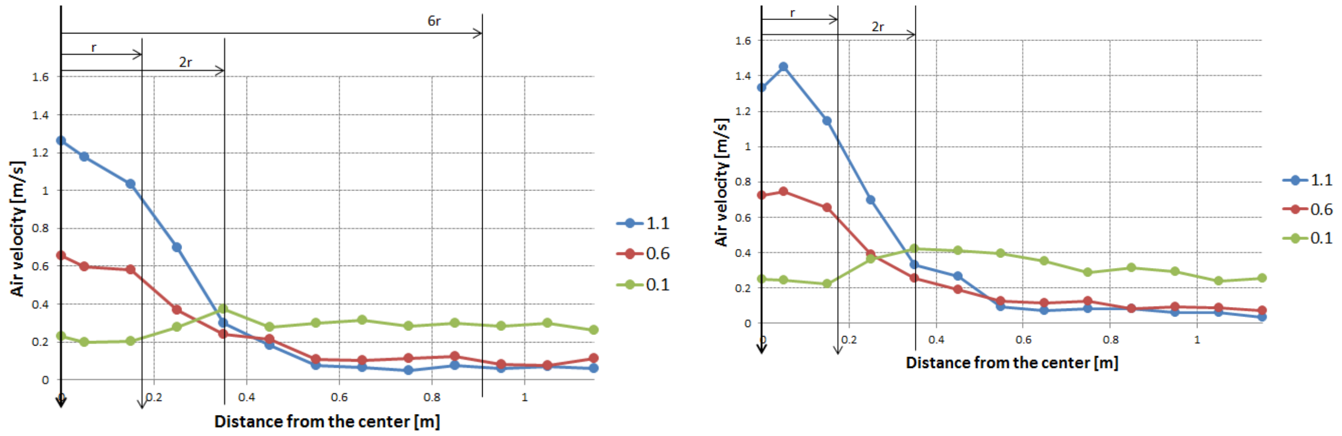
Airspeed was measured with omnidirectional spherical hotwire anemometers, with a response time of 2s and an accuracy of 0.05 of the 5 m/s full scale reading.

4.2.4 Results

4.2.4.1 Air speed profiles for a mocked-up Dyson ceiling fan (fixed vs. oscillating)

Airspeed (scalar velocity) profiles are shown for the Dyson fan without oscillation, at 0.1m, 0.6m, 1.1m height, and power levels minimum and medium (Figure 79). Both sets of profiles show that beyond 2 diameters of the induction ring (30 cm from the center of the ring), the airspeed at all three heights is low, only around 0.2 m/s.

The airspeed at the 0.1m height ankle height is higher (above 0.2 m/s). Outside of the 2 diameter circle, the airspeeds at the 0.6m and 1.1m height were low, only about 0.1 m/s.



minimum power

medium power

Figure 79 Air speed (scalar velocity) profiles for Dyson fan without oscillation

In contrast, with oscillation, the fan brought higher airspeeds to locations farther from the fan. Figure 80 shows airspeeds at 90 cm (6 diameters of the Dyson ring) away from the ring center, when the fan is at minimum fan power. The fan oscillates at a frequency of 13 seconds, so the airspeed peaked every 13 seconds. At the 1.1m height, the airspeed reached 0.6 – 1 m/s. At the 0.1m and 0.6m heights, airspeed was steadily about 0.2 m/s.

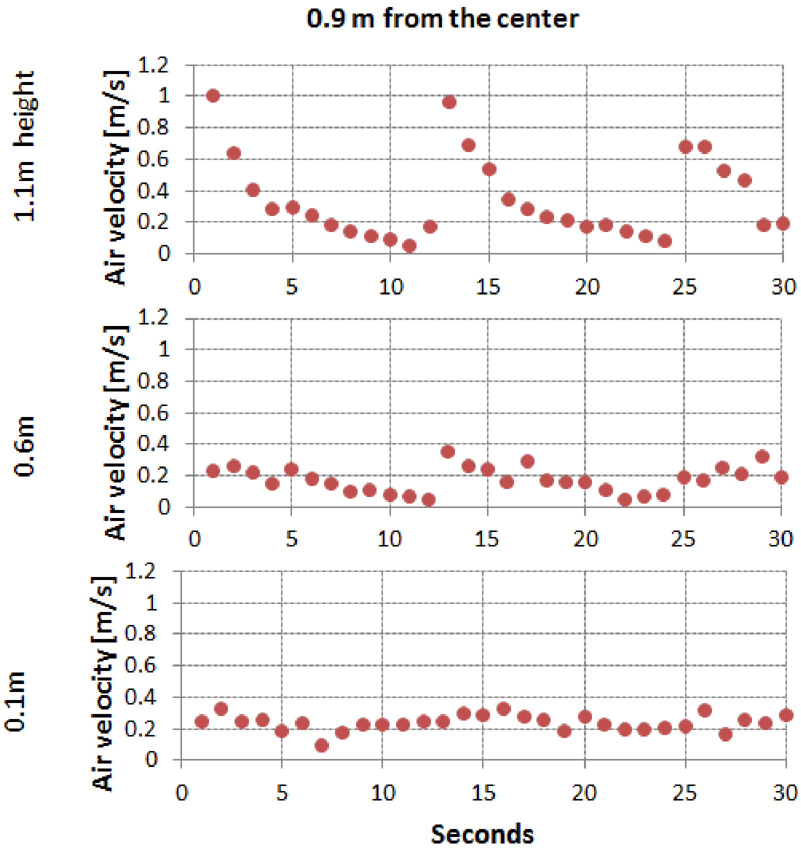


Figure 80 Airspeed at three measurement heights in oscillating mode

4.2.4.2 Airspeed profiles for a mocked-up Toshiba ceiling fan

Airspeed was measured for all the 20 configurations (Table 10) in 27 points where the subjects were located (described in project 3), and for 3 heights (1.1 m, 0.6 m, and 0.1m). In this report, only the airspeed values at the location 20 cm from the center of a desk for fixed fan configuration are reported (Table 18). For the configurations with the oscillating fan, the maximum air velocities for the three heights are comparable with value measured with the fixed fan, but there is a transitory phenomenon due to the fan oscillation. Figure 81 shows an example of measured data for the configuration “3 Oscillating Front”. The speed at 1.1m height peaked every 22 seconds. Airspeed sensors at 0.6m and 0.1 m were shaded from the air movement by the desk, so over time the airspeed at these levels was always low.

Table 18 Measured airspeeds and standard deviations

Configuration code	Mean value [m/s]/SD[m/s]		
	1.1 m	0.6 m	0.1 m
2 Fixed Front	0.68/0.16	0.51/0.11	0.24/0.06
3 Fixed Front	0.88/0.13	0.62/0.12	0.28/0.07
2 Fixed Side	0.70/0.21	0.76/0.11	0.34/0.09

3 Fixed Side	0.81/0.29	0.79/0.48	0.35/0.19
2 Fixed Below	0.66/0.20	0.54/0.25	0.24/0.12
3 Fixed Below	0.88/0.49	0.69/0.27	0.31/0.11

Note: "2 Fixed Front" means fan level 2, fixed fan position in front of the subject.

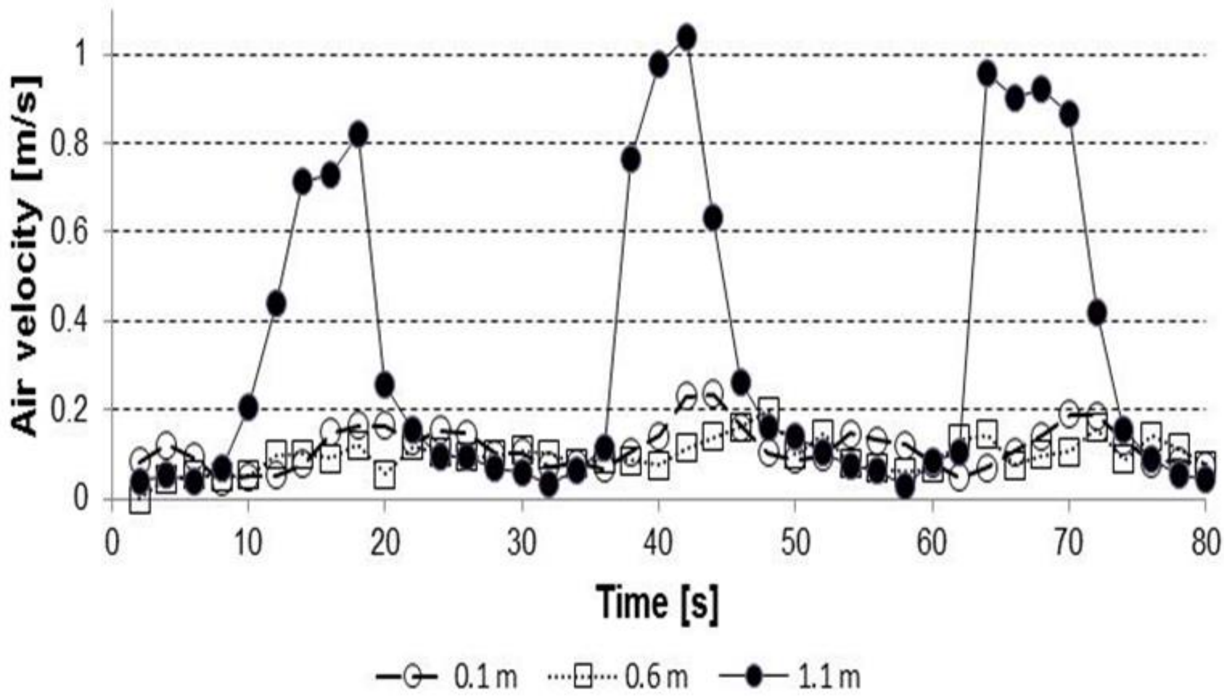


Figure 81 Airspeed (scalar velocity) profiles for the Toshiba mock-up ceiling fan

4.2.5 Conclusion

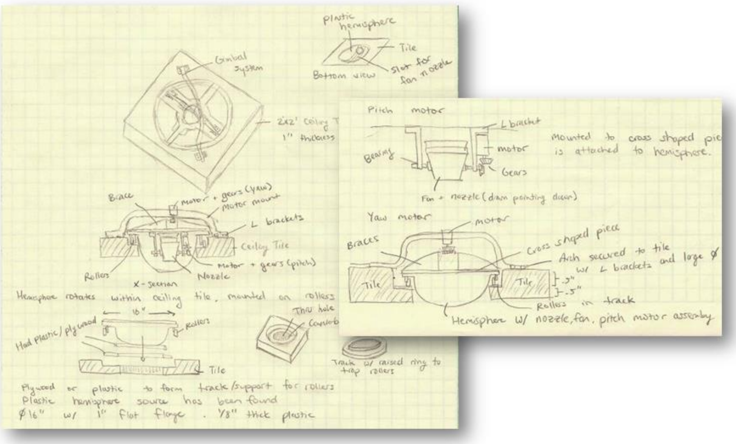
We mocked up two oscillating ceiling fans, and tested their velocity performances. The oscillating fans cover a wider area in the occupied zone than a fixed ceiling fan. Velocity at head level fluctuates significantly, going to still air for significant periods of time within the cycle. If this provides a comfort problem (see Project 3 described above), it would make sense to include some other source of minimal air velocity from a separate source.

4.3 Project 9: Occupant-tracking nozzle fan

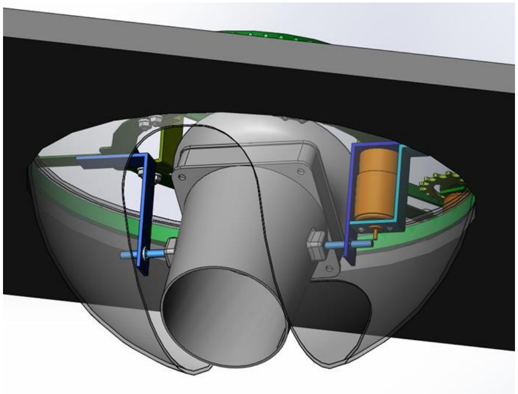
4.3.1.1 Introduction

Another way to overcome the disadvantage of ceiling fans which in fixed position only project air movement to a limited floor area, which in oscillating mode leave the occupant intermittently in still air, is to have the fan actively recognizing and tracking the occupants. This permits continuous airspeed at the occupant's location, even if they move about within their workplace.

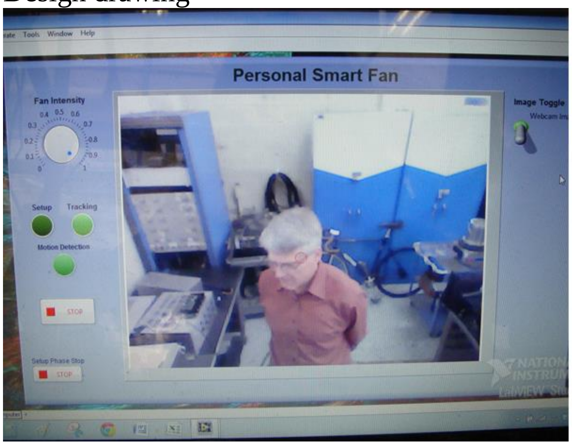
Working with Mechanical Engineering department, we designed and built a mock of such an occupant-tracking nozzle fan (Figure 82). The fan nozzle is mounted in a ball turret. This was completed and demonstrated internally in May 2013.



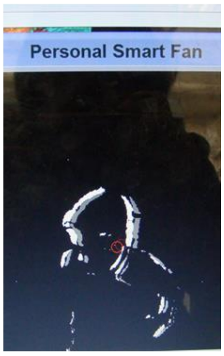
Design drawing

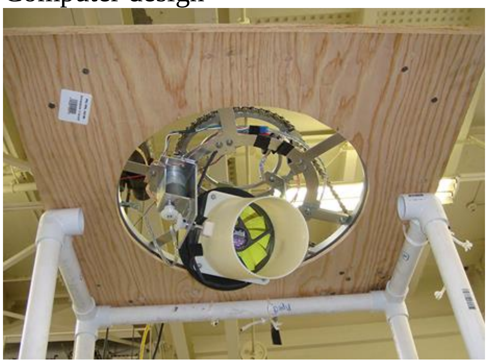


Computer design



Occupant recognition software



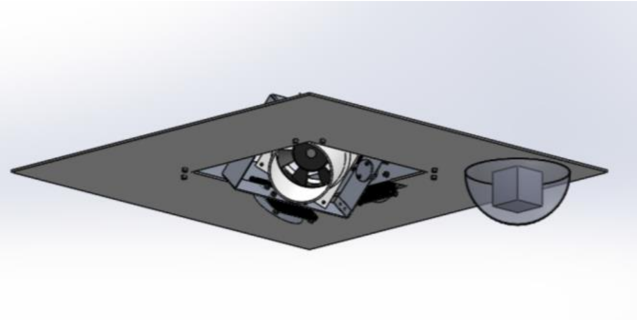


Version 1 occupant-tracking mockup

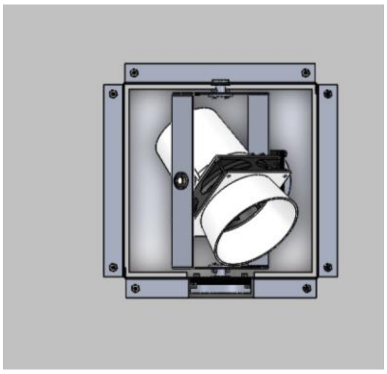
Figure 82 A mockup of occupant-tracking fan

In the fall 2013, we worked with another group of ME students on another variant of the tracking mechanism and software, and is building another prototype (

Figure 83). In the modified design, the fan is installed in a gimbal, not a ball turret, capable of more efficiently aiming the air jet anywhere within a wide area within its field of view.



Isometric view



Top view

Figure 83 Design of the modified occupant-tracking fan

We are applying for a provisional patent for the occupant-tracking nozzle fan. UC Berkeley requires us to keep the detailed design information confidential for now.

4.3.1.2 Conclusion

Occupant-tracking is a new idea developed by this team. It overcomes the disadvantage of ceiling fans which in fixed position only project air movement to a limited floor area. It actively recognizes and tracks the occupants. This permits continuous airspeed at the occupant's location. It turns off when no one is present. It can be applied in suspended ceiling tiles, or mounted in walls, furniture.

Since we have are applying for a provisional patent for the nozzle fan, UC Berkeley requires us to keep the detailed design information confidential for now.

4.4 Project 10: Evaluating heat transfer performance of radiant slab systems with acoustical panels and integrated fans

4.4.1 Background

Slab radiant systems are characterized by several benefits, such as good thermal comfort and low energy consumption, but also by frequently observed poor acoustic performance. The relatively high noise level is related to the exposed hard radiant surfaces. Using suspended acoustical ceiling panels reduces the noise level in the room, but also reduces the cooling and heating capacity of the radiant ceiling slab by obstructing the radiant exchange with the occupied space. Using ceiling panels with integrated fans could provide a possible solution to this problem. If designed properly, the fan could increase the air velocity close to the radiant ceiling surface, thereby increasing the convective heat exchange between the radiant ceiling and the room below.

4.4.2 Object

The objectives of this study are: (1) to investigate, through CFD modeling, the impacts of acoustical ceiling panels on radiant slab systems; and (2) to study the ability of a well-designed ceiling integrated fan to increase the ceiling convection coefficient, to verify if the improved convective heat exchange can offset the reduced radiation heat exchange caused by the presence of the acoustic panels.

4.4.3 Method

CFD model. The CFD model used for the simulations is presented in the figure below (Figure 84). The model, with 129 ft^2 of floor (and ceiling) area, represents a portion of a typical office. The load in the office is generated by one occupant, a computer, and a vertical heat source. The total internal load is 330 watts. The fresh air is delivered at 68° F through a diffuser located in the lower part of the wall, and extracted through the outlet located in the top part of the wall. The total airflow is 50 cubic feet per minute (cfm). The concrete temperature, at the level of the tubing, was fixed at 59° F.

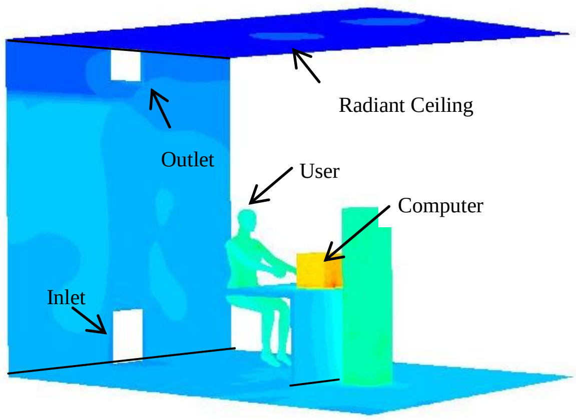
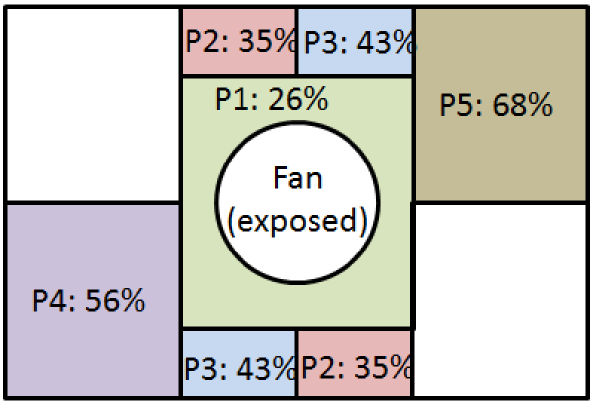
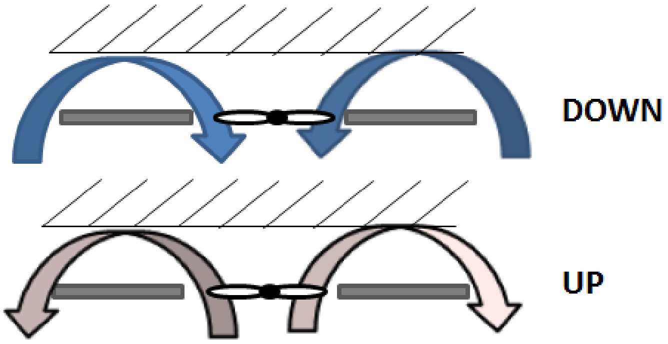


Figure 84 CFD model used for simulations

Configurations. Five different levels of acoustical panel coverage (26%, 35%, 43%, 56%, 68%), and two fan operative modes (fan blowing up or down) were simulated (Figure 85). A matrix of 15 possible combination were generated, plus the base case scenario without acoustical panels and fans. The air velocity right after the fan was set to 98 ft/min. The fan diameter is 3.2 feet. All boundary conditions were kept constant throughout the different configurations.



Five different levels of acoustical panel coverage (26%, 35%, 43%, 56%, 68%)



Fan configurations (fan blowing up or down)

Figure 85 Simulation configurations

4.4.4 Results

The heat flux from the radiant ceiling for each of the 15 configurations (Table 19), with and without acoustical panels, with and without fans, fan blowing up and down), is presented in Table 5. The ceiling heat flux for the base case scenario (no acoustical panel coverage, no fans) was calculated to be 27 W/m^2. Table 19 shows the comparisons with the base case scenario.

When the radiant ceiling is covered with acoustical panels up to 68%, the radiative and convective heat loss is reduced to 84% of the original heat flus without acoustical panels (Table 20). Adding fans in the ceiling, the heat flus increased dramatically, as much as 150% of the heat flus in the base case scenario without any acoustical panel coverage.

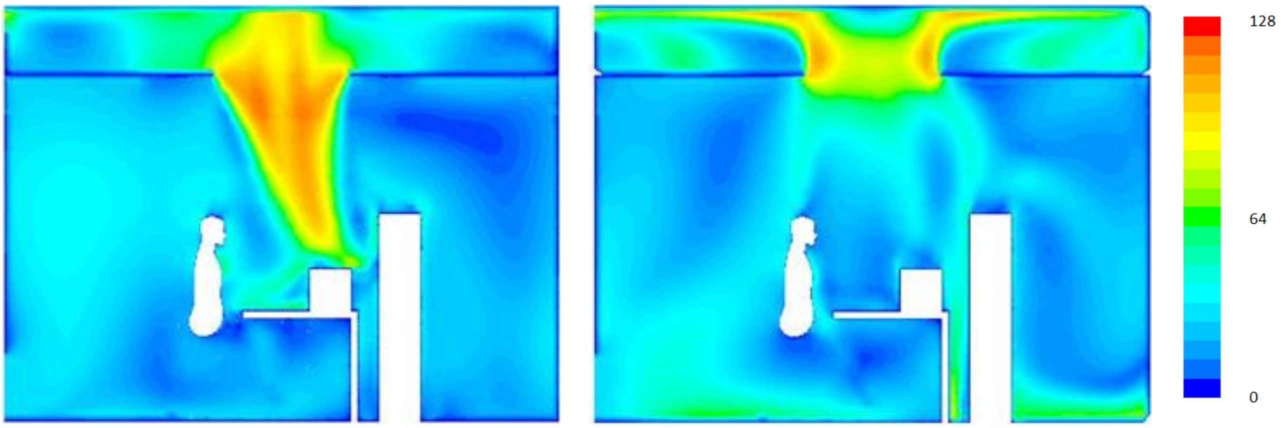
Table 19 Ceiling heat flux [w/m2]

Panels coverage	No fan	Fan down	Fan up
0%	27		
26%	25.9	39	38.9
35%	24.7	37.6	41.39
43%	22.7	37.7	41.6
56%	23.9	37.7	40.8
68%	24.1	35.9	41.19

Table 20 Ceiling heat fluxes compared with the base case

Panels coverage	No fan	Fan down	Fan up
0%	100%		
26%	96%	144%	144%
35%	91%	139%	153%
43%	84%	139%	154%
56%	88%	139%	151%
68%	89%	132%	152%

The heat flux increase from the ceiling is caused by the fans increasing the velocity between the ceiling and the acoustical panels (Figure 86). The upward-facing fan causes a bigger velocity increase and therefore results in the greater heat flux increase.



Fan down Fan up scales

Figure 86 Velocity distributions for fan blowing up and down configurations

There are several interesting results that need to be highlighted:

- Although the radiant ceiling was radiatively shaded as much as 68% by the acoustical panels, the heat flux reduction compared to the case without panels never exceed 16%. This result is in line with the literature (Weitzmann et al. 2008, Crocker et al. 2012), and comparable with the physics of this kind of system. A radiant ceiling exchanges 40% through convection and 60% through radiation. Since the acoustical panels mainly affect the radiant heat exchange, they do not greatly impact the performance of this system.

- The presence of ceiling fan, especially in the configurations with the fan blowing up, drastically increases the ceiling heat flux. Furthermore, the use of ceiling fans increases the air velocity in the occupied space, so the indoor temperature can be controlled at a higher setpoint. This opens a completely new scenario about the energy performance and the control of this type of radiant system.

4.4.5 Discussion

The result of this study and the literature on this topic clearly show that covering up to 68% of the radiant ceiling with acoustical panels has little effect on the heat exchange. Moreover, the use of ceiling fans has a far greater impact that more than makes up for the heat exchange reduction due to the presence of the panels. It can increase the heat exchange through the radiant ceiling by 50% over the unobstructed case. A certain amount of prudence is needed analyzing these results. The steady-state nature of a CFD simulation could have an effect on the magnitude of the results. For this reason more research needs to be done to better investigate and understand the phenomena behind the integration of radiant ceiling, acoustical panels, and ceiling fans. More research needs to be done to better investigate and understand the phenomena behind the integration of radiant ceiling, acoustical panels, and ceiling fans.

4.4.6 Conclusions

- Covering the radiant slab with suspended acoustical ceiling (up to 68%) does not significantly disrupt the heat flux to the slab.

- The use of ceiling fans has a greater impact than just making up for the heat flux reduction due to the presence of the panels. Adding fans in the ceiling, the heat flus increased dramatically, as much as 150% of the heat flus in the base case scenario without any acoustical panel coverage.

References

Crocker, C. and J. Hggins. 2012. Radiant heating and cooling systems: *A theoretical discussion*, *literature review*, Technical Report, Acoustics & Noise Research Group, University of British Columbia (*2012*).

Weitzmann, P., E. Pittarello, B.W. Olesen. 2008. The cooling capacity of the thermo active building system combined with acoustical ceiling, *Nordic Symposium on Building Physics*, Denmark, 2008

5 Energy saving analysis

5.1 Project 11: Analysis of energy savings enabled by use of fans

Abstract

The thermostat setpoint range (deadband) in office buildings impacts both occupant thermal comfort and energy consumption. Zones operating within the deadband range of temperatures require no heating or cooling, and the terminal unit airflow volume rate may be reduced to its design minimum. Wider deadbands allow energy savings, as well as allowing lower minimum airflows through the terminal. The extent of such savings has not been systematically quantified. Reference models representing standard HVAC and building design practice were used to simulate the impact of thermostat setpoint ranges on annual HVAC energy consumption. Heating and cooling setpoints were varied parametrically in seven ASHRAE climate zones and in six distinct medium-sized office buildings, each representing a new building design or a building controls retrofit. The minimum airflow volume rates through the VAV terminal units were also varied. The simulations are compared to empirical data from monitored buildings.

By increasing the nominal cooling setpoint of 72°F (22.2°C) to 77°F (25°C), an average of 29% of cooling energy and 27% total HVAC energy savings are achieved. Reducing the nominal heating setpoint of 70°F (21.1°C) to 68°F (20°C) saves an average of 34% of terminal electric heating energy. Satisfaction levels achieved with narrow temperature bands such as 70°F–72°F have been shown to be no higher than with wider temperature bands shifting seasonally within the range of 68°F–77°F (Arens et al. 2009, Zhang et al. 2011). Further widened temperature bands achieved with fans or personal controls can result in HVAC savings in the range of 30-70% (or more in coastal climates such as San Francisco). It is demonstrated that in order to fully realize energy savings from widening thermostat temperature setpoints, today's typical VAV minimum volume flow rates should also be reduced.

The energy use by the room fans themselves is very small. It is about 1% of the HVAC energy use.

5.1.1 Introduction

Typical office buildings equipped with an overhead variable air volume (VAV) system consume large amounts of energy to maintain their occupied spaces within temperature ranges that their designers and operators consider acceptable. Their thermostat setpoint ranges are often narrow, around 4F (2K), even though there is very little scientific evidence supporting such a range. Examination of the extensive ASHRAE RP-884 field study database has shown that thermal environments controlled to narrow temperature ranges do not result in higher occupant satisfaction than environments with wider ranges, such as 7-10F (4-6K) (Arens 2009). Wider temperature control ranges might therefore be implemented in some climates without a reduction in the occupants' thermal comfort. We aim to demonstrate, through a parametric simulation in several climates, the energy savings that result from implementing higher cooling setpoints and lower heating setpoints.

In addition, there are several ways to further widen the controlled temperature range while maintaining the same level of occupant thermal comfort. These may include provisions for air movement such as ceiling fans or desk fans to increase convective cooling of the occupant, foot warmers to provide heating, and heated and cooled seats and furniture surfaces. Tests of low-energy Personal Comfort Systems (PCS) have demonstrated high levels of thermal comfort in a wide range of ambient conditions (Zhang et al. 2010).

The primary benefit of widening the thermostat setpoint is to lessen energy consumption. This occurs as a result of zones spending more hours within the thermostat setpoint range and not calling for cooling or activating terminal heating coils. The throttling range of the VAV air flow volume is a key factor dictating how much time is spent inside the thermostat setpoint range. During periods of low internal heat loads, if a terminal unit cannot reduce its volume low enough, excessive cool air from the central system is delivered, pushing the zone temperature down, often to the heating setpoint. The minimum volume setpoint is often specified by HVAC designers according to longstanding rules of thumb about the diffuser's ability to mix cold supply air with room air, or for the terminal unit to accurately control itself. Such rules have recently been challenged and disproven, allowing a wider throttling range.

Changing thermostat setpoints, providing personal controls, and retrofitting VAV terminal minimum flow rates are easy and low-cost retrofits that typically do not require any upgrade to existing HVAC hardware. Each of these measures plays an important role in realizing both occupant comfort and energy savings. A portion of the simulations are thus dedicated to demonstrating the retrofit potential in existing buildings by using a reference model for buildings constructed after 1980. In these simulations the HVAC sizing and design are fixed independently of the changes in operation. We also simulate the case for new construction using a new building reference model and sizing HVAC equipment according to the widened temperature setpoint range. Further simulations demonstrate the relationship of the temperature setpoint range and VAV minimum flow setpoint fractions.

5.1.2 Methods

The whole-building energy and utility cost simulations were carried out with *EnergyPlus* version 7.2 [http://apps1.eere.energy.gov/buildings/energyplus/]. Reference models created by the U.S. Department of Energy (DOE) [http://www1.eere.energy.gov/buildings/commercial/ref_buildings.html] are used to represent realistic engineering practices and serve to simplify the assumptions made in the simulation study. By using these reference models, targeting medium-sized office buildings, and varying control setpoints parametrically we aim to achieve a high level of generality without creating a large number of energy models. In this study we target three domains of analysis using the Medium Office DOE reference model: (1) new construction in which each of the simulated zone heating and cooling setpoints is designed with appropriately sized HVAC equipment, (2) existing buildings constructed in or after 1980 in which only the zone setpoints are altered, and (3) existing buildings as in (2) in which the zone setpoints and maximum VAV terminal flowrates are altered as part of a low-cost controls retrofit. The base case setpoint range is 70°F – 72°F. The simulations and analysis were carried out for 7 cities, each representative of an ASHRAE climate zone. The cities and respective climate zones are Miami (1A), Phoenix (2B), Fresno (3B), San Francisco (3C), Baltimore (4A), Chicago (5A), and Duluth (7). The DOE reference buildings include models tailored specifically for each of these climates.

Upon execution of each simulation, EnergyPlus performs a detailed load calculation in order to size central and terminal equipment (e.g. the nominal capacity of central heating coils) as well as to fix control variables (such as the maximum VAV terminal flow rate) that determine how the equipment is operated during the simulation. This process is known as autosizing. In case (1) above, all equipment is autosized, representing a building that may be designed according to specific heating and cooling setpoints. In order to represent case (2), we fixed the sizing results yielded from the nominal case where the setpoint range is 70°F – 72°F, and altered only the heating and cooling setpoints in the remaining simulations. In case (3), the sizing results from the nominal case are held fixed, with the exception of VAV terminal maximum air flow rates, which are autosized. This assumption represents the ability to reduce maximum airflow settings in VAV terminals without any hardware modifications.

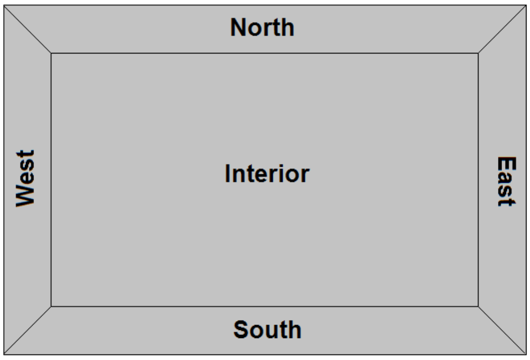
Recent research has discovered that the VAV Minimum Volume Setpoint (MVS) is a highly significant factor in determining the savings of thermostat setpoint adjustments (Taylor et al. 2012). A rule of thumb in engineering practice is to specify the MVS as a fraction of the VAV unit's maximum flow capacity. The DOE reference models use 30% for the MVS Fraction (MVSF), and standard engineering practices implement values as high as 50% (Arens et al. 2012). Flow rates at this level provide a significant amount of cooling, in effect continuing to cool the zone well below the cooling setpoint and often below the heating setpoint despite high outside air temperatures, a phenomenon known as overcooling [cite]. This restricts the energy savings that can be realized by increasing the cooling setpoint and/or decreasing the heating setpoint, because less time is spent in the region between the setpoints (also known as the deadband) where air is supplied at the minimum volume. Thus we have repeated the simulations representing the three domains above, changing only the VAV MVSs to 10%. Earlier research has shown that VAV MVSs can be reduced to approximately 10% (or less), and still provide adequate mixing and fresh air (Arens et al. 2012). By simulating these cases, we aim to demonstrate the energy savings potential of reducing the VAV MVS as well as the impact of the VAV MVS on energy savings as a result of implementing a wider thermostat setpoint range. Table 21 lists the final model and simulation types.

Table 21 simulation configurations

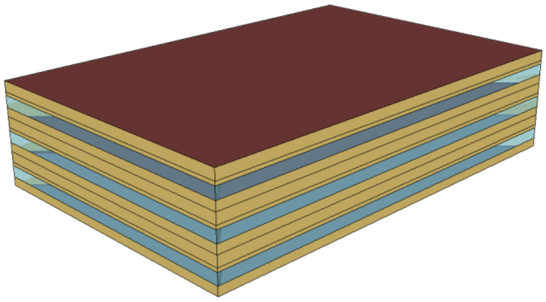
Model Type	VAV MVS fraction	Vintage	VAV capacity sizing
High-New-VAVAuto (1)	High (30%)	New construction	Yes
High-Existing-VAVAuto (2)	High (30%)	Post-1980 construction	Yes
High-Existing-VAVFixed (3)	High (30%)	Post-1980 construction	No
Low-New-VAVAuto (4)	Low (10%)	New construction	Yes
Low-Existing-VAVAuto (5)	Low (10%)	Post-1980 construction	Yes
Low-Existing-VAVFixed (6)	Low (10%)	Post-1980 construction	No

The post-1980 and new construction DOE reference building models adhere to ASHRAE Standards 90.1-1989 and 90.1-2004 respectively, and are identical with few exceptions. Depending on the climate, these exceptions include fan and DX coil efficiency, lighting loads, envelope insulation thickness, glazing U-values, and/or infiltration rates. The properties and diagrams below are common to both vintages and all climates.

The HVAC system is VAV with terminal electric reheat coils. There are three floors and one packaged air handling unit per floor, each containing a direct expansion (DX) coil, a gas heating coil, and a variable volume supply fan. The building model is a typical 5-zone floor plate (Figure 87) with a large interior zone and perimeter zones with depth 15ft. Equipment loads peak at 1 W/ft^2, and occupancy at 200 ft^2/person. Ribbon windows span the length of the façade, with a window-to-wall ratio of 33%.



Plan view of the reference model



Isometric view of the reference model

Figure 87 Plan view of the reference model

The simulations consider increasing the cooling setpoint and decreasing the heating setpoint independently. In other words, the heating setpoint is fixed at the nominal value of 70°F while the cooling setpoint is varied in the range of 72°F – 80°F. Similarly, the cooling setpoint is fixed at 72°F while the heating setpoint is varied in the range of 64°F – 70°F. Note that the heating setpoint can affect the behavior of the cooling system and vice versa. To carry out the parametric simulations, the software *JEPlus* was used [http://www.iesd.dmu.ac.uk/~jeplus/]. This software allows the user to parameterize fields in an *EnergyPlus* model and specify a discrete set of values for these fields. Upon execution, the set of values will supply the parameterized fields in the model, and the simulations are automated. In our case, the heating and cooling setpoints during occupied hours are parameterized in the reference models for each climate. Summary results were collected and hourly results stored for detailed zone temperature analysis. A total of 1,638 simulations were carried out as a result of 7 climates, 6 model types, and 39 distinct setpoint combinations (including 29 cooling setpoints, 11 heating setpoints, and 1 baseline combination).

A smaller set of simulations were carried out to demonstrate the effect of simultaneously increasing the cooling setpoint and decreasing the heating setpoint. 7 distinct temperature setpoint ranges were considered in this analysis: (69°F – 74°F), (68F – 76°F), (67F – 78°F), (66°F – 80°F), (65F – 82°F), (64F – 84°F), and (63°F – 86°F). As in the main analysis, these simulations are carried out for 7 climates and 6 model types, totaling 294 simulations.

5.1.3 *Results*

In model type 3, High-Existing-VAVFixed, changes to the cooling setpoints have resulted in a distinct lack of energy savings compared to the other model types. In this model type no changes are made to the VAV system as the setpoint is increased, preserving the design minimum and maximum flow rates according to the baseline setpoint range. The savings are thus constrained by the high rate of consumption occurring while the VAV units operate at minimum volume. The cooling delivered by the minimum air volume will prevent the zone temperature from reaching the cooling setpoint.

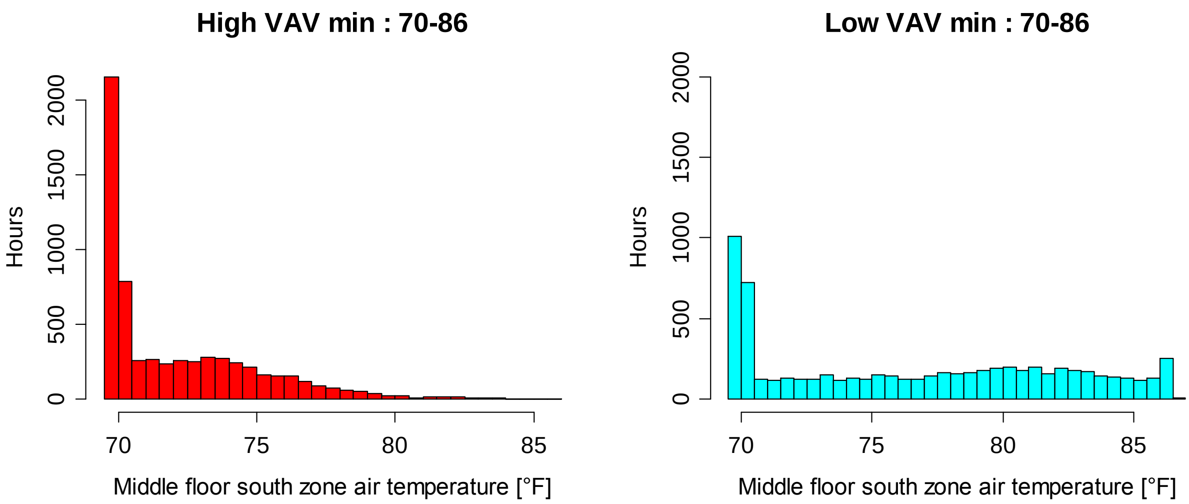


Figure 88 Zone air temperature distributions with 30% and 10% minimum flow rate for a wide thermostat setpoint range

Figure 88 shows the behavior of this model type in Chicago for a very large thermostat setpoint range of 70°F – 86°F. The left histogram represents the zone temperature distribution during occupied hours for a middle-floor south-facing zone with a 30% minimum flow rate. The first two bins show that more than half of annual occupied hours are spent at the heating set point of 70°F, indicating that this condition is often caused by unnecessary cooling. Conversely, in the low VAV minimum case, the zone temperature varies freely according to the internal load and climate conditions. The cooling delivered in the high VAV minimum case prevents the zone temperature from staying inside the widened setpoint range, thus requiring constant reheat and not saving energy as a result.

All other model types exhibit comparable cooling energy savings as the cooling setpoint is increased (Table 22). In the subsequent aggregates and analysis of individual climates and end uses, we exclude this model type, while the remaining five represent effective retrofit strategies for implementing changes to the thermostat setpoints.

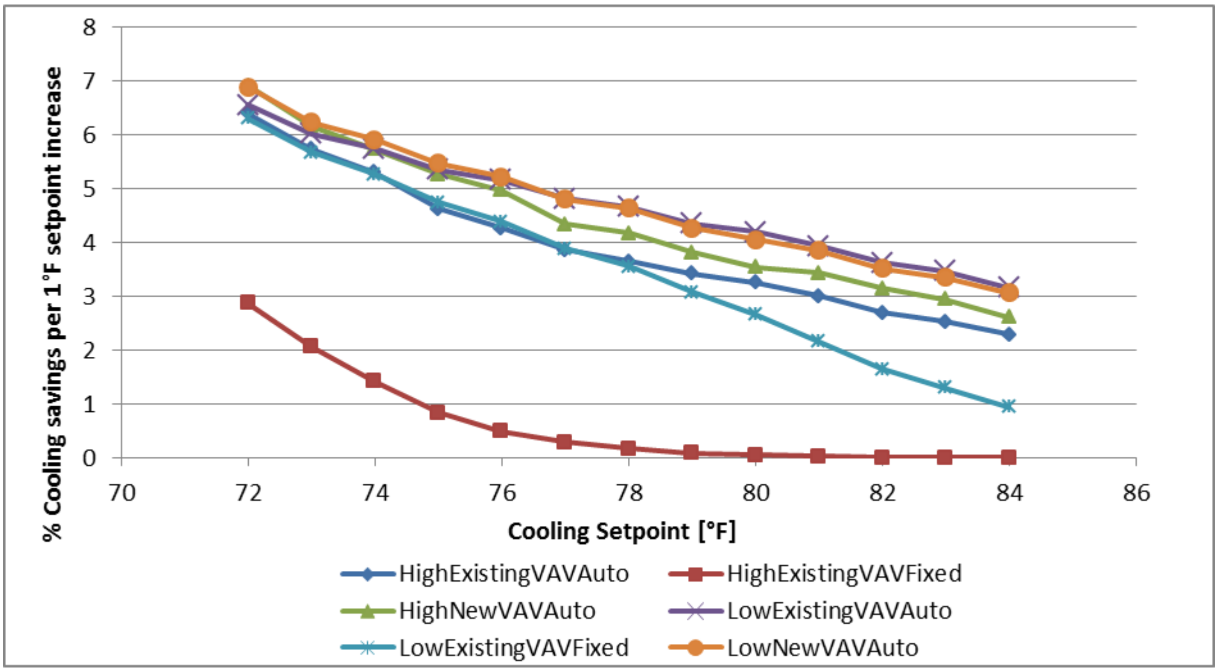


Figure 89 Incremental cooling HVAC savings with raised setpoints

Table 22 Energy savings when raising cooling setpoint

	HVAC Savings [kBTU/sf-year]			*HVAC Savings [%]*		
Cooling Setpoint [°F]	*Mean*	*Maximum*	*Minimum*	*Mean*	*Maximum*	*Minimum*
72				**Baseline**		
74	2.80	5.19	0.25	13	26	1
76	4.93	8.81	0.45	23	45	1
78	6.57	11.50	0.73	31	58	2
80	7.82	13.57	0.98	37	66	3
82	8.80	15.14	1.16	42	70	4
84	9.62	16.40	1.27	46	73	4
86	10.36	18.07	1.32	50	77	4

The same effect is not present in simulations in which the heating setpoint is decreased, and all model types exhibit significant energy savings. The proportions of terminal electric heating savings are shown in (Table 23), and are comparable for all model types. Thus, changes to the heating setpoint for purposes of energy savings may be implemented without major changes to nominal VAV operation. However, as shown in (Arens et al. 2012), lowering VAV minimum volume setpoints may save heating energy. The benefit of extending both the heating and cooling setpoints is summarized in Figure 86.

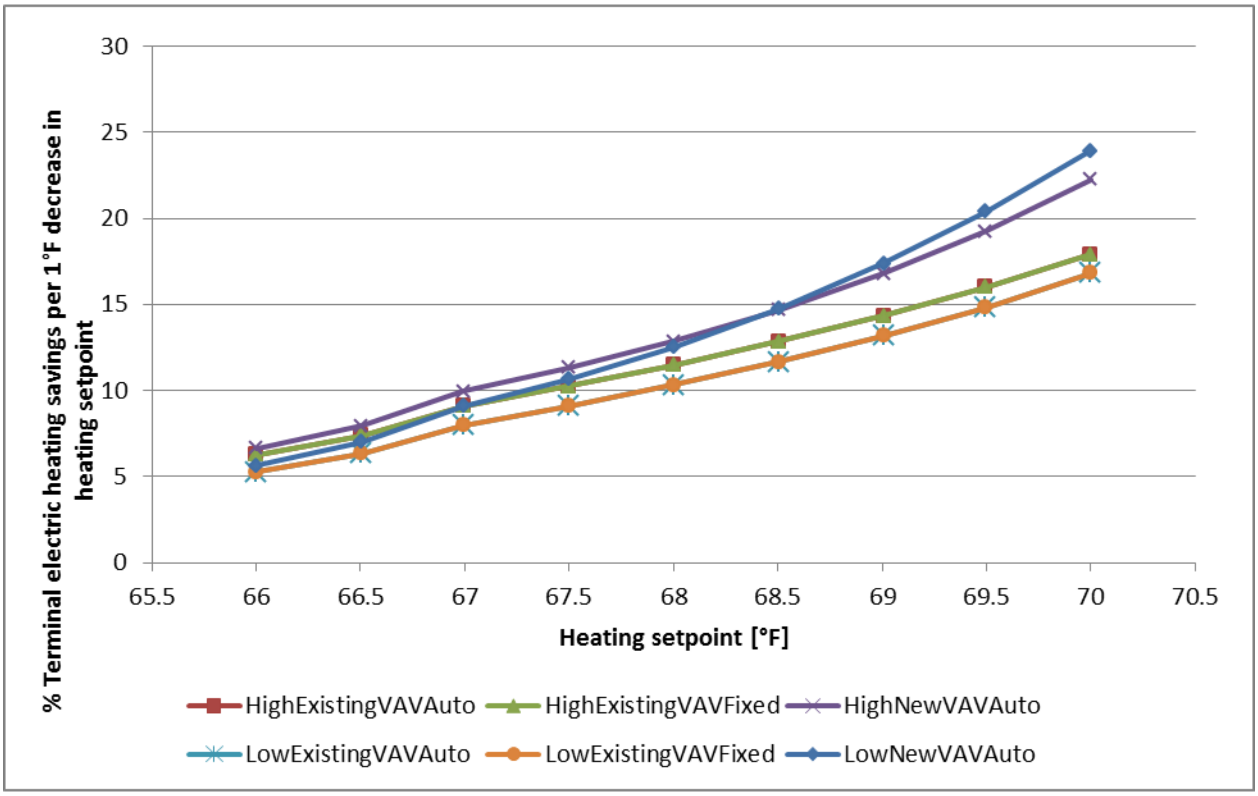


Figure 90 Incremental heating HVAC savings with reduced heating setpoints

Table 23 Energy savings when lowering heating setpoint

	HVAC Savings [kBTU/sf-year]			*HVAC Savings [%]*		
Heating Setpoint [°F]	*Mean*	*Maximum*	*Minimum*	*Mean*	*Maximum*	*Minimum*
70.0				**Baseline**		
69.5	0.60	1.36	0.02	3	8	0
69.0	1.12	2.58	0.04	5	15	1
68.5	1.61	3.76	0.06	8	22	1
68.0	2.06	4.85	0.07	10	29	1
67.5	2.46	5.83	0.07	12	34	2
67.0	2.83	6.72	0.08	13	40	2
66.5	3.15	7.48	0.08	15	44	2
66.0	3.44	8.18	0.08	16	48	2
64.7	4.07	9.59	0.09	19	57	3
63.8	4.41	10.29	0.09	21	61	3

For each setpoint, 35 simulation results in 7 climates and 5 model types were aggregated to create the summary in Figure 88.

The maximum HVAC cooling savings as a result of increasing the cooling setpoint occurred in the Miami climate, while the minimum occurred in Duluth. The magnitude of the temperature difference between the indoor environmental control conditions and the outside air conditions is high during Miami's summer. Accordingly, the highest heating savings occurred in Duluth, and the smallest in Miami.

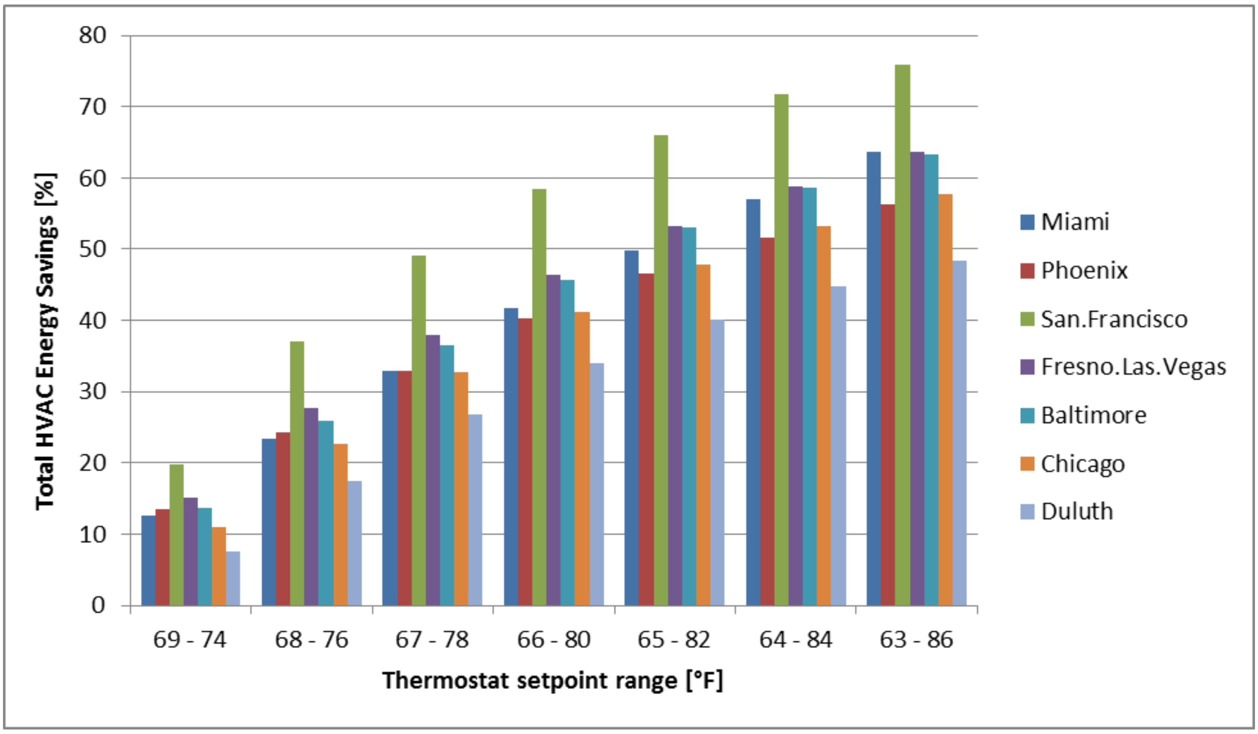


Figure 91 HVAC energy savings for extended setpoint ranges relative to baseline range [70°F – 72°F]

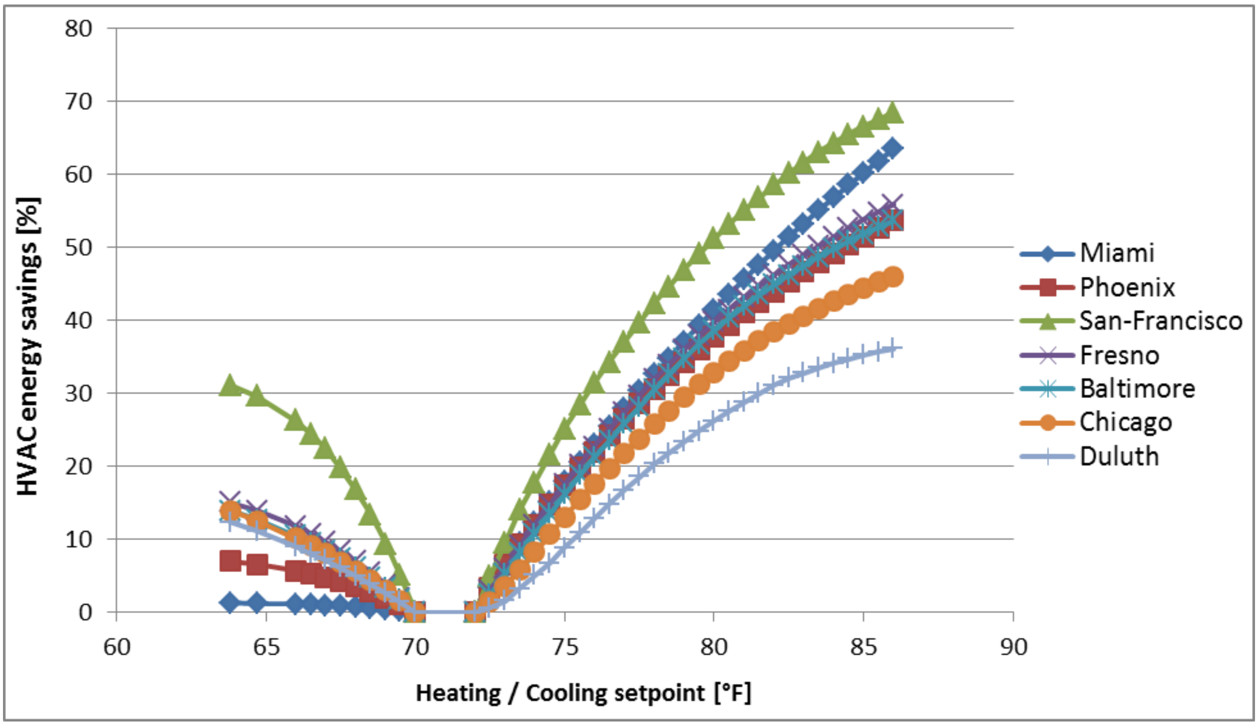


Figure 92 Aggregated HVAC savings from decreased heating setpoints and increased cooling setpoints in representative climates

5.1.4 Discussion: Empirical corroboration

The high terminal airflow rates and overcooling in these simulations are corroborated by recent empirical data collected at the Yahoo! Campus. A field study of the energy impact and indoor environmental effects of the MVSF was carried out in several buildings, in which periods of low and high MVSF settings were examined. We will look specifically at a representative south-facing perimeter zone with a VAV reheat terminal. During the high MVSF period, the VAV terminal has a maximum of 2,000cfm and a minimum flow rate of 600cfm or 30%. During the low MVSF period, the minimum flow rate was changed to 385cfm or 19%. These periods are compared respectively to the high and low MVSF simulations. In a middle floor south zone of the simulation model, the maximum flow rate is 2130cfm, the minimum flow rate is 639cfm (30%) in the high minimum case, and 213 (10%) in the low minimum case. All simulated data represents annual hourly data. Empirical data collected at the Yahoo! site includes measurements from November 2010 to August 2012, during which the MVSF was toggled between high (30%) and low (10%).

In both simulated and empirical data for a high minimum flow mode, the zone airflow remained at or very close to the minimum setpoint. This suggests that the minimum setpoint tends to exceed the cooling load in the space, and often continues to cool the space to the heating setpoint, which in turn activates the terminal reheat coils. In the simulated high minimum case, the zone spends almost 70% of hours at the heating setpoint, often due to overcooling. The low minimum simulated case shows much less time being spent at the heating setpoint due to overcooling, and modulates the flow rate to meet the load as necessary.

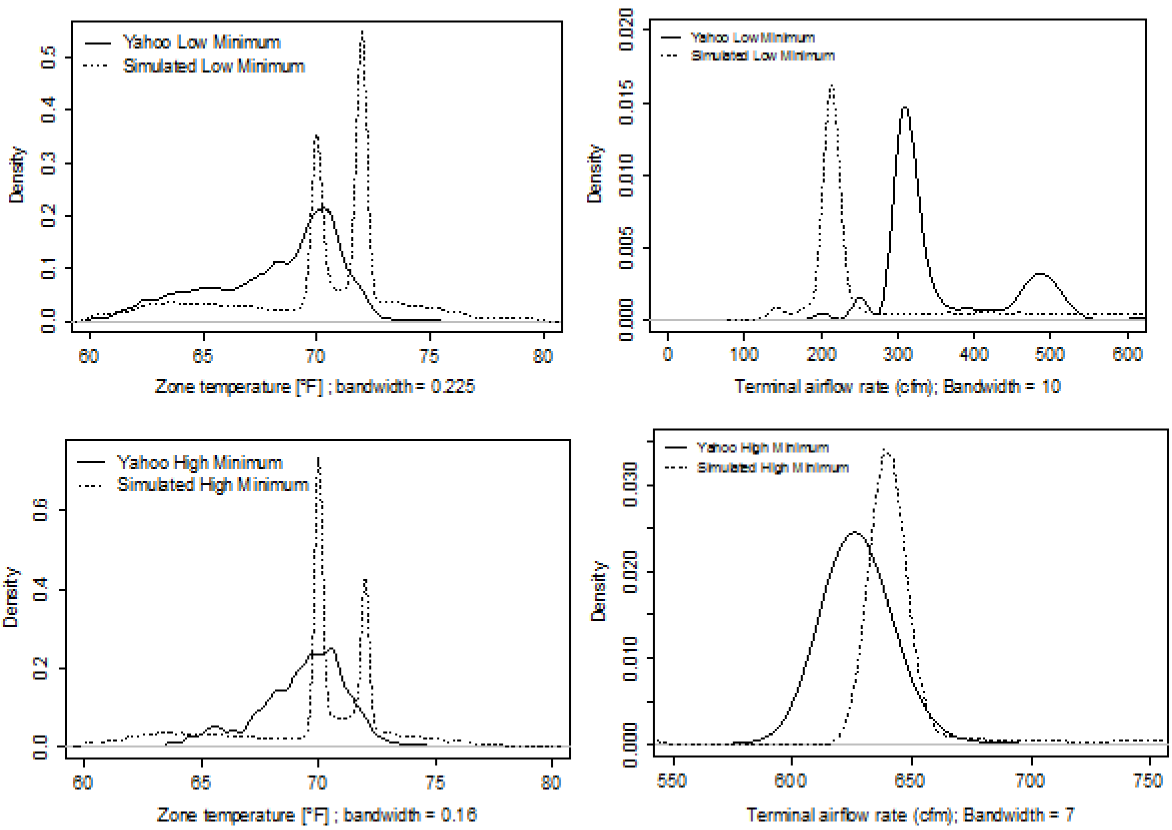


Figure 93 Density charts comparing simulated data to empirical data collected in the Yahoo! campus showing the cooling effect of air delivered at the minimum volume setpoint

5.1.5 Conclusions

In a large parametric simulation study of seven climates and six model types, we examined the benefit of widening thermostat heating and cooling setpoints to save energy in a typical medium office building. Comfort in the warm seasons would be maintained by air movement cooling indoors. The energy required for this is negligibly small, on the order of 1% of the HVAC system's energy.

If implemented correctly, a widened thermostat setpoint range results in significant HVAC energy savings. When the VAV boxes have adequate throttling range as a result of appropriately low minimum volume setpoints, the zone temperature will often remain within the heating and cooling setpoints, resulting in less heating and cooling required by the zone. If the minimum volume setpoints are high, the cooling delivered by this volume will unnecessarily cool the zone, often to the heating setpoint. This is the costly phenomenon of overcooling and must be avoided. A simple controls retrofit of the VAV box minimum volume setpoints can usually remedy this problem. A large field study conducted on the Yahoo! campus showed that the VAV minimum volume setpoints can cause the zone to be cooled significantly when cooling is not necessary.

Hot climates will see more benefit from increased cooling setpoints, whereas cold climates see more benefit from decreased heating setpoints. Temperate climates such as San Francisco may see great benefit from a widened thermostat setpoint range. In climates with a warm season, increasing the cooling setpoint by 5°F can result in HVAC savings in the range of 20%. In climates with a cold season, decreasing the heating setpoint by 5°F can result in HVAC savings in the range of 15%. The benefit is cumulative, and small incremental changes to the setpoints result in proportional savings. In practice, the type of heating or cooling system will have a large impact on actual energy savings resulting from this method. We considered a system with a central chiller and terminal electric coils.

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6 Standard 55 activities

6.1.1 Elevated air movement (Standard 55 2010) implemented in the CBE comfort webtool

CBE independently developed during this project period a Web-based thermal comfort calculator with unique graphic and analytical capabilities. The calculator was programmed to exactly match the requirements of ASHRAE Standard 55-2010, and of the existing ASHRAE Windows-based Comfort Tool software. Once implemented, the CBE Web-tool's graphic display revealed a major logical bug in the air movement provisions of the standard and in the ASHRAE software. This bug was then fixed in the CBE Web Tool, and in the preparation of the new ANSI/ASHRAE Standard 55-2013, the written description of the process was corrected. The CARB project assisted with the latter.

The Web Tool is available at: http://smap.cbe.berkeley.edu/comforttool . It has been adopted by a large number of users worldwide: it has received 1500 visits per month over the last year; since its launch in October 2012 there have been 23,090 visits and 13,060 unique visitors. In January 2014 ASHRAE adopted the Web Tool as the official ASHRAE Standard 55 compliance tool; the details are now under consideration at ASHRAE. CBE will maintain and perform future upgrades to the Tool.

The tool has seamlessly implemented the elevated air movement for comfort (ASHRAE Standard 55, 2010, -2013). The following screen shot from the Web Tool (Figure 94) shows the comfort zone in psychrometric chart format, shifted to the right 4.5C (8F) as a result of a modest level of elevated air movement (0.4 m/s; 80 fpm; 1 mph). 4.5C is the cooling effect of 0.4 m/s air movement over still air (0.15m/s).

To point out the relevance of this cooling effect: if a Fresno building's thermostat setpoint for cooling is increased 4.5C, its total HVAC energy use (cooling+heating) is reduced 40%. For a San Francisco building, the total HVAC energy use is reduced 52%. (Hoyt et al. 2008).

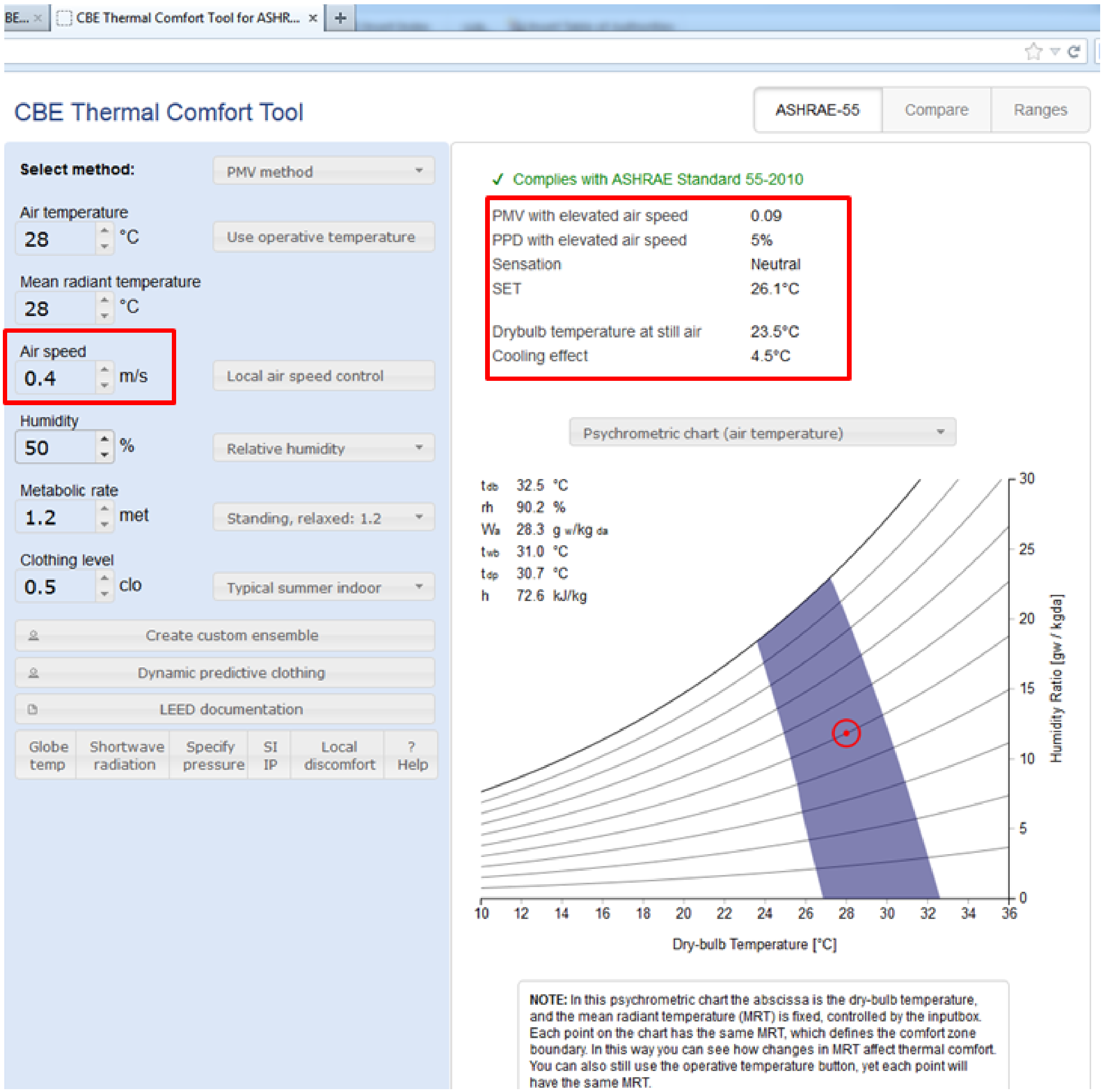

Figure 94 Web-based thermal comfort calculator

6.1.2 ASHRAE Standard 55 rewritten in code language

The elevated air movement provisions in Std 55 Section 5.3 *('Analytic comfort zone method')* are essential for the use of fans for cooling within the occupied zone. The provisions had been introduced in the 2010 version of the standard, which was (as in all previous versions of the standard) in informative language unsuitable for adoption in building codes.

In 2011-2013 ASHRAE mounted an extensive effort to rewrite Std 55 in normative code-compliant language. The Standard and its appendices were subjected to the ANSI consensus process with extensive public review and resolution of all comments. ANSI/ASHRAE Standard 55-2013 is being published December 2013.

PI Edward Arens was the lead for the Section 5.3 and Section 5.4 air movement provisions, and for two relevant technical appendices (B and G) defining the SET model and the procedure for using SET to evaluate cooling effect of air movement. He was also lead author of Section 7 and its appendices, on methods for measuring and evaluating comfort in existing buildings. These properly account for the comfort effects of air movement, when present, in buildings.

(Quote below: Section 5.3 air movement text reproduced from the new 2013 Standard 55:)

> **5.3.3 Elevated Air Speed.** This section is permitted to be used to increase the maximum allowable operative temperature and maximum allowable average air speed (V_a) determined from Sections 5.3.1 and 5.3.2, provided that the conditions described in Sections 5.3.3.1 and 5.3.3.2 are met.
>
> The Standard Effective Temperature (SET) model in the *ASHRAE Thermal Comfort Tool*[4] is used to evaluate all cases of comfort under elevated air speed above 0.2 m/s (40fpm). Figure 5.3.3A represents two particular cases of equal skin heat loss contours computed by the SET model and shall be permitted as a compliance method for the conditions specified in the figure.
>
> ***Notes:***
>
> a. The SET model is available as part of the *ASHRAE Thermal Comfort Tool*,[4] as described in Informative Appendix G of this standard. Any other codings of the SET model must be validated against this code.
>
> b. The flowchart in Figure 5.3.3B describes the steps for determining comfort under elevated air speed.

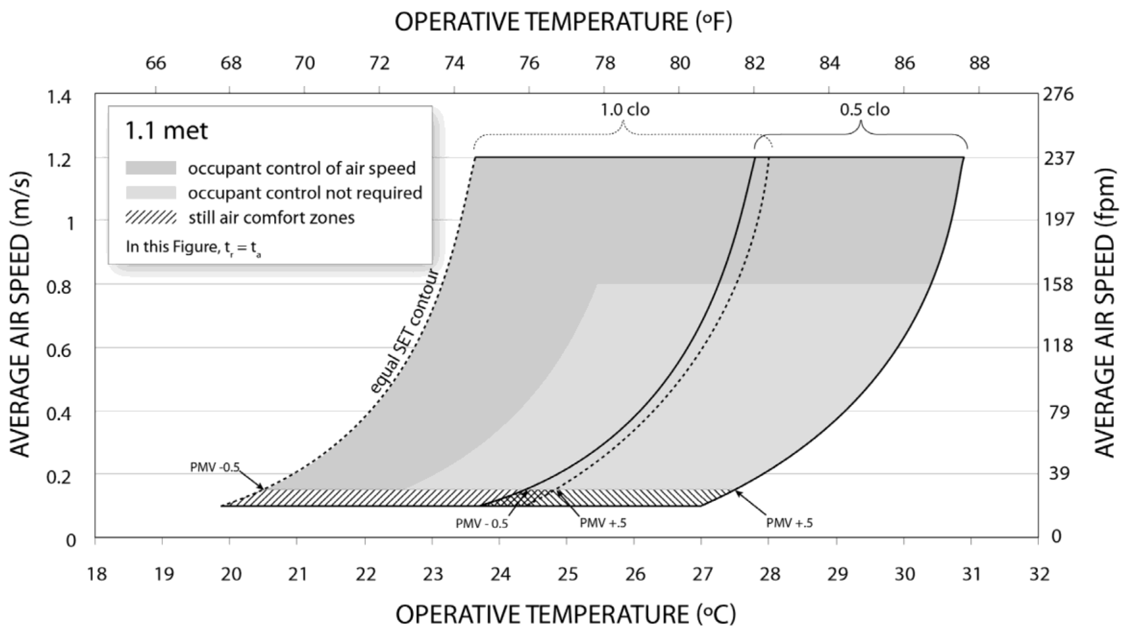

FIGURE 5.3.3A Acceptable ranges of operative temperature and average air speeds (*V_a*) for the 1.0 and 0.5 clo comfort zone presented in Figure 5.3.1.1, at humidity ratio 0.010.

5.3.3.1 Limits to Average Air (*V_a*) Speed with Occupant Control. When control of local air speed is provided to occupants, the maximum air speed shall be 1.2 m/s (240 fpm) for the SET model and Figure 5.3.3A.

When using either method, control shall be directly accessible to occupants and be provided either (a) for every six occupants or fewer or (b) for every 84 m2 (900 ft2) or fewer. The range of control shall encompass air speeds suitable for sedentary occupants. The air speed should be adjustable continuously or in maximum steps of 0.25 m/s (50 fpm) as measured at the occupant's location.

Note: These limits are shown by the fully bounded area for each clothing level in Figure 5.3.3A.

Exception: In multi-occupant spaces where groups gather for shared activities, such as classrooms and conference rooms, at least one control shall be provided for each space, regardless of size. Multi-occupant spaces that can be subdivided by moveable walls shall have one control for each space subdivision.

The air speed control must extend to still air as measured at the occupant's location and be adjustable continuously or in maximum steps of 0.25 m/s (50 fpm) as measured at the occupant's location.

Exception: Above activity levels of 1.3 met, the 1.2 m/s (240 fpm) limit does not apply.

5.3.3.2 Limits to Average Air Speed (*V_a*) without Occupant Control. If occupants do not have control over the local air speed meeting the requirements of Section 5.3.3.1, the following limits apply to the SET model and Figure 5.3.3A.

a. For operative temperatures above 25.5°C (77.9°F), the upper limit to average air speed (V_a) shall be 0.8 m/s (160 fpm).

b. For operative temperatures below 22.5°C (72.5°F), the limit to average air speed (V_a) shall be 0.15 m/s (30 fpm).

c. For operative temperatures between 22.5°C and 25.5°C (72.5°F and 77.9°F), the upper limit to average air speed (V_a) shall follow the curve shown between the dark and light shaded areas in Figure 5.3.3A. It is acceptable to approximate the curve in I-P and SI units by the following equation:

$V = 50.49 – 4.4047\ t_a + 0.096425(t_a)2$ (m/s, °C)
$V = 31375.7 – 857.295\ t_a + 5.86288(t_a)2$ (fpm, °F)

Note: These limits are shown by the light gray area in Figure 5.3.3A

Exception: Above activity levels of 1.3 met, the limits in Section 5.3.3.2 do not apply when using the SET model and Figure 5.3.3A.

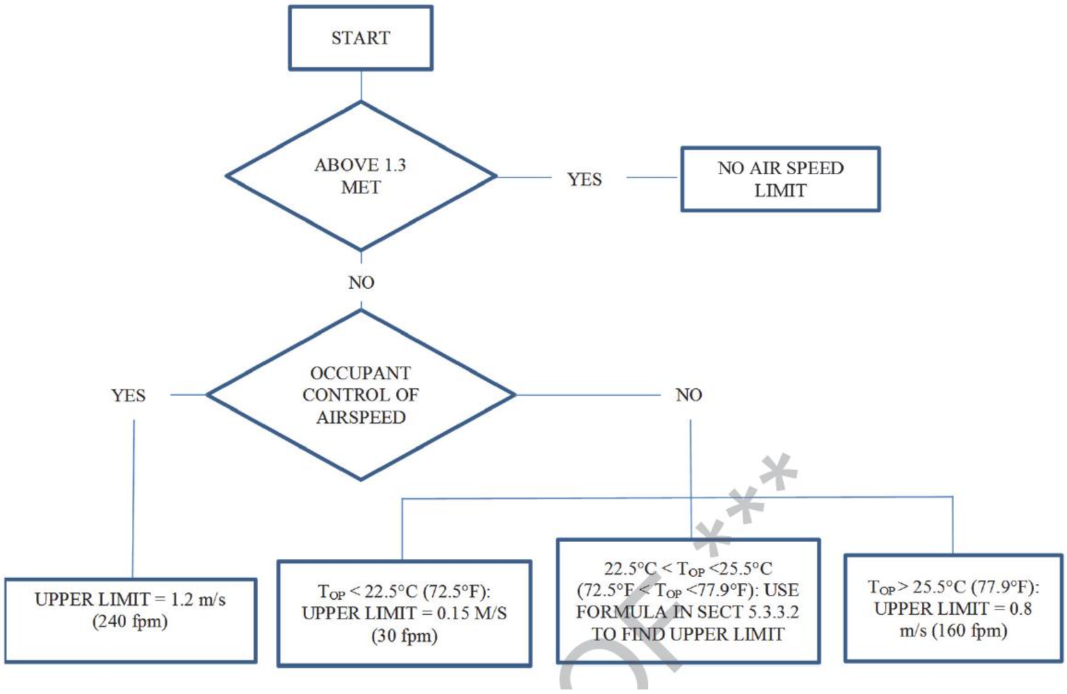

FIGURE 5.3.3B Flowchart for determining limits to airspeed inputs in SET model.

(End of quote from ANSI/ASHRAE Std 55-2013)

One of the provisions of the new standard is useful for design of indoor fan systems. The 'air speed' affecting comfort is now defined throughout Std 55 as 'average air speed', formally defined as the arithmetic average of three heights (ankle, midbody, and neck). This averaging has the effect of allowing the the fan system to include higher maximum local airspeeds in the occupied zone, since flows from fans are rarely equally high at all three levels. The designer of a ceiling fan system now has more flexibility than in the past when a single-point maximum speed was in place, since airflow maxima at a point are difficult to predict .

6.1.3 Elevated air movement accounted for in the Standard 55 Adaptive Model

ANSI/ASHRAE 55 Section 5.4 provides an alternative path to determining thermal comfort in naturally ventilated buildings. Termed the 'Adaptive Model', it is based on empirical results of comfort field studies in buildings in which occupants had control of windows. The comfort zone extends to higher temperatures than the analytical comfort zone method in Section 5.3. This is due to several factors, such as adaptive clothing behavior and beneficial effects of having a sense of control, but also including the physical effect of access to higher indoor air movement via the windows. The relative effect of these adaptive factors had not been determined.

During this project, we analyzed the actual measured air speeds in each of the field studies underlying the Adaptive Model, and found that the values were on average low. Higher air speeds when measured were associated with higher acceptable temperatures. We therefore introduced an airspeed correction to the Adaptive Model using the same elevated airspeed method used in Section 5.3. The corrected elevated temperatures match those obtained using the Analytic Method in Section 5.3, as well as the field data. The correction was introduced and shepherded through the ANSI/ASHRAE standards process, and is now incorporated in code-compliant normative language in Standard 55-2013.

(Reproduction of the Section 5.4 airspeed provisions:)

5.4.2.3 The following effects are already accounted for in Figure 5.4.2 and the equations in Section 5.4.2.2, and therefore it is not required that they be separately evaluated: local thermal discomfort, clothing insulation (I_{cl}), metabolic rate, humidity, and air speed.

5.4.2.4 If t_0 > 25°C (77°F), then it shall be permitted to increase the upper acceptability temperature limits in Figure 5.4.2 and the equations in Section 5.4.2.2 by the corresponding Δt_0 in Table 5.4.2.4.

TABLE 5.4.2.4 Increases in Acceptable Operative Temperature Limits (Δt_0) in Occupant-Controlled, Naturally Conditioned Spaces (Figure 5.4.2) Resulting from Increasing Air Speed above 0.3 m/s (59 fpm)

Average Air Speed (V_a) 0.6 m/s (118 fpm)	**Air Speed (V_a) 0.9 m/s (177 fpm)**	**Air Speed (V_a) 1.2 m/s (236 fpm)**
1.2°C (2.2°F)	1.2°C (2.2°F)	2.2°C (4.0°F)

(end of quote from Std 55-2013)

6.1.4 Conclusion

In addition to the activities above, CBE was selected to be on a team of contractors developing the first *User's Manual* to ASHRAE Standard 55. This document explains the standard and provides guidance to designers on how to best use it in design. It includes many illustrated practical examples. CBE has several staff providing technical input to the *Manual*. PI Edward Arens has authored the sections that are most pertinent to air movement comfort, both for design of new spaces and for evaluation of existing buildings. This is being covered in detail since air movement design is not familiar to most of today's engineers and architects. Some of the *Manual's* practical content comes directly from work done under this CARB project.

7 Conclusions

The 11 projects address a wide range of topics from fundamental human subject studies to new product design. They also focus on the removal of barriers to fan implementation in design practice, and comfort and energy standards.

The human-subject *laboratory tests* examined fan-related comfort at a fundamental level. Several types of fan configurations were tested. The projects establish air temperature and humidity comfort thresholds for spaces with fans, obtain subjects' preferred air speed at different ambient conditions, test for whether air movement causes dry-eye discomfort. They also examine whether the comfort model for elevated air movement in ASHRAE Standard 55 is applicable to realistic environments with fans. The model assumes that the entire body is exposed to uniform low-turbulence air movement, while actual air movement is only on parts of the body, and with varying levels of turbulence. The numerous results of this project are all promising for the advancement of air movement cooling as an important energy efficiency strategy. They show that comfort can be provided by fans drawing 2 - 8 watts per person, up to 30C (86F) and 60% RH (relative humidity) ambient conditions.

Real buildings and occupancies contain features that cannot be tested in laboratory studies. The *field studies* investigated human responses to fans under long-term exposures. The results show that occupants like building with fans, and fan provides comfort under warm environments. Most people (75% in DPR and 70% in UW) felt that thermal comfort in their buildings enhanced their ability to get their job done. However, the Oakland school study showed that when using fans with automatic controls in designs, occupants should be trained how to interact with the controls.

The team explored ways to integrate fans into ceilings. We incorporated oscillating floor fans into the ceiling. We also proposed a design solution that eliminates the heat transfer reduction from radiant ceilings caused by suspended acoustical panels. Acoustical panels have been a major barrier for radiant ceilings. This design solution increases total heat transfer to 150% of the original heat transfer without acoustical panels. The team also designed two versions of an occupant-tracking fan, which have resulted in a provisional patent. The occupant-tracking fan overcomes the disadvantage of ceiling fans which in fixed position only project air movement to a limited floor area. It actively recognizes and tracks the occupants and permits continuous airspeed at the occupant's location, even if they move about within their workplace. This new idea will encourage various types of occupant-tracking fans in the future.

Fans providing comfort in warm environments save HVAC energy in air-conditioned building by allowing the setpoints to be controlled over wider ranges of temperature. EnergyPlus simulations show that each 1F increase in cooling setpoints corresponds to about 5% of total HVAC air-conditioning energy use.

To confidently use fans in buildings, designers need better information from fan manufacturers to select them and space them. Current fan specifications do not provide much design guidance, and are not based on an appropriate evaluation index. Our laboratory studies suggest that there may be simplified approaches to developing a new fan evaluation index, based on airspeed and comfort rather than volumetric flow. The development of a fan performance test method and index has been approved by ASHRAE TC 5.1 (Fans). We will assist this committee and use the findings from this study to help formulate a good test method and index.

The team also did a great deal of work under this contract to implement our work in the new code-compliant ANSI/ASHRAE Standard 55-2013, published in December 2013. The air movement sections were improved over 55-2010, and are one of the major contributions to the new standard.

Members of the team were also selected to prepare the new *User's Guide* to Standard 55. This guide will provide detailed design information for the use of air movement for cooling occupants in buildings.

8 Future work

- We may conduct lab studies under very dry environmental conditions to explore fan applications in hot-dry environments.
- As more fan-enabled buildings become available, conduct more field studies to understand as much as possible the issues with fan use in real buildings. Publicizing their energy use will be very useful for gaining the building industry's attention. Identify if there are age impacts on fan acceptability by occupants.
- Finish the occupant-tracking fan patent application and arrange for the manufacture of this type of fan system.
- Continuously work with fan manufacturers and designers to refine the fan evaluation index.
- Work with ASHRAE Standard 55 and the *User's Guide* project to encourage effective uses of fans to provide comfort efficiently in buildings.

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Appendix A Right-now survey questionnaire used in the Alameda office

1. TEMPERATURE

Right now, how acceptable is the temperature at your workplace?

Very acceptable ○ ◉ ○ ○ ○ ○ ○ Not at all acceptable

You feel (*Please mark on the scale*)?

Hot	Warm	Slightly warm	Neutral	Slightly cool	Cool	Cold

You would prefer to be:

Cooler	No change	Warmer
○	○	◉

2. AIR MOVEMENT

Right now, how acceptable is the air movement at your workplace?

Very acceptable ○ ○ ◉ ○ ○ ○ ○ Not at all acceptable

You would prefer:

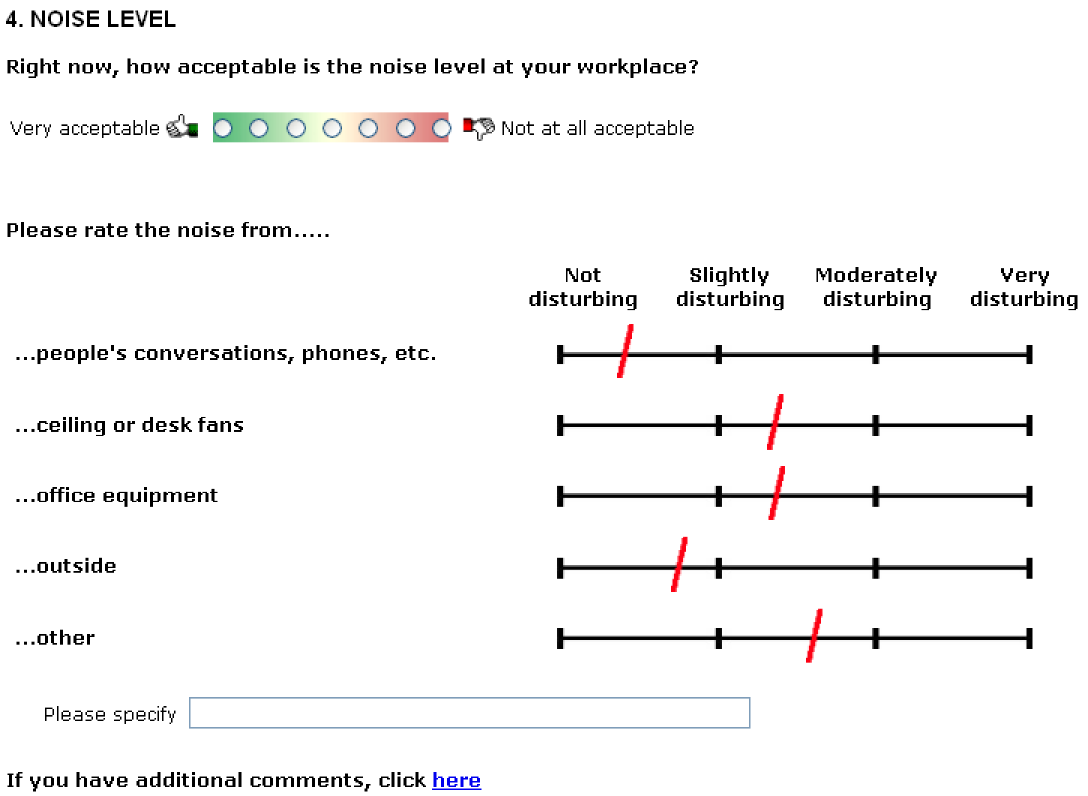
More air movement	No change	Less air movement
○	◉	○

3. AIR QUALITY

Right now, how acceptable is the air quality at your workplace?

Very acceptable ◉ ○ ○ ○ ○ ○ ○ Not at all acceptable

Continue >>



4. NOISE LEVEL

Right now, how acceptable is the noise level at your workplace?

Very acceptable ○ ○ ○ ○ ○ ○ ○ Not at all acceptable

Please rate the noise from.....

	Not disturbing	Slightly disturbing	Moderately disturbing	Very disturbing
...people's conversations, phones, etc.				
...ceiling or desk fans				
...office equipment				
...outside				
...other				

Please specify

If you have additional comments, click here