VOLUME 15, NUMBER 6 HVAC&R RESEARCH NOVEMBER 2009

Introduction of a Cooling-Fan Efficiency Index

Stefano Schiavon, PhD
Student Member ASHRAE

Arsen Krikor Melikov, PhD
Fellow ASHRAE

Received November 22, 2008; accepted July 24, 2009

In a warm environment, air movement with elevated velocity is a well-known cooling strategy. The local air movement is typically generated by cooling fans (e.g., ceiling fan, table fans, etc.). Appearance, power input, and price are the main parameters considered today when purchasing cooling fans, while cooling capacity and efficiency of energy use are unknown.

To address this knowledge gap, this paper introduces the cooling-fan efficiency (CFE) index, defined as the ratio between the cooling effect (measured with a thermal manikin) generated by the device and its power consumption. The index was determined for a ceiling fan, a desk fan, a standing fan, and a tower fan in a real office at three room air temperatures and at different fan speed levels. The results reveal that the index is sensitive enough to identify differences in the performance of the cooling devices. A standard method for testing fan cooling effect and an index for determining fan efficiency, such as the CFE index proposed in this study, need to be developed.

The cooling fans generate a nonuniform velocity field around occupants, which cannot be described with a single air-velocity value. Therefore, it is not clear how to apply in practice the recommended elevated velocities in warm environments presented in the present standards. The standards need to be revised.

INTRODUCTION

In a warm environment, elevated air movement is a widely used strategy for cooling occupants. The air movement increase can be produced by several devices, such as cooling fans (ceiling, floor standing, tower, and table fans); furniture-installed personalized ventilation; and body-attached ventilation devices with, under certain conditions, operable windows. The underfloor air distribution system, which is one of the total volume ventilation principles used in practice, also allows for increase or decrease of the velocity close to workplaces. The cooling capacity of cooling fans is limited, because they operate under isothermal conditions (i.e., the cooling of the body is a result of increased velocity only). The use of cooling fans in practice is easy and does not require special installations. The personalized ventilation systems (Melikov 2004) and the task-ambient conditioning systems (Arens et al. 1991; Bauman et al. 1998) perform better with regard to thermal comfort, since they may operate under nonisothermal conditions (i.e., the supplied air can be cooled below the room air temperature in addition to elevated velocity). Appearance, power consumption, and price are the main parameters considered when purchasing cooling fans, while cooling capacity and efficiency of energy use are unknown. Other factors, such as ergonomics, control options, etc. are also important. Comparison of the

Stefano Schiavon is a postdoctoral scholar at the Center for the Built Environment, University of California, Berkeley. This paper was written when he was a doctoral candidate at the Department of Applied Physics, University of Padova, Padova, Italy, and a guest PhD at the International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, Lyngby, Denmark. **Arsen Krikor Melikov** is an associate professor at the International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, Lyngby, Denmark.

performance of cooling fans from the point of view of cooling capacity and energy consumption is important for their application in practice.

According to the international standards on thermal comfort (ASHRAE 2004; ISO 2005; CEN 2007) elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. This can be better achieved by providing occupants with the opportunity to individually control locally applied air movement (i.e., air speed). A relationship between the air speed and the upper operative temperature limits is included in the present standards in graphical form. The body surface area exposed to the air movement and uniformity of the velocity field is also important for the heat exchange between the body and the environment. This, however, is not discussed in the standards.

It has been suggested (Sekhar 1995; Olesen and Brager 2004; Aynsley 2005) that a set high room temperature and cooling of the body by elevated air movement lead to substantial energy savings. Schiavon and Melikov (2008), by means of energy simulations, found that the required power input of the fan is to energy savings. The results obtained for the boundary conditions of their study reveal that traditional cooling devices, such as ceiling, standing, tower, and desk fans, may consume more electrical energy than is saved by not using a traditional HVAC system. Thus, knowledge as to how efficiently fans of different types use electrical energy to cool occupants is needed to justify the strategy of elevating the room temperature at increased air movement.

In this paper, a cooling-fan efficiency (CFE) index is introduced that relates the cooling effect of fans generating local air movement in the vicinity of occupants to their energy consumption. Experiments with different cooling fans are performed to validate the usefulness of the index.

COOLING-FAN EFFICIENCY INDEX

The efficiency is the ratio of the output to the input. It can be improved by reducing input and/or improving output. In the case of fans, which are used to cool people in warm environments by increasing the air velocity around the human body, the input is the electrical energy needed to run the fan (the power requirement of a fan is almost constant, and it can be used instead of energy to make the input variable time independent), and the output is the body cooling effect.

The body cooling effect produced by a fan depends on generated air velocity and turbulence field, body area exposed to moving air, body posture, air and mean radiant temperature, air humidity, clothing insulation, metabolic rate, humidity, and skin wettedness. Sophisticated thermal manikins with full body size and a complex shape were developed and used to determine the dry-heat loss from the human body under different environmental conditions (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002). A manikin's body is typically divided into several segments. They can be operated to maintain constant heat flux from the body, constant body surface temperature, or to have surface temperature equal to the skin temperature of an average person in a state of thermal comfort under the particular environmental condition of the exposure. Thermal manikins can be used to measure the fan cooling effect and, thus, to determine the CFE index. Thermal manikins that can measure dry-heat loss from the human body are commonly used today, though sweating thermal manikins are under development as well (Psikuta et al. 2008). Therefore, at this stage, dry-heat loss from the human body can be used to determine the CFE. In the future, more precise or effective ways of measuring the cooling effect may be developed and used instead of thermal manikins. Clothing thermal insulation and metabolic rate (personal factors that may vary substantially in real life) can be assumed to be constant, while air humidity and skin wettedness are not taken into account. The equivalent temperature (t_{eq}) is a well-known parameter that can be used to determine the CFE index. The equivalent temperature (formerly *equivalent homogenous temperature*) is defined as "The uniform temperature of the imaginary enclosure with air velocity equal to zero in which a person will exchange the same dry heat by radiation and convection as in the actual non-uniform environment" (SAE 1993; ISO 2004). In the definition, it is assumed that the body posture, the activity level, and the clothing design and thermal insulation are the same in both environments. The equivalent temperature is a pure physical quantity that integrates the independent effects of convection and radiation on human body heat loss in a physically sound way. The equivalent temperature t_{eq} does not take into account human perception and sensation or other subjective aspects, but may correlate with them. It is important to notice that t_{eq} is not a temperature that can be measured by a thermometer and that t_{eq} cannot be translated to an air temperature in a complex climate (Bohm et al. 1999). The body cooling effect achieved by air movement can be quantified by the change in whole-body manikin-based equivalent temperature t_{eq} from the reference condition t_{eq}* (similar indoor environmental conditions but without air movement) (i.e., $\Delta t_{eq} = t_{eq} - t_{eq}^*$). The concept of Δt_{eq} already has been used by several authors to quantify the whole-body cooling effect of air movement (Tanabe et al. 1994; Tsuzuki et al. 1999; Melikov et al. 2002; Watanabe et al. 2005; Sun et al. 2007). Thus, the CFE is defined by Equation 1.

$$CFE = \frac{Cooling\ effect}{Fan\ power} = (-1)\frac{\Delta t_{eq}}{P_f} \tag{1}$$

where

P_f = input power of the fan, defined according to CEN (2003)
Δt_{eq} = whole-body cooling effect

The measuring unit of CFE is °C/W (°F/W). Δt_{eq} usually would be negative (the equivalent temperature of the body cooled by a fan would be lower that the temperature without the fan). To ease interpretation of the index, the ratio between the cooling effect and the fan power is multiplied by –1 (Equation 1). The higher the CFE index is, the better the fan performance.

Figure 1 shows the CFE as a function of the fan power calculated at cooling effect Δt_{eq} of –0.5°C, –1°C, –2°C, –3°C, and –4°C (–0.9°F, –1.8°F, –3.6°F, and –7.2°F). It has been reported

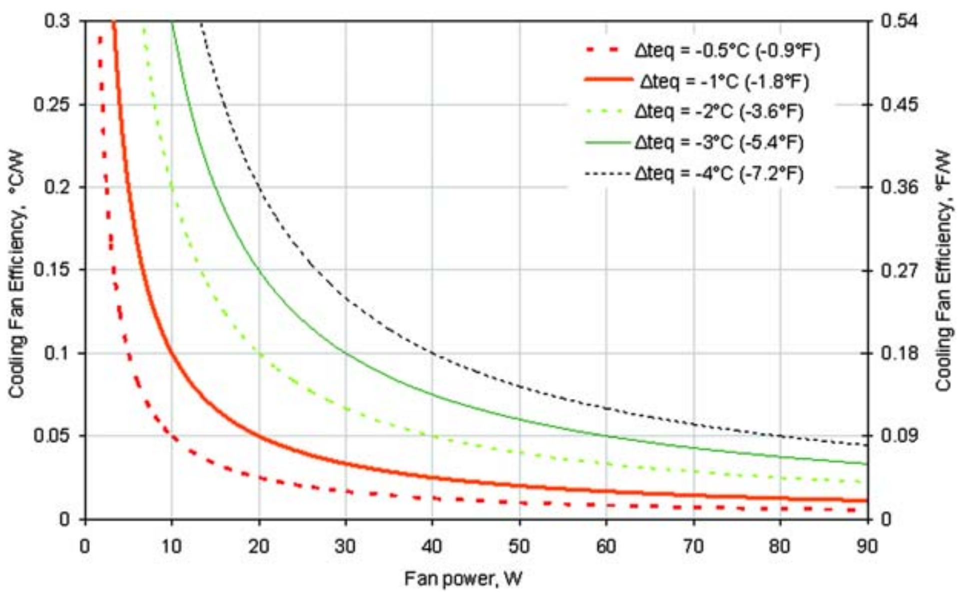

Figure 1. CFE versus fan power for five cooling effect levels.

that a cooling effect of –4°C (–7.2°F), obtained by local body cooling, can be acceptable for people (Watanabe et al. 2005, 2009). An Internet survey showed that the typical power consumption of cooling fans is lower than 90 W. The results in the figure show that at constant cooling effect the CFE increases with the decrease of the fan power (i.e., fans with different power may have the same cooling effect). The results also show that fans with the same fan power may have a different cooling effect due to differences in the generated flow (e.g., different target area, velocity, and turbulence field, etc).

Knowing the CFE and its cooling effect (Δt_{eq}) helps customers to purchase more efficient fans, fan designers/manufacturers to assess and develop better products, and policymakers to fix minimum values or classes of fan efficiency as is usually done with other electrical appliances (e.g., air-conditioner, refrigerators, boilers, etc.). HVAC designers may be helped in choosing the summer maximum allowed room temperature, depending on the cooling capacity of the fan, and to evaluate the possibility for energy savings based on the strategy of increased air movement at elevated room air temperature.

EVALUATION OF THE CFE INDEX OF COOLING FANS

The usefulness of the introduced CFE for comparison of cooling fans was demonstrated. Experiments were performed with four fans available on the market, including a ceiling fan (CF), a desk fan (DF), a standing fan (SF), and a tower fan (TF). The index of the cooling fans was determined and compared.

Method

Experimental Facilities. The fans used in this study and purchased for the purpose of these experiments are described in Table 1. The rotation speeds of the fans (velocity of the generated flow is expected to increase with the rotation speed) are defined by the manufacturers and can be varied in steps. The DF and SF have two speed levels, and the CF and TF have three speed levels. Experiments were performed in a real office room (5.8 × 4.42 × 3.5 m [19 × 14.5 × 11.5 ft]) with a suspended ceiling (0.5 m [1.6 ft] from the top). A double-pane strip window (5.80 m

Table 1. Main Characteristics of the Fans Used

Type	Speed Levels	Dimension, m (in.)	Power,[a] W
Ceiling fan (CF)— Three-blade axial fan	3	$\phi_{CF}{}^{b}$ = 1.20 (47); $d_{CF}{}^{b}$ = 0.5 (20)	65
Desk fan (DF)— Three-blade axial fan	2	$\phi_{DF}{}^{c}$ = 0.22 (9); $h_{DF}{}^{c}$ = 0.22 (9)	30
Standing fan (SF)— Three-blade axial fan	2	$\phi_{SF}{}^{d}$ = 0.39 (15); $h_{SF}{}^{d}$ = 1.10 (43)	50
Tower Fan (TF)— Centrifugal fan	3	$\phi_{TG}{}^{e}$ = 0.15 (6); $h_{TF}{}^{e}$ = 0.45 (18); $w_{TF}{}^{e}$= 0.08 (3); $d_{TF}{}^{e}$ = 0.46 (18)	50

[a]Nameplate fan power declared by the manufacturer
[b]ϕ_{CF} = external diameter of the blades of the CF; d_{CF} = distance between the blades and the ceiling.
[c]ϕ_{DF} = external diameter of the blades of the DF; h_{DF} = height over the floor of the rotation axis of the DF.
[d]ϕ_{SF} = external diameter of the blades of the SF; h_{SF} = height of the rotation axis of the SF.
[e]ϕ_{TF} = external diameter of the blades of the TF; h_{TF} = height of the inlet opening of the TF; w_{TF} = width of the inlet opening of the TF; d_{TF} = diagonal of the opening of the TF.

[19.0 ft] width and 1.85 m [6.1 ft] height) is located in one of the walls. The lower edge of the window is 1.15 m (3.8 ft) above the floor. The window faces north-west. Solar radiation was shielded with internal blinds. During the experiments, the outdoor temperature was always lower than 22°C (71.6°F). The room temperature was controlled with an electrical heater managed by a proportional-integral-derivative controller. The room was not equipped with ventilation and air-conditioning systems. A workplace was arranged in the room (see Figure 2), and a desk was placed in the center of the office.

Measuring Instruments. A thermal manikin was used to simulate an occupant and to evaluate the cooling effect of the fans. The thermal manikin is 1.68 m (5.51 ft) tall and shaped as an average-size Scandinavian woman. The total area of the manikin is 1.48 m^2 (15.93 ft^2). The body of the manikin consists of 23 independently controlled segments (see Appendix A) manufactured as polystyrene shells wound with embedded nickel wire, which serves to heat the body parts and to monitor the skin temperature. Low-voltage power is pulsed to each segment at a rate needed to keep the surface temperature of the manikin equal to the skin temperature of an average person in a state of thermal comfort. The power consumption under steady-state conditions is then a measure of the convection, radiation and conduction heat losses (dry-heat loss). For each body segment, the segmental equivalent temperature $t_{eq,i}$ can be calculated using the following equation:

$$t_{eq,i} = t_{sk,i} - \frac{Q_{t,i}}{h_{cal,i}} \tag{2}$$

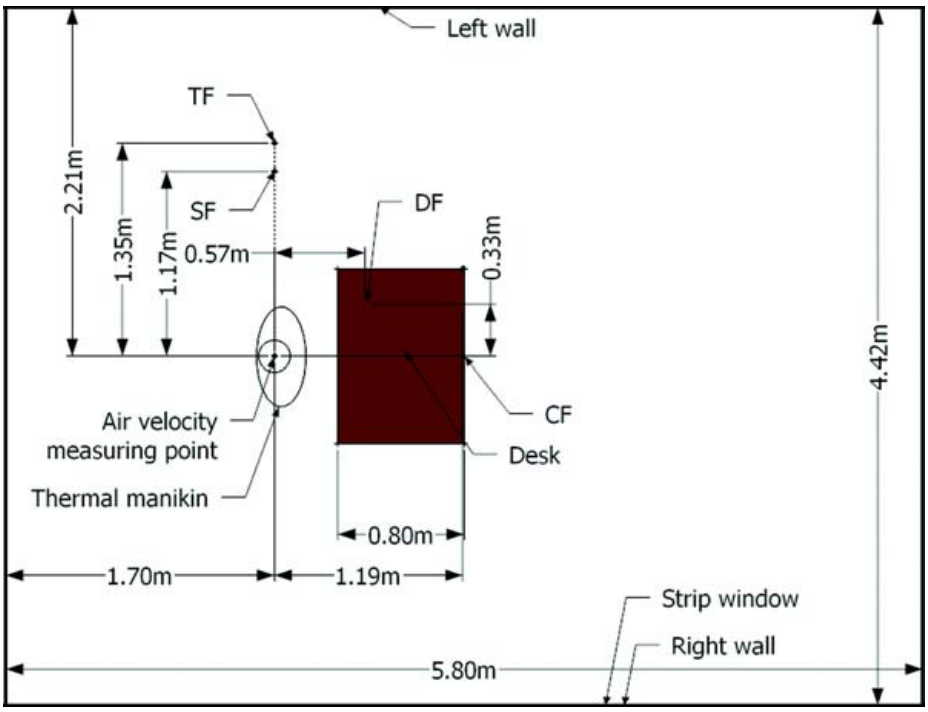

Figure 2. Room plan (5.8 × 4.42 × 3.5 m [19 × 14.5 × 11.5 ft]). Location of the office desk, thermal manikin, ceiling fan (CF), desk fan (DF), standing fan (SF), and tower fan (TF).

where *tsk,i* is surface temperature measured for the *i*-th segment, $Q_{t,i}$ is sensible heat loss (power consumption) of the *i*-th segment, and $h_{cal,i}$ is dry-heat transfer coefficient determined during calibration of the manikin in a standard environment. The determination of the t_{eq} is described in Appendix A. The t_{eq} for the whole body is obtained by computing the area-weighted average over all the body segments (see Appendix A).

A multichannel low-velocity thermal anemometer with omnidirectional velocity transducers was used to perform mean velocity, turbulence intensity, and air temperature measurements at several points in the room. The characteristics of the anemometer comply with the requirements for such instruments specified in the standards (ISO 1998; ASHRAE 2005). The room air temperature was measured also with a mercury thermometer. The relative humidity (RH) was monitored but not controlled. The resolution of the used hygrometer was 0.1% RH. The fan power input was measured with a powermeter.

Experimental Conditions. The performance of the four fans was studied at three room air temperature levels: t_a= 25°C, 27°C, and 30°C (77°F, 80.6°F, and 86°F). Two speed levels for the DF and SF and three speed levels for the CF and TF were explored. Measurements were also performed in a still environment without a fan. The experiments were run in a randomized sequence in order to protect the results against uncontrolled and/or unknown influences of variables that are not part of the experiments (Barrentine 1999). Throughout the experiments, the room temperature and RH vary within the ranges reported in Table 2.

The experimental setup and the location of the fans is shown in Figure 2. The CF was installed in the center of the room. The distance between the suspended ceiling and the blades was 0.25 m (0.82 ft), and between the blades and the floor it was 2.75 m (9.02 ft). The DF was located in front of the manikin on the table on the left side of the laptop at a distance of 0.66 m (26 in.) (three times its diameter) from the manikin. The SF was located on the left side of the manikin at a distance of 1.17 m (46 in.) (three times its diameter) distance. The TF was located on the left side of the manikin at a distance of 1.35 m (53 in.) (three times the diagonal of its opening). The thermal manikin was dressed with a long-sleeved shirt, thin long trousers, panties, ankle socks, and shoes. This typical summer office clothing was equal to 0.47 clo. The manikin was seated upright on an office chair (0.15 clo).

Experimental Procedure. The surface temperature $t_{sk,i}$ and the power consumption $Q_{t,i}$ were recorded for ten minutes after steady-state conditions were obtained (i.e., when the difference in the average surface temperature of the manikin during the last ten minutes was less than 0.05°C [0.09°F]). The fan power was manually recorded while logging the manikin data. The manikin was then moved from the desk, and the mean air velocity and the turbulence were measured at its location at four heights (0.2, 0.6, 1.1, and 1.7 m) (8, 24, 43, and 67 in.). Three-minute velocity measurements were performed as recommended in the indoor climate standards.

Measurements Uncertainty. The description of the uncertainties of the measured and derived quantities is reported in Appendix B.

Table 2. Room Air Temperature and Relative Humidity During the Experiment

Temperature Setpoint		Measured Room Air Temperature		Relative Humidity, %
°C	°F	°C	°F	
25	77	24.9–25.2	76.82–77.36	22.7–44
27	80.6	26.8–27.2	80.24–80.96	23.5–34.5
30	86	29.9–30.2	85.82–86.36	21.5–31.7

Statistical Analysis. T-tests were performed to determine whether the obtained results were statistically different. A paired sample t-test was applied when testing the same sample for different levels of treatment. A two-sample t-test was applied when different samples were compared (equality/inequality of variance was tested with F-test). The tests were performed using S-Plus (Insightful 2007). P-values less than 5% ($p < 0.05$) were considered to be statistically significant.

Results

The cooling effect Δt_{eq}, fan power P_f, and CFE were obtained for each of the four fans under the experimental conditions studied (see Table C1 in Appendix C). The results identify a large variation in the whole-body cooling effect (between –3.2°C and –0.4°C [–5.76°F and –0.72°F]), in the fan power (between 15.6 and 49.3 W), and in the CFE index (between 0.009°C/W and 0.177°C/W [0.016°F/W and 0.319°F/W]).

Cooling-Fan Efficiency (CFE) Index. The results obtained with the four fans at the room air temperatures and speed levels studied are compared in Figure 3. The DF has the highest CFE index (CFE varies between 0.095°C/W and 0.177°C/W [0.171°F/W and 0.319°F/W]) and the smallest power consumption (P_f varies between 16 and 20 W). The CFs, SFs, and TFs have similar results: CFE and P_f for the CF, SF, and TF varied in the ranges 0.018°C/W and 0.079°C/W (0.032°F/W and 0.142°F/W) and 37–48 W, 0.038°C/W and 0.058°C/W (0.068°F/W and 0.104°F/W) and 33–40 W, and 0.009°C/W and 0.066°C/W (0.016°F/W and 0.119°F/W) and 37–49 W, respectively. The results also indicate that the CFE of the DF is substantially more sensitive to the changes in the room air temperature and speed level than the CFE of the other three fans. The CFE of the SF is least affected by the change of the room air temperature and fan velocity.

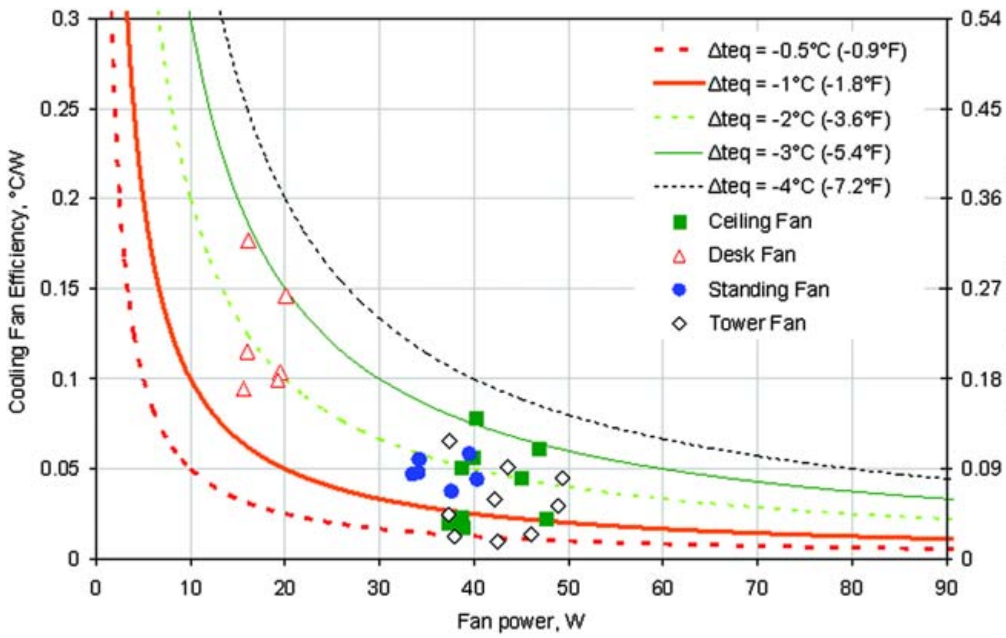

Figure 3. Fan power versus CFE index for the ceiling fan (CF), the desk fan (DF), the standing fan (SF), and the tower fan (TF). Lines with constant cooling effect (Δt_{eq}) are plotted.

The average of the CFE index obtained for different room air temperatures and speed levels with each of the fans was calculated. The results are compared in Figure 4. The sample standard uncertainty of the index is equal to ±0.009°C/W (±0.016°F/W) (see Appendix B). The DF is the most effective cooling device; its CFE (*CFE* = 0.123°C/W [0.221°F/W]) is more than double the index of the other fans (between *CFE* = 0.032°C/W [0.058°F/W] and *CFE* = 0.048°C/W [0.086°F/W]). The TF is the least-efficient cooling device. The significance in the differences in the average CFE for the four types of fan were statistically evaluated by a two-sample t-test assuming unequal variance. The result are plotted in Figure 4. The efficiency of the DF is significantly ($p < 0.01$) higher than the CFE of the remaining three fans. No significant difference in efficiency of these three fans was found (except that the CFE of the SF is higher than the CFE of the TF).

The influence of the room air temperature on the CFE was analyzed. The average of the CFE index obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 5. From a heat-transfer point of view, the room air temperature has an influence on the cooling effect and, thus, should have an influence on the CFE index. A paired sample t-test was used to identify whether the difference in the mean between each temperature level is significant. A paired test hypothesis is valid, because the same sample has been tested at different temperature levels. The results of the t-test are plotted in Figure 5. Significant differences in the average were found between all tested averaged CFEs. The results reported in Table C1 (see Appendix C) reveal that the room air temperature has no effect on the power consumption of the fan.

Cooling Effect. As expected, the cooling effect of the fans varied when the room air temperature and the speed changed (Table C1, Appendix C). For the tested conditions, the cooling effect of the CF varied between –3°C and –0.5°C (–5.4°F and –0.9°F), of the DF between –3°C

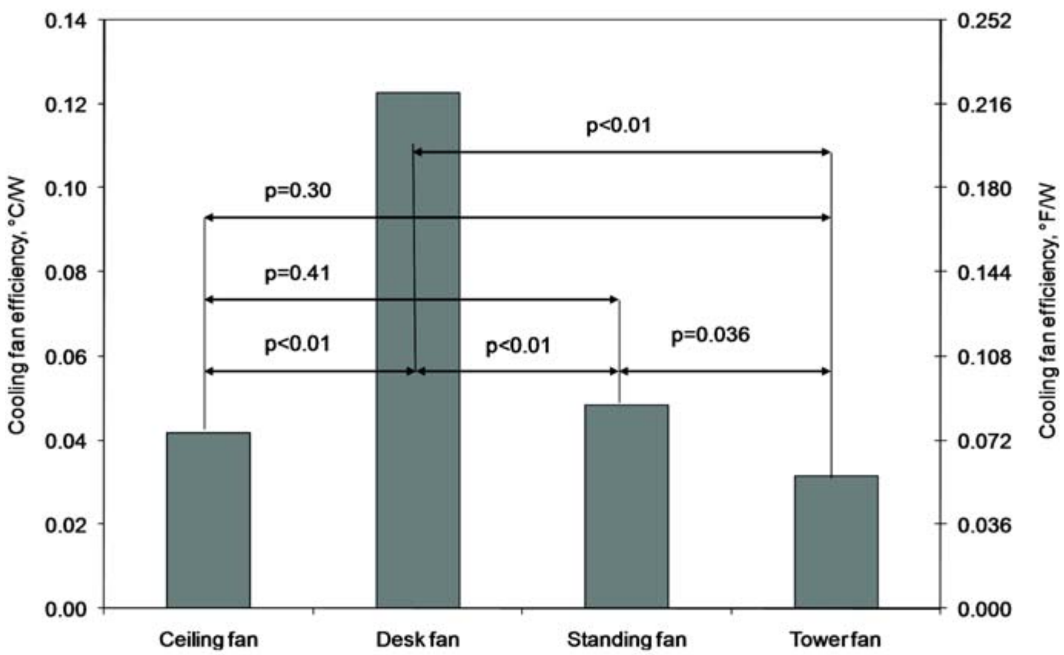

Figure 4. Averaged (over the speed levels and room air temperature) CFE index for the CF, DF, SF, and TF.

and –1.5°C (–5.4°F and –2.7°F), of the SF between –2.5°C and –1.5°C (–4.5°F and –2.7°F), and of the TF between –2.5°C and 0.5°C (–4.5°F and –0.9°F). The cooling effect of the SF was least affected by the change in the experimental conditions.

The influence of the room air temperature on the cooling effect was analyzed. The average of the cooling effect obtained with the four fans at the tested velocities was calculated for each of the room air temperatures. The results are compared in Figure 6. The significance of the difference in

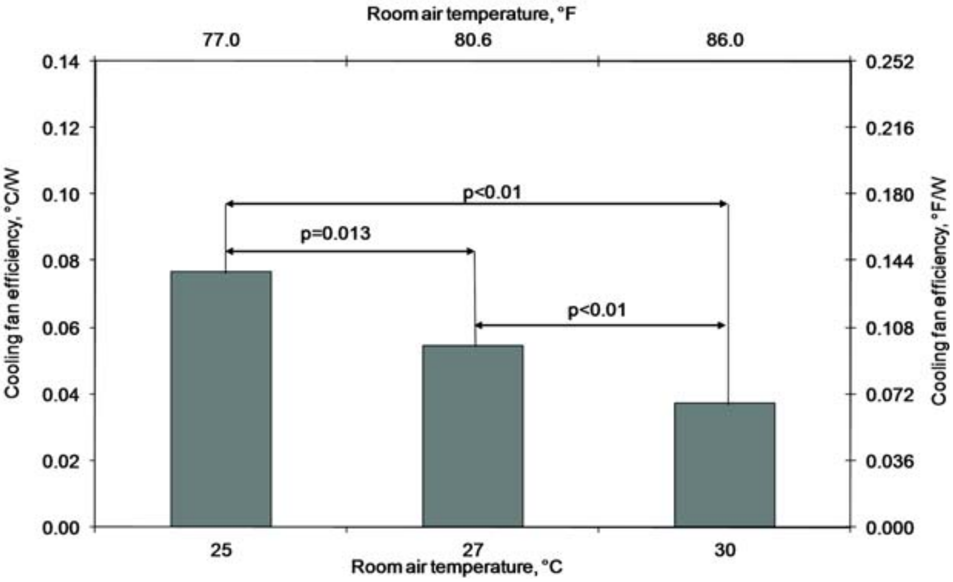

Figure 5. Averaged (over the type of cooling fan and speed level) cooling fan efficiency for three room air temperatures.

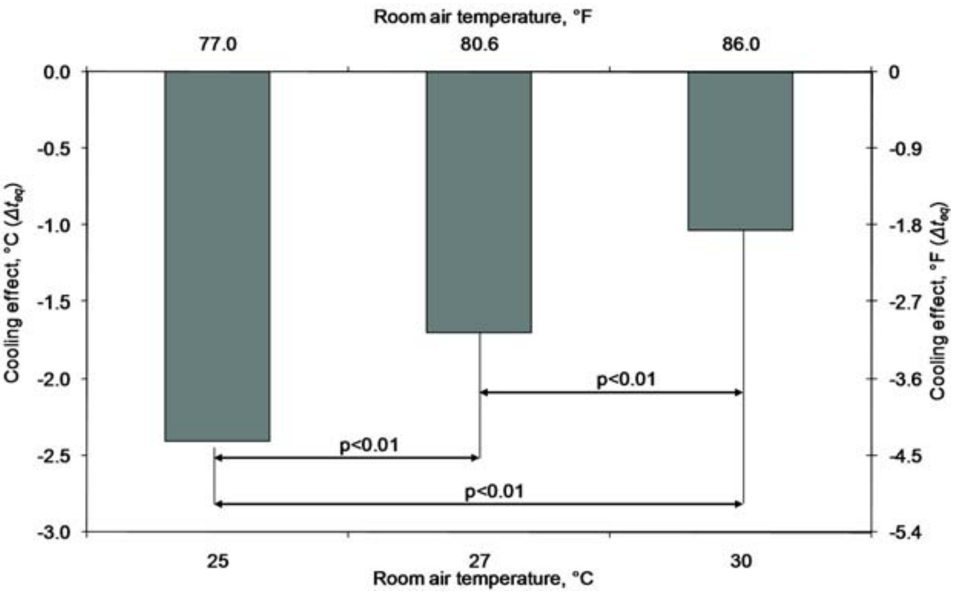

Figure 6. Averaged (over the type of cooling fan and speed level) cooling effect for three room air temperatures.

the cooling obtained at different temperature was analyzed by a paired sample t-test. As expected, the cooling effect decreased significantly ($p < 0.01$) with the increase of the room air temperature.

The whole-body cooling effect (Δt_{eq}), discussed previously and reported in Appendix C, is the weighted average of the cooling effect of each body segment. The cooling of the body segments depends on the local flow field generated by the fans. Analyses of the local cooling effect obtained by the tested fans for each body segment were performed. In the following only, the results obtained at a room air temperature of 25°C (77°F) are shown and discussed, because the conclusions were rather similar to the results obtained at 27°C (80.6°F) and 30°C (86°F). The local cooling effect of the four fans on each of the 22 body segments of the manikin is shown in Figure 7. The DF and SF had two speed levels, while the CF and TF had three levels. The cooling effect increases with the increase of the speed level. However, the exposure to the airflow has a much stronger effect. The body segments exposed directly to the flow are cooled much more than for those in shadow. The impact of the speed level on the cooling effect is greater for the exposed body segments than the segments in shadow. The cooling effect of the CF is quite symmetrical. The body segments that are exposed to the air movement generated by the fan (left and right front thigh, left and right face, back of the neck, right hand, left and right forearm, left and right chest) are cooler than the rest of the body. The air movement

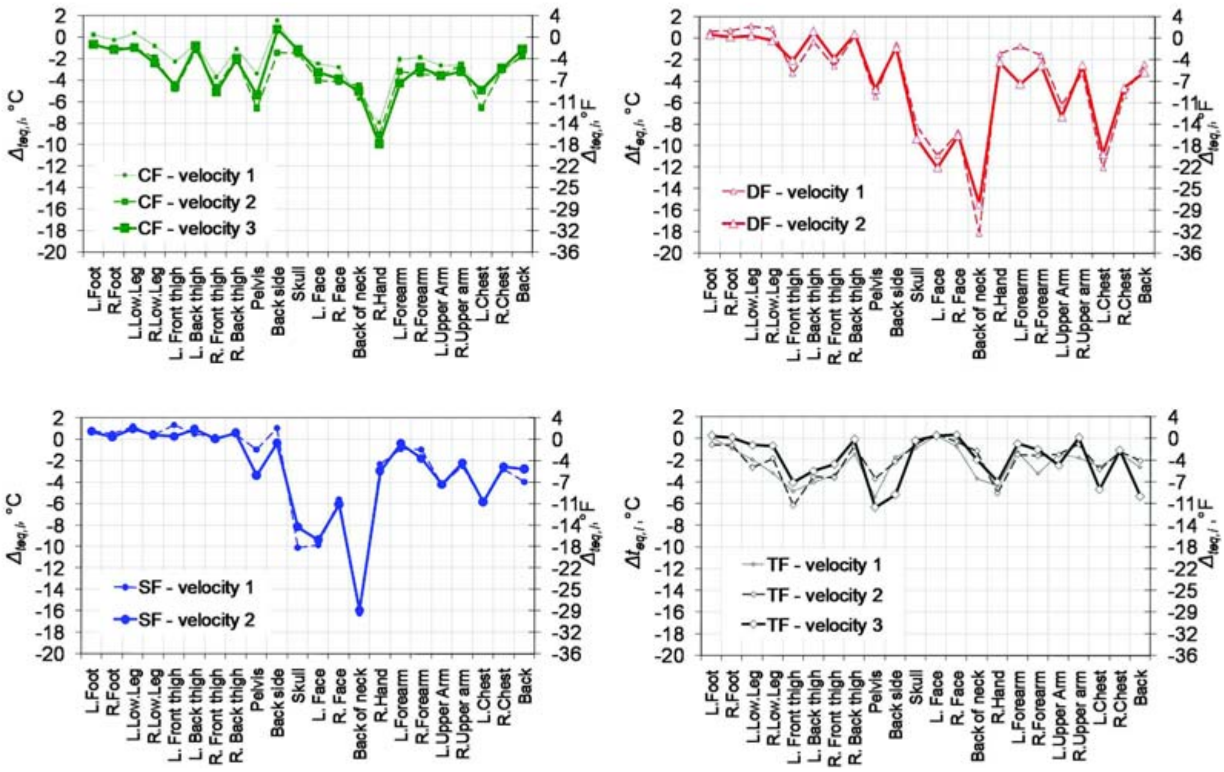

Figure 7. Change in manikin-based equivalent temperature ($\Delta t_{eq,i}$) on each body part from the reference condition (room temperature equal to 25°C and no devices used for air movement) for the ceiling fan (top left), desk fan (top right), standing fan (bottom left), and tower fan (bottom right). Step-change control of the fan velocity is possible. The $\Delta t_{eq,i}$ was calculated for the different speed levels reported in Table 1.

generated by the ceiling fan runs over the manikin from top-front. The desk fan provides a nonuniform cooling effect on the body. The airflow generated by the fan attacks the manikin's body from the left (the fan is located only 0.66 m [26 in.] from the manikin). The coolest segments are those exposed to the flow generated by the fan (i.e., skull, left and right face, back of neck, left chest, left upper arm, and left forearm. The local cooling provided by the SF has a pattern rather similar to the cooling provided by the DF. Similarly, the SF generates a nonuniform cooling effect. The rotation axis of the SF is located at 1.1 m (43 in.) above the floor (i.e., the highest velocities are generated at the manikin's head level). Therefore, the coolest parts are the left and right face, the skull, and the back of the neck. The lower segments of the manikin are slightly warmer with the SF in operation than in the reference case without fan (up to 1°C [1.8°F] warmer). The uncertainty in determining the cooling effect (estimated to be 0.3°C [0.54°F]; see Appendix B) alone cannot explain the difference. Complex airflow interaction in the vicinity of the body may be the reason. This needs to be further studied. The tower fan generated a uniform cooling effect. The coolest parts are those on the lower left (i.e., the left front thigh, the pelvis, and the lower back). The cooling effect of the head is almost negligible.

The flow field generated by the fans was nonuniform and, therefore, caused nonuniform local cooling of the manikin's body. The asymmetric cooling of the body areas was investigated further. The average cooling effect for the upper body segments (right hand [left hand was broken], forearm [right and left], upper arm [right and left], chest [right and left], and back) and for the head (skull, face [right and left] and back of neck) was determined. The total area of the upper body segments was 0.68 m^2 (7.32 ft^2), of the head was 0.13 m^2 (1.40 ft^2), and of the whole-body was 1.48 m^2 (15.93 ft^2). The results for the upper body parts shown compared in Figure 8 were obtained by averaging over the same segments (i.e., the same surface area). Averaging over different number of segments (i.e., different surface area will lead to different cooling effect).

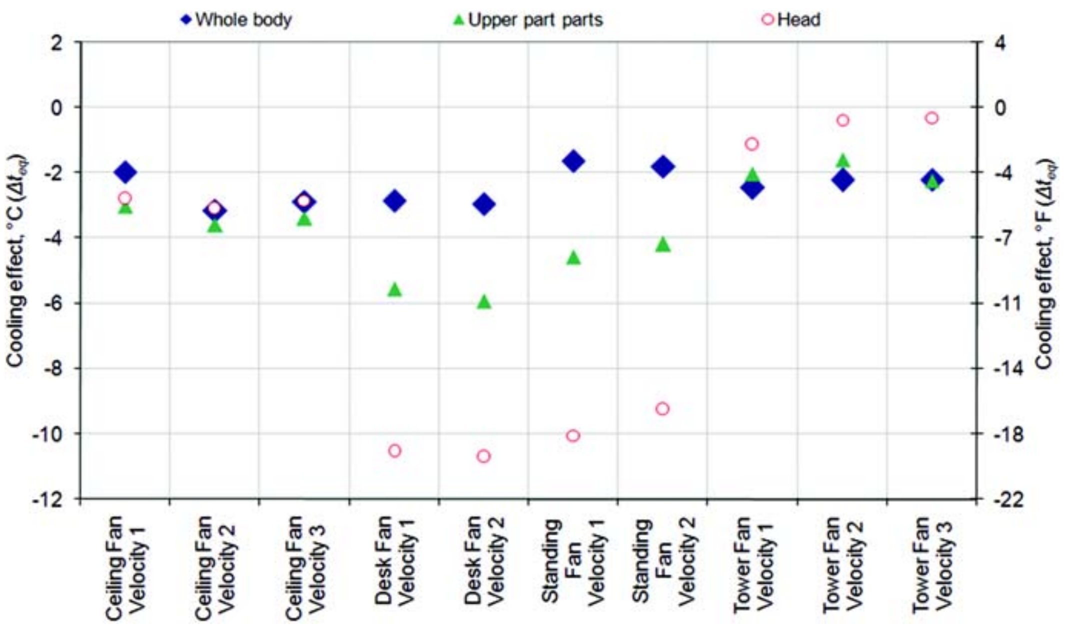

Figure 8. Cooling effect for the whole body (22 body segments), the upper body part (12 body segments), and the head (4 body segments) for the CF, DF, SF, and TF when the room temperature was set to 25°C.

The cooling effect of the ceiling fan was the most uniform. The difference in the whole-body cooling effect for the four types of fans is less than 2°C (3.6°F). The cooling effect of the upper body parts is always higher than the cooling effect of the head and the whole body. The DF and SF generate the largest nonuniformity in the local cooling effect. The head and the upper body parts are substantially cooler than the whole body. The head is much cooler than the reference condition (11°C [19.8°F] for the DF, and between 9°C [16.2°F] and 10°C [18°F] for the SF), and it is cooler than the whole body (8°C [14.4°F] for the DF and between 7.5°C [13.5°F] and 8.5°C [15.3°F] for the SF). The TF causes a quite uniform but weak cooling of the body. The whole body and the upper parts are cooler than the head (between 1°C and 2°C [1.8°F and 3.6°F] cooler). The speed level does not affect significantly the whole-body cooling effect except for the ceiling fan. The impact of the speed level on the cooling of the upper body part and the head is also smaller in comparison with the effect of exposure to the flow.

In Figure 9, the whole-body cooling effect determined is plotted versus the fan power measured. The relative uncertainties are shown. The whole-body cooling effect of the desk fan and the ceiling fan (speed level 2 and 3) is almost the same (around –2.5°C [–4.5°F]). However, the desk fan needs less than half of the electrical power used by the ceiling fan (around 20 W compared to 40 W). The DF and CF have a higher cooling effect than the TF and the SF. The SF has the lowest cooling effect, lower than –2°C (–3.6°F), with a fan power that varies in the range 35–40 W. For the TF, an increase of the speed level implies a slight reduction of the cooling effect with an increase of the needed power. Increase in speed level always implies an increase in power requirement, but this does not always cause a higher cooling effect. In fact, except for the CF, the increase of the cooling effect due to increase of the rotation speed is always lower than the measurement uncertainty. The cooling effect of the CF increased dramatically when the lowest speed level was changed to the second speed level. In this case, the power consumption was

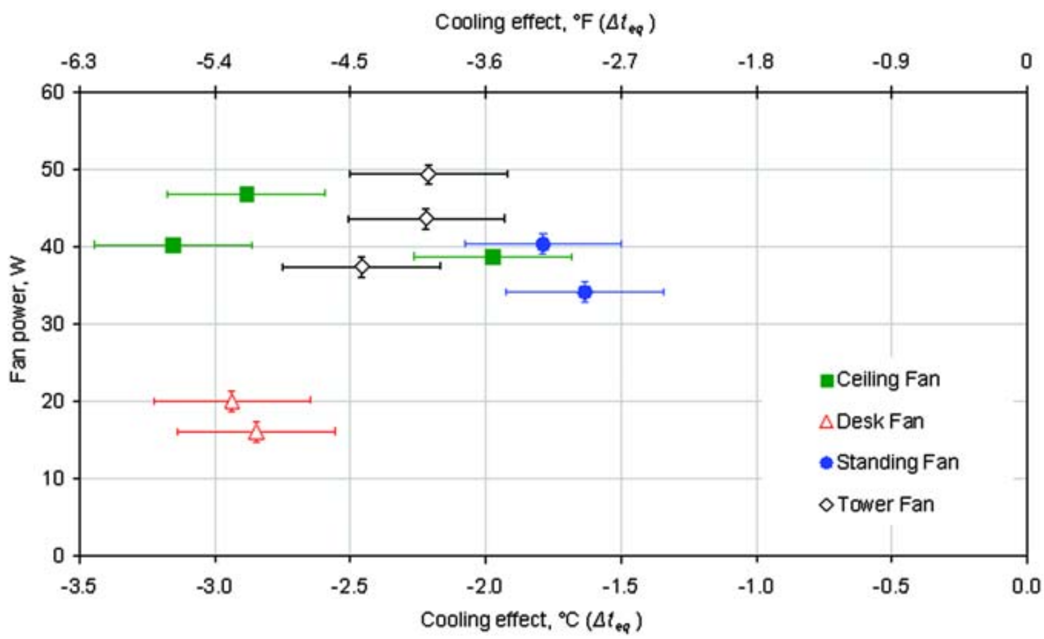

Figure 9. Cooling effect versus fan power for the CF, DF, SF, and TF for the tested speed levels when the room temperature was set to 25°C.

not very different. The reason may be due to the fluid dynamics characteristics of the fan (i.e., to the relation between the rotation speed and the generated flow). In Figure 9, only the results obtained when the room temperature was equal to 25°C (77°C) are reported. The same substantial increase in the cooling effect caused by the ceiling fan when the speed level was changed from level one to level two was also recorded when the room temperature was equal to 27°C and 30°C (80.6°F and 86°F). From the results shown in Figure 9, it can be concluded that, generally, changing the speed level is not an effective way of controlling the cooling effect.

The air velocities measured at 0.2, 0.6, 1.1, and 1.7 m (8, 24, 43, and 67 in.) height above the floor with the CF and the TF are shown in Figure 10a, and with the DF and the SF in Figure 10b. The air velocity was measured with a low velocity thermal anemometer described in the "Measuring Instruments" section above; the accuracy of the instrument is reported in Appendix B. The importance of reporting the air velocities is discussed later in the paper. The air velocity field generated by the four fans is different. The CF generates downward airflow from the ceiling to the floor. The highest velocity (2.2 m/s [433 fpm]) is measured at the floor level. Therefore, it may be expected that the generated flow will cool mostly the lower part of the manikin (legs and feet). This, however, is not seen from the results of the segmental cooling effect, because the air velocity was measured while the manikin was moved away from the desk. Furthermore, the manikin was seated in front of the desk with its legs under the table far from the location of the velocity measurement. The blocking effect of the manikin's body and the interaction between the fan flow and the thermal plume generated by the thermal manikin may have had an impact on the cooling of the body segments. The TF also causes air movement mainly in the lower part of the room. The highest velocity (3.2 m/s [630 fpm]) was measured at the flow level.

The DF and the SFs generated similar air velocity profiles. In both cases, the maximum air velocity (2.4 m/s [472 fpm] for the DF and 1.8 m/s [354 fpm] for the SF) was recorded at 1.1 m (43 in.) above the floor (i.e., the height of the manikin's head). The high velocity at the head level caused the strong nonuniform cooling of the body segment (Figure 8) as previously discussed.

DISCUSSION

Elevated air speed is widely used to provide comfort for occupants in warm environments. Cooling fans (e.g., ceiling fans, desk fans, etc.) are used to generate air movement. It is accepted

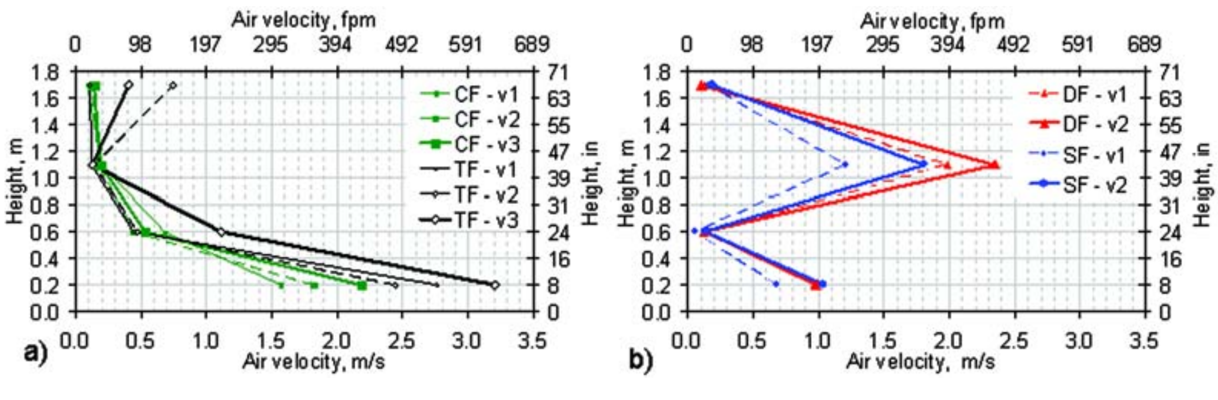

Figure 10. Air velocity measured at 0.2, 0.6, 1.1, and 1.7 m (8, 24, 43, and 67 in.) height above the floor where the manikin was located during the experiments (a) for the CF and TF and (b) for the DF and SF when the room temperature was set to 25°C.

that energy savings can be achieved with this strategy as opposed to air-conditioning the whole building. Due to different design, installation, and use, the performance of cooling fans with regard to their cooling effect can differ greatly. As the results of the present study show, at the same cooling effect, the power consumption of different fans can be nonuniform. The CFE index introduced in the present study makes it possible for the first time to evaluate and compare cooling fans. This index, in a single value, combines fan performance with regard to cooling effect and energy use. The experiments performed with four cooling fans of different designs available on the market (i.e., ceiling [CF], desk [DF], tower [TF], and floor standing fans [SF]) document that the CFE index is sensitive to identifying differences in the performance of the cooling devices. The body cooling effects caused by the fans differed. The CF and DF had a rather similar cooling effect, which was substantially higher than the cooling effect of the SF and TF. However, the electrical power used by the DF was twice as low as that used by the CF; the DF, therefore, had a significantly higher CFE index than the remaining three fans. The index can be used by HVAC engineers and policymakers, as well as for classifying fans according to their performance.

As already discussed, elevated air speed under individual control is recommended in the present indoor climate standards (ASHRAE 2004; ISO 2005; CEN 2007) for providing occupants with thermal comfort in warm environments. A relationship between the air speed and the upper operative temperature limits is provided in the standards. The recommended speed increase depends not only on the air temperature but also on the difference between mean radiant temperature and air temperature (t_a). When the mean radiant temperature is lower than the air temperature, elevated air speed is less effective for increasing the heat loss from the body. Conversely, elevated air velocity is more effective for increasing the body heat loss when the mean radiant temperature is higher than the air temperature. The relationship included in the standards is based on a theoretical calculation of the body cooling when it is exposed to uniform airflow. The relationship has been verified in human subject experiments performed under laboratory conditions when the air temperature is equal to the mean radiant temperature (Toftum et al. 2003). However, the validity of the relationship is not easily applicable when cooling fans are applied, because, as the present results reveal, body cooling by such fans is nonuniform due to large nonuniformity in the generated velocity field. The velocity field and its direction cannot be described with a single value. Therefore, it is not clear how to apply in practice the recommendations in the standards. Other methods for quantification of the cooling effect of air movement have been suggested as well (Szokolay 1998; Aynsley 2007). Aynsley proposed using the SET* index (Gagge et al. 1971), since it includes the impact of humidity and the thermal insulation of clothing, which are not considered in the relationship for elevated velocity included in the present standards. This method is included in Addendum f of ASHRAE Standard 55 (2004). This approach has the same limitation, namely that there is no unique velocity that can describe the complex air velocity field generated by cooling fans. This is demonstrated with the following example based on the data collected in the present study. The air velocity values of the four tested fans, measured at mean radiant and air temperatures equal to 27°C (80.6°F) and the lowest fan speed level, were used to calculate the SET* index. The measured RH was equal to 26%, the clothing thermal insulation of the manikin was 0.62 clo (including the thermal insulation of the chair) and the activity level was 1.1 met. The results of the calculations are listed in Table 3 and plotted in Figure 11. The SET* calculations were performed with ASHRAE Thermal Comfort Tool (Fountain and Huizenga 1997). From the results listed in Table 3, it can be seen that SET* depends on the used air velocity. This is an important point, because the new addendum of ASHRAE Standard 55 does not specify which velocity should be used. In Figure 11, the SET* values calculated with the maximum velocity, the average velocity (average of velocity measured at 0.2, 0.6, and 1.1 m [8, 24, 43 in.] above the floor), and the velocity measured at 0.6 m

Table 3. SET* Calculated when the Room Temperature Was 27°C (80.6°F)

Measuring Height		Air Velocity		Type of Fan	SET*		Cooling Effect Calculated with SET*		Cooling Effect Measured Thermal Manikin	
m	in.	m/s	fpm		°C	°F	°C	°F	°C	°F
0.2	8	1.35	266	Ceiling fan	23.4	74.1	–3.1	–5.6	–0.9	–1.6
0.6	24	0.32	63		25.5	77.9	–1	–1.8		
1.1	43	0.14	28		26.5	79.7	0	0.0		
1.7	67	0.13	26		26.5	79.7	0	0.0		
Avg		0.60	119		24.6	76.3	–1.9	–3.4		
0.2	8	0.74	146	Desk fan	24.3	75.7	–2.2	–4.0	–1.8	–3.2
0.6	24	0.10	20		26.5	79.7	0	0.0		
1.1	43	1.76	346		23	73.4	–3.5	–6.3		
1.7	67	0.11	22		26.5	79.7	0	0.0		
Avg		0.87	171		24	75.2	–2.5	–4.5		
0.2	8	1.27	250	Standing fan	23.5	74.3	–3	–5.4	–1.9	–3.4
0.6	24	0.18	35		26.4	79.5	–0.1	–0.2		
1.1	43	1.77	348		23	73.4	–3.5	–6.3		
1.7	67	0.12	24		26.5	79.7	0	0.0		
Avg		1.07	211		23.7	74.7	–2.8	–5.0		
0.2	8	3.27	644	Tower fan	22.4	72.3	–4.1	–7.4	–0.9	–1.6
0.6	24	0.77	152		24.2	75.6	–2.3	–4.1		
1.1	43	0.27	53		25.7	78.3	–0.8	–1.4		
1.7	67	0.12	24		26.5	79.7	0	0.0		
Avg		1.44	283		23.3	73.9	–3.2	–5.8		
		0.15		None	26.5	79.7				

[24 in.] and 1.1 m [43 in.] for each of the tested fans are plotted; the actual indoor climate standards recommend velocity measurements of 0.1, 0.6, and 1.1 m (4, 24, 43 in.) in order to predict thermal comfort of sedentary occupants. The results plotted in Figure 11 show substantial differences in the cooling effect calculated by SET* based on the above defined velocities. The measured cooling effect with the thermal manikin is also very different and is not correlated with any of the defined velocities used for calculation. For each of the tested cooling fans, the difference in the cooling effect defined with the SET* values calculated with the maximum velocity, average velocity, and velocity measured at 0.6 m (24 in.) and 1.1 m (43 in.) are comparable with the entire range of elevated room temperatures recommended in the present standards. In a nonuniform air velocity field, as it occurs in practice, the approach recommended in the present standards, as well as the SET*, cannot be applied. This issue needs to be carefully considered and addressed in the standards.

The DF was found to have the highest efficiency index of the four tested fans (Figures 3 and 4; Table C1, Appendix C). The whole-body cooling effect of this fan was largest. The nonuniformity of the local cooling effect of this fan was also greatest, with the head region being mostly

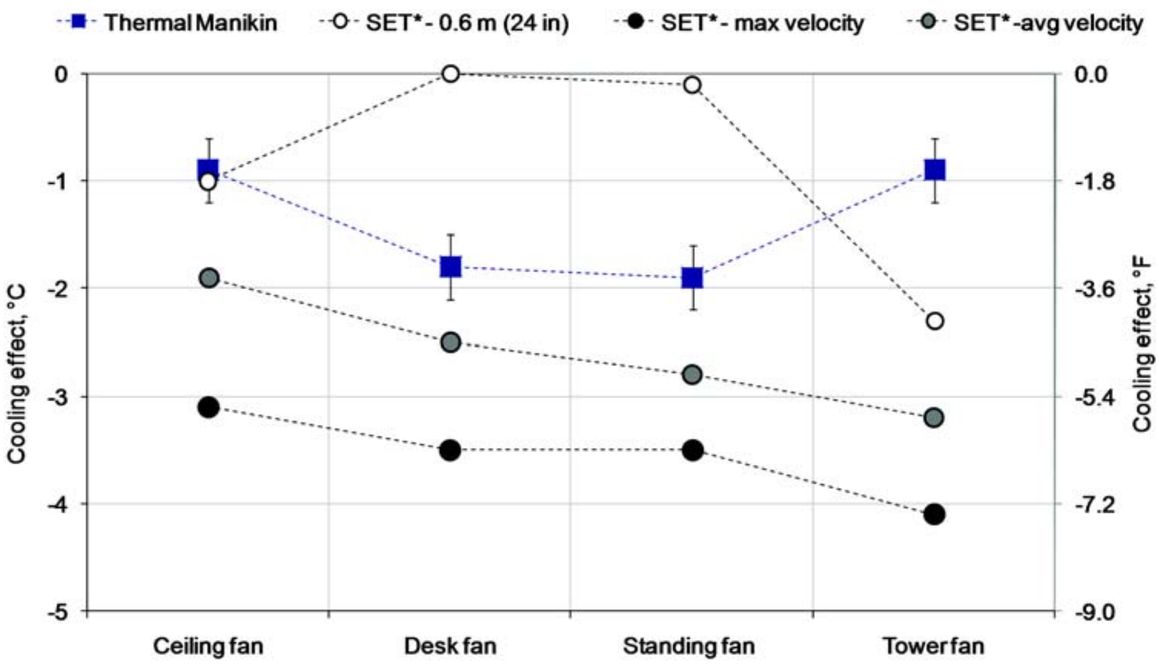

Figure 11. Comparison of the cooling effect measured with the thermal manikin and calculated with SET*. For each cooling fan, SET* is calculated based on generated maximum velocity, average velocity, and velocity measured at 0.6 m (24 in.) and 1.1 m (43 in.) above the floor. SET* calculated with the minimum velocity was always equal to 0. The accuracy for the measurements with the manikin is reported. The dotted line connecting the values obtained from the thermal manikin was added to increase the readability of the chart.

assessing the performance of cooling fans, because the head is an active heat dissipater, and in warm environments the whole-body thermal sensation follows the head region thermal sensation closely (Melikov et al. 1994a, 1994b; Arens et al. 2006; Watanabe et al. 2009). Thus, at the same efficiency, the performance of the fan that provides greater cooling of the head may be considered to be better. However, these selection criteria may fail to be correct in practice, because human response to airflow from the front and from the back is different (Mayer and Schwab 1988; Toftum et al. 1997).

In this study, the cooling effect of air movement was quantified by measuring the dry-heat loss. The evaporative heat loss was not taken into account, because the thermal manikin used cannot sweat. Several studies have used dry-heat loss measured by a thermal manikin to quantify the cooling effects of air movement on the human body. Tsuzuki et al. (1999) studied the performance of three designs of task ambient air-conditioning systems and found that the cooling effect of the combined evaporative and sensible cooling may double the total whole-body cooling rate due to dry-heat loss alone when 20% of the surface was wet. The cooling effect of the evaporative heat loss increases with the increase of the room temperature. In the future, the determination of fan efficiency can be made more accurately by sweating thermal manikins. The sweat glands are not uniformly distributed over the human body. Therefore, use of the thermal manikins available today, with simulated sweat glands on the surface areas corresponding to the site of the human skin where they are most dense, can be considered.

A considerable number of studies focused on the use of fans to cool people in a warm environment (McIntyre 1978; Rohles et al. 1983; Jones et al. 1986; Tanabe and Kimura 1987; Scheatzle et al. 1989; Bauman et al. 1993; Melikov et al. 1994a, 1994b; Fountain et al. 1994; Arens et al. 1998; Szokolay 1998; Tsuzuki et al. 1999; Khedari et al. 2000; Hayashi et al. 2004; Sekhar et al. 2005; Aynsley 2005 and 2007; Atthajariyakul and Lertsatittanakorn 2008; Sun et al. 2007 and 2008; Watanabe et al. 2009). Only in one study was the fan power reported (Sun et al 2008). The power consumption of cooling fans is considered negligible (usually less then 90 W) and, therefore, it is not reported. However, as already discussed, it was demonstrated that the required power input of cooling fans is a critical factor for an energy-saving strategy used in warm environments (Schiavon and Melikov 2008). Based on comprehensive simulations as well as on defined outdoor conditions and building characteristics, it has been shown that in some buildings the use of cooling fans with power input of more than 20 W actually increases the energy consumption compared to that needed to cool the whole building. For the same cooling effect, the power input of the DF tested in the present study was 16–20 W (i.e., twice as low as the power input of the CF [~40 W]) and, therefore, its CFE index was twice as high. Nevertheless, one should be cautious when recommending the use of the DF instead of the CF. The CF may provide cooling to several occupants, while the DF provides cooling to only one occupant. Individual control with a CF is difficult to achieve in practice when it aims to provide cooling to several occupants who may have different preferences with regard to the air movement. The development of desk fans with a strong cooling capacity and low energy consumption of a few watts, as for example the fans used by Watanabe et al. (2009), Sun et al. (2008), and Schiavon and Melikov (2009), is recommended.

The convection heat loss from the body with cooling fans is mainly based on the velocity and the turbulence intensity of the generated flow. As discussed, the fans tested in the present study generated a nonuniform flow. The velocity distribution at the location of the thermal manikin was rather different as well. The CF and TF generated flow with the highest velocity near the floor, up to 0.6 m (24 in.) above the floor, while the highest velocity generated by the DF and the SF was measured at the head region. The indoor climate standards recommend individual control of the airflow at elevated velocity. Speed control at two or three levels was provided for the fans tested. The control, however, affected the flow mostly in the high-velocity region (i.e., near the floor for the CF and the TF and at the head region for the DF and the SF) and, therefore, resulted mainly in an increase of the local cooling of the body segments exposed to the flow and affected only slightly the whole-body cooling (Figure 8). In this respect, the layout, furniture arrangement, etc. were also factors affecting the local air distribution around the manikin's body.

The used thermal manikin with 23 body segments was able to capture the differences in the local cooling caused by the tested fans generating velocity fields with different nonuniformity. In nonuniform airflow, the local cooling effect depends on the area of the exposed segment. A thermal manikin with only a few segments (large segmental area) may not be able to capture clearly the differences in the performance of fans generating flows with different velocity and turbulence intensity filed. This and other factors affecting the determination of the equivalent temperature discussed by Bohm et al. (1999) should be carefully considered when fan performance is studied.

The CFE index of the four cooling fans was determined under well-defined conditions, based on assumptions of their use in practice in order to verify the sensitivity of the proposed CFE index. The clothing thermal insulation and its distribution over the manikin's body (naked/covered body area ratio), the relative distance and direction between the fan and the manikin, the location of the furniture, the metabolic rate, the mean radiant and air temperatures, and manikin's body posture were not changed, and the latent heat loss was not simulated. Different results would be obtained if one or more of these and other parameters, such as fan size and design, clothing design, air humidity, sweat rate, uniformity of thermal room environment, room boundary conditions, etc., were changed. The influence of these parameters was not quantified in this research, but it should be studied in the future. In order to apply the index in practice, a standard procedure for determining the index should be developed. Other factors should also be considered, such as common location; use of tested fan in practice; number of occupants who can benefit from one cooling fan; maximum nonuniformity of body cooling that is acceptable to the occupants; maximum velocity limitations to avoid blowing of paper; and nonthermal discomfort, such as eye blinking, etc. A standard method for testing fan cooling effect and an index for determining fan efficiency, such as the CFE index proposed in this study, need to be developed and used to allow designers and buyers to make the optimal selection for each practical application.

CONCLUSION

The need to evaluate the cooling effect and the cooling efficiency of fans applied to provide-thermal comfort in warm environment was highlighted. A new index, the *cooling-fan efficiency* (CFE) *index*, defined as the ratio between the cooling effect of the used device and its power consumption was introduced for evaluation of the performance of cooling fans.

The measurements performed with a ceiling fan, desk fan, standing fan, and tower fan in a real office at three room air temperatures and different fan speed levels revealed that the index is sensitive enough to identify differences in the performance of the cooling devices. The results identify a large variation in the whole-body cooling effect (between –3.2°C and –0.4°C [–5.76°F and –0.72°F]), in the fan power (between 15.6 and 49.3 W), and in the CFE index (between 0.009°C/W and 0.177°C/W [0.016°F/W and 0.319°F/W]). The local cooling effect for body segments caused by the fans was strongly nonuniform.

A standard method for testing fan cooling effect and an index for determining fan efficiency, such as the CFE index proposed in this study, need to be developed and used.

The cooling fans generate a nonuniform speed field around occupants that cannot be described with a single value. This makes the recommendation in the present thermal comfort standards for elevated velocity in warm environments difficult to use in practice. The present thermal comfort standards need to be revised to better address the issue of thermal comfort in warm environments.

ACKNOWLEDGMENTS

For valuable advice on the experimental stage, the authors would like to thank Dr. Fang Lei. The present work was supported by the International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, and by the Ing. Aldo Gini Foundation.

NOMENCLATURE

CFE = cooling-fan efficiency index, °C/W

$h_{cal,i}$ = dry-heat transfer coefficient of i-th segment of the manikin, determined during calibration, W/m^2°C

Pf = fan power, W

$Q_{t,i}$ = sensible heat loss of i-th segment, W/m^2

Δt_{eq} = whole-body cooling effect, °C

$\Delta t_{eq,i}$ = segmental cooling effect, °C

t_a = ambient air temperature or room air temperature, °C

t_{eq} = whole-body manikin based equivalent temperature, °C

$t_{eq,i}$ = segmental equivalent temperature, °C

$t_{sk,i}$ = skin temperature of i-th segment of the manikin, °C

REFERENCES

Arens, E.A., T. Xu, K. Miura, H. Zhang, M. Fountain, and F. Bauman. 1998. A study of occupant cooling by personally controlled air movement. *Energy and Building* 27:45–9.

Arens, E.A., F.S. Bauman, L.P. Johnson, and H. Zhang. 1991. Testing of localized ventilation systems in a new controlled environmental chamber. *Indoor Air* 1(3):263–81.

Arens, E.A., H. Zhang, and C. Huizenga. 2006. Partial- and whole-body thermal sensation and comfort. Part I: Uniform environmental conditions. *Journal of Thermal Biology* 31:53–9.

ASHRAE. 2004. *ANSI/ASHRAE Standard 55-2004*, *Thermal Environmental Conditions for Human Occupancy*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2005. *ANSI/ASHRAE Standard 113-2005, Method of Testing for Room Air Diffusion*. Atlanta: American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.

Atthajariyakul, S., and C. Lertsatittanakorn. 2008. Small fan assisted air conditioner for thermal comfort and energy saving in Thailand. *Energy Conversion and Management* 49(10):2499–504.

Aynsley, R. 2005. Saving energy with indoor air movement. *International Journal of Ventilation* 4(2):167–75.

Aynsley, R. 2007. Quantifying the cooling sensation of air movement. *International Journal of Ventilation* 7(1):67–76.

Barrentine, L.B. 1999. *An Introduction to Design of Experiments: A Simplified Approach*. Milwaukee: American Society for Quality Press.

Bauman, F.S., H. Zhang, E.A. Arens, and C. Bentonm. 1993. Localized comfort control with a desktop task conditioning system: laboratory and field measurements. *ASHRAE Transactions* 99(2):733–49.

Bauman, F.S., T.G. Carter, A.V. Baughman, and E.A. Arens. 1998. Field study of the impact of a desktop task/ambient conditioning system in office building. *ASHRAE Transactions* 104(1):1153–171.

Bohm, M., O. Norén, I. Holmér, and H. Nilsson. 1999. Factors affecting the equivalent temperature measured with thermal manikins. *Proceedings of Third International Meeting on Thermal Manikin Testing, 3IMM, Stockholm, Sweden,* pp. 45–57.

CEN. 2003. *EN 12792-2003, Ventilation for Buildings—Terminology, Symbols and Units*. Brussels: European Committee for Standardization.

CEN. 2007. *EN 15251-2007, Criteria for the Indoor Environment Including Thermal, Indoor Air Quality, Light and Noise.* Brussels: European Committee for Standardization.

Fountain, M.E., and C. Huizenga. 1997. A thermal sensation prediction tool for use by the profession. *ASHRAE Transactions* 103(2):130–36.

Fountain, M., E. Arens, R. de Dear, F. Bauman, and K. Miura. 1994. Locally controlled air movement preferred in warm isothermal environments. *ASHRAE Transactions* 100(2):937–52.

Gagge, A.P., J. Stolwijk, and Y. Nishi. 1971. An effective temperature scale based on a simple model of human physiological regulatory response. *ASHRAE Transactions* 77(1):247–62.

Hayashi, J., S. Lee, Y. Tomioka, J. Harigaya, T. Nobe, and S. Tanabe. 2004. The effect of isothermal task unit on thermal comfort. RoomVent 2004, September 5–8, Coimbra, Portugal.

IEC. 1980. *Publication 688-1 Electrical Measuring Transducers for Converting AC Electrical Quantities Into DC Electrical Quantities, Part1: General Purpose Transducers.* Geneva: International Electrotechnical Commission.

Insightful. 2007. *S-PLUS 6 for Windows User's Guide*. Seattle: Insightful Corporation.

ISO. 1993. *Guide to the Expression of Uncertainty in Measurement (GUM)*. Geneva: International Organization for Standardization.

ISO. 1998. *ISO 7726, International Standard: Ergonomics of the Thermal Environment—Instruments for Measuring Physical Quantities.* Geneva: International Organization for Standardization.

ISO. 2004. *ISO 14505-2, Ergonomics of the Thermal Environment—Evaluation of Thermal Environment in Vehicles—Part 2: Determination of Equivalent Temperature.* Geneva: International Organization for Standardization.

ISO. 2005. *ISO 7730, Moderate Thermal Environment—Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort.* Geneva: International Organization for Standardization.

Jones, B.W., K. Hsieh, and M. Hashinaga. 1986. The effect of air velocity on thermal comfort at moderate activity levels. *ASHRAE Transactions* 92(2B):761–69.

Khedari, J., N. Yamtraipat, N. Pratintong, and J. Hinrunlabbh. 2000. Thailand ventilation comfort chart. *Energy and Buildings* 32(3):245–49.

Mayer, E., and R. Schwab. Direction of low turbulent air flow and thermal comfort. *Proceedings of Healthy Buildings '88, Stockholm, Sweden*, pp. 577–81.

McIntyre, D.A. 1978. Preferred air speeds for comfort in warm conditions. *ASHRAE Transactions* 84(2):264–77.

Melikov, A.K. 2004. Personalized ventilation. *Indoor Air* 14 (supplement 7):157–67.

Melikov, A.K., R. Arakelian, L. Helkjaer, P.O. Fanger. 1994b. Spot cooling—Part 2: Recommendations for design of spot cooling systems. *ASHRAE Transactions* 100(2):500–10.

Melikov, A.K., R. Cermak, and M. Majer. 2002. Personalized ventilation: Evaluation of different air terminal devices. *Energy and Buildings* 34(8):829–36.

Melikov, A.K., L. Helkjaer, R. Arakelian, and P.O. Fanger. 1994a. Spot cooling—Part 1: Human response to cooling with air jet. *ASHRAE Transactions* 100(2):476–99.

Melikov, A.K., Z. Popiolek, M.G. Silva, I. Care, and T. Sefker. 2007. Accuracy limitations for low velocity measurements and draft assessment in rooms. *HVAC&R Research* 13(6):971–86.

Olesen, B.W., and G.S. Brager. 2004. A better way to predict comfort: The new ASHRAE Standard 55-2004. *ASHRAE Journal* August:20–26.

Psikuta, A., M.G.M. Richards, and D. Fiala. 2008. Single-sector thermophysiological human simulator. *Physiological Measurements* 29:181–92.

Rohles, F., S. Konz, and B. Jones. 1983. Ceiling fans as extenders of the summer comfort envelope. *ASHRAE Transactions* 89(1A):245–63.

SAE. 1993. *SAE J2234, Equivalent temperature*. SAE Information Report, Society of Automotive Engineers, Troy, MI.

Scheatzle, D., H. Wu, and J. Yellott. 1989. Extending the summer comfort envelope with ceiling fans in hot, arid climates. *ASHRAE Transactions* 100(1):269–80.

Schiavon, S., and A. Melikov, A. 2008. Energy saving and improved comfort by increasing air movement. *Energy and Buildings* 40(10):1954–60.

Schiavon, S., and A. Melikov. 2009. Evaluation of the cooling fan efficiency index for a desk fan and a computer fan. *Proceedings of International Conference Roomvent 2009, Busan, South Korea,*

Sekhar, S.C. 1995. Higher space temperatures and better thermal comfort a tropical analysis. *Energy and Buildings* 23:63–70.

Sekhar, S.C., N. Gong, K.W.D. Cheong, K.W. Tham, A.K. Melikov, D. Wyon, and P.O. Fanger. 2005. Findings of personalized ventilation studies in a hot and humid climate. *HVAC&R Research* 11(4):603–20.

Sun, W., A.K. Melikov, and K.W.D. Cheong. 2008. Local climate and heat loss from human body with a novel enhanced displacement ventilation system. Indoor Air 2008, August 17–22, Copenhagen, Denmark.

Sun, W., K.W. Tham, W. Zhou, and N. Gong. 2007. Thermal performance of a personalized ventilation air terminal device at two different turbulence intensities. *Building and Environment* 42:3974–83.

Szokolay, S. 1998. Thermal comfort in the warm-humid tropics. *Proceeding of the 31th Annual Conference of the Australian and New Zealand Architectural Science Association, University of Queensland, Brisbane, Australia,* pp. 7–12.

Tanabe, S., and K. Kimura. 1987. Thermal comfort requirements under hot and humid conditions. ASHRAE Far East Conference, September 3–5, Singapore.

Tanabe, S., E.A. Arens, F.S. Bauman, H. Zhang, and T.L. Madsen. 1994. Evaluating thermal environments by using a thermal manikin with controlled skin surface temperature. *ASHRAE Transactions* 100(1):39–48.

Toftum, J., G. Zhou, and A.K. Melikov. 1997. Airflow direction and human sensitivity to draught. Proceedings of CLIMA 2000, August, Brussels, Belgium.

Toftum, J., A.K. Melikov, A. Tynel, M. Bruzda, and P.O. Fanger. 2003. Human response to air movement: Evaluation of ASHRAE's draft (RP-843). *HVAC&R Research* 9(2):187–202.

Tsuzuki, K., E.A. Arens, F.S. Bauman, and D.P. Wyon. 1999. Individual thermal comfort control with desk-mounted and floor-mounted task/ambient conditioning (TAC) systems. *Proceedings of Indoor Air 1999* 2:368–73.

Watanabe, S., A.K. Melikov, and L.G. Knudsen. 2005. Evaluation of an individually controlled environment by means of a breathing thermal manikin. *Proceedings of The Third International Conference on Human-Environment System ICHES '05, Tokyo, Japan*, pp. 58–61.

Watanabe, S., T. Shimomuraa, and H. Miyazakia. 2009. Thermal evaluation of a chair with fans as an individually controlled system. *Building and Environment* 44(7):1392–98.

APPENDIX A

The body of the thermal manikin consists of 23 independently controlled segments. The thermal manikin measures the power consumption or heat loss Q_t (in W/m^2) and the surface temperature t_{sk} (in °C). The dry-heat transfer coefficient due to free convection $h_{cal,i}$ for each body segment and for the whole body were obtained from calibration of the manikin in an indoor environmental chamber with a uniform thermal environment (i.e., air temperature equal to the mean radiant temperature and air velocity lower than 0.06 m/s [12 fpm]). Under these conditions, the room air temperature is equal to the equivalent temperature (defined by Equation 2). The heat loss and surface temperature of the manikin change when measurements are performed in a nonuniform environment with different radiant temperature and increased velocity in comparison with the uniform environment of calibration. The control of the manikin keeps its surface temperature equal to the skin temperature of an average person in a state of thermal comfort (Tanabe et al. 1994). Knowing surface temperature, power consumption, and heat transfer coefficient, the equivalent temperature of uniform environment, which causes the same heat loss from the body, is calculated. The calibration was performed at three air temperatures, 24°C, 27°C, and 31°C (75.2°C, 80.6°C, and 87.8°F). The results listed in Table A2 show that $h_{cal,i}$ was different for the body segments of the manikin, and for most of them it did not change substantially for the calibration temperature range. During these experiments the left hand of the manikin was broken and, therefore, is not included in the measurements and the calculations. During the calibration, the manikins' clothing and posture were as in the experiments. The names and the body surface areas of the manikin's body segments are listed in Table A1.

APPENDIX B

Uncertainty of the Measurement

The data were analyzed in accordance with the ISO guideline (1993) for the expression of uncertainty. The sample standard uncertainty U was calculated as the combination of the maximum uncertainty of measurement (random error) U_{meas} and the uncertainty of the instrument (calibration) U_{instr}. Table B1 summarizes the typical values of absolute uncertainty based on the analyses of measurements. The values are given for each uncertainty component together with the sample uncertainty U and the uncertainty of a derived quantity U_c. The instrument uncertainty U_{instr} was the strongest component in the case of manikin-based equivalent temperature and air temperature. The uncertainty of process stability was the strongest component in the case of fan power. When presented, the uncertainty is indicated by means of error bars. The level of confidence is 95% (coverage factor of 2). The thermal anemometer was carefully calibrated before the measurements. Detail uncertainty analyses of this type of anemometer are reported by Melikov et al. (2007).

APPENDIX C

The measured cooling effect and fan power and the determined CFE index for the experimental conditions tested is shown in Table C1.

Table A1. Surface Area of Manikin's Body Segments

Number	Body part	Area of the Body Part	
		m^2	ft^2
1	Left foot	0.043	0.463
2	Right foot	0.043	0.463
3	Left lower leg	0.09	0.969
4	Right lower leg	0.09	0.969
5	Left front thigh	0.08	0.861
6	Left back thigh	0.08	0.861
7	Right front thigh	0.083	0.893
8	Right back thigh	0.083	0.893
9	Pelvis	0.055	0.592
10	Back side	0.11	1.184
11	Skull	0.05	0.538
12	Left face	0.0258	0.2777
13	Right face	0.0258	0.2777
14	Back of neck	0.0248	0.2670
15	Left hand	0.038	0.409
16	Right hand	0.037	0.398
17	Left forearm	0.05	0.538
18	Right forearm	0.05	0.538
19	Left upper arm	0.073	0.786
20	Right upper arm	0.078	0.840
21	Left chest	0.07	0.753
22	Right chest	0.07	0.753
23	Back	0.13	1.399
	Total Area	**1.48**	**15.92**

Table A2. Dry-Heat Transfer Coefficient of *i*-th Segment of the Manikin*

Body part	$h_{cal,i}$, W/(m^{2}K)			$h_{cal,i}$, Btu/(h·ft^2°F)		
	24°C	27°C	31°C	75.2°F	80.6°F	87.8°F
Left foot	4.62	4.72	5.20	0.814	0.831	0.916
Right foot	4.57	4.60	4.98	0.805	0.811	0.877
Left lower leg	5.38	5.10	5.85	0.947	0.899	1.029
Right lower leg	5.06	5.02	5.60	0.891	0.884	0.987
Left front thigh	5.86	5.12	6.19	1.033	0.902	1.090
Left back thigh	6.05	5.42	6.05	1.066	0.955	1.065
Right front thigh	6.00	5.04	6.02	1.057	0.888	1.060
Right back thigh	6.40	5.68	6.09	1.127	1.001	1.073
Pelvis	2.90	2.99	2.94	0.511	0.526	0.518
Back side	4.11	3.75	4.41	0.723	0.660	0.777
Skull	2.71	2.89	3.30	0.477	0.510	0.581
Left face	6.50	7.13	7.54	1.145	1.256	1.328
Right face	6.51	7.53	8.59	1.147	1.326	1.513
Back of neck	3.94	4.06	4.05	0.693	0.715	0.714
Right hand	6.98	7.71	6.37	1.230	1.358	1.122
Left forearm	5.23	5.41	5.16	0.922	0.952	0.910
Right forearm	4.95	5.09	4.99	0.872	0.897	0.878
Left upper arm	4.52	4.30	4.39	0.797	0.758	0.774
Right upper arm	4.47	4.37	4.30	0.788	0.769	0.757
Left chest	3.39	3.29	3.42	0.597	0.580	0.603
Right chest	3.48	3.43	3.28	0.613	0.604	0.577
Back	4.28	4.36	4.54	0.754	0.767	0.800

* Coefficients were determined by calibration at three air temperatures: 24°C, 27°C, and 31°C (75.2°F, 80.6°F, and 87.8°F).

Table B1. The Sample Uncertainty, *U*, and the Uncertainty of a Derived Quantity, U_c, Are Reported with a Level of Confidence of 95%

Quantity	U_{meas}	U_{instr}	U	U_c
Manikin-Based Equivalent Temperature, t_{eq}	<0.05°C, 60 readings	0.2°C	0.21°C	Δt_{eq}: 0.3°C *CFE*: 0.009°C/W
Air Temperature, t_a	0.2	0.1°C	0.22°C	
Fan Power, P_f	1 W	0.5 W[a]	1.3 W	*CFE*: 0.009°C/W
Air Velocity	—	See footnote b	See footnote b	
Relative Humidity		See footnote c	See footnote c	

[a] The fan power (P_f) was measured with an accuracy of ±0.5% of the full scale (100 W). The instrument was in class 0.5 according to IEC (1980).
[b] 0.02±1% of the readings for velocity range between 0.05 and 1 m/s, and accuracy of ±3% of the readings for velocity range between 1 and 5 m/s.
[c] ±2% of the readings for the relative humidity range from 0% to 90%, and ±3% for the range from 90 to 100%.

Table C1. Whole-Body Cooling Effect, Fan Power, and Cooling-Fan Efficiency Index for the Four Cooling Fans (Ceiling Fan, Desk Fan, Standing Fan and Tower Fan) and for the Three Room Temperature Levels (t_a = 25°C, 27°C, and 30°C [77°F, 80.6°F, and 86°F])

Type of Fan	Speed Level	Room Air Temperature		Cooling Effect (Δt_{eq})		Fan Power, W (P_f)	Cooling-Fan Efficiency (CFE)	
		°C	°F	°C	°F		°C/W	°F/W
CF	1	25	77	–2	–3.6	38.7	0.051	0.092
CF	2	25	77	–3.2	–5.8	40.2	0.079	0.142
CF	3	25	77	–2.9	–5.2	46.8	0.062	0.112
CF	1	27	80.6	–0.9	–1.6	38.7	0.023	0.041
CF	2	27	80.6	–2.3	–4.1	39.8	0.057	0.103
CF	3	27	80.6	–2	–3.6	44.9	0.045	0.081
CF	1	30	86	–0.7	–1.3	37.2	0.02	0.036
CF	2	30	86	–0.7	–1.3	38.8	0.018	0.032
CF	3	30	86	–1.1	–2.0	47.6	0.022	0.040
DF	1	25	77	–2.8	–5.0	16.1	0.177	0.319
DF	2	25	77	–2.9	–5.2	20.1	0.146	0.263
DF	1	27	80.6	–1.8	–3.2	16	0.115	0.207
DF	2	27	80.6	–2	–3.6	19.5	0.104	0.187
DF	1	30	86	–1.5	–2.7	15.6	0.095	0.171
DF	2	30	86	–1.9	–3.4	19.3	0.099	0.178
SF	1	25	77	–1.6	–2.9	34.1	0.048	0.086
SF	2	25	77	–1.8	–3.2	40.3	0.044	0.079
SF	1	27	80.6	–1.9	–3.4	34.2	0.055	0.099
SF	2	27	80.6	–2.3	–4.1	39.5	0.058	0.104
SF	1	30	86	–1.6	–2.9	33.5	0.047	0.085
SF	2	30	86	–1.4	–2.5	37.6	0.038	0.068
TF	1	25	77	–2.5	–4.5	37.4	0.066	0.119
TF	2	25	77	–2.2	–4.0	43.6	0.051	0.092
TF	3	25	77	–2.2	–4.0	49.3	0.045	0.081
TF	1	27	80.6	–0.9	–1.6	37.3	0.024	0.043
TF	2	27	80.6	–1.4	–2.5	42.2	0.033	0.059
TF	3	27	80.6	–1.4	–2.5	48.9	0.029	0.052
TF	1	30	86	–0.5	–0.9	38	0.012	0.022
TF	2	30	86	–0.4	–0.7	42.5	0.009	0.016
TF	3	30	86	–0.6	–1.1	46	0.014	0.025