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# Ceiling Fans as Extenders of the Summer Comfort Envelope

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## ABSTRACT

The ASHRAE Standard 55-1981 specifies temperature limits or zones for winter and summer comfort. It states that the upper limit of the summer comfort zone, which is 79°F or 26°C, can be extended to 82°F or 28°C with air velocities of 160 fpm or 0.8 m/s. The manufacturers of ceiling fans claim comfort may be obtained at velocities considerably below the 160 fpm (0.8 m/s) level. They further claim that 82°F (28°C) with a ceiling fan will provide the same amount of comfort as 75°F (24°C) without a fan. Since ceiling fans require less than a penny per hour to operate, their use, as opposed to air conditioning, could represent a large energy savings without affecting human comfort.

To examine these claims, 256 human subjects were exposed to 76°F (24°C), 79°F (26°C), 82°F (28°C), and 85°F (29°C) at 50% rh. During the first hour of the three-hour exposure, no fan was used; this "still air" velocity constituted the control velocity, $V_0$, 10 fpm (0.06 m/s). Following this, a ceiling fan was turned on. During the next four half-hour periods, the subject experienced four different air velocities by moving to different locations in the room. These velocities were: $V_1$ (30 fpm or 0.15 m/s), $V_2$ (50 fpm or 0.25 m/s), $V_3$ (90 fpm or 0.46 m/s), and $V_4$ (200 fpm or 1.02 m/s). These, together with the "no-fan" or control condition, represented five velocity parameters.

Four subjective measurements were (1) thermal sensation, (2) thermal comfort, (3) temperature preference, and (4) air plume quality.

The velocity/temperature trade-off is estimated as 25 fpm/°F (.22 m/s /°C). The temperature/velocity trade-off thus is 0.04°F/fpm (4.5°C/1 m/s).

For a representative velocity of 140 fpm (0.7 m/s), this was equivalent to lowering the ambient temperature by (140 fpm)x(.04°F/fpm) = 5.6°F (3.1°C). The National Bureau of Standards suggests a reduction in air conditioning demand of 3%/°F (5.4%/C). Thus the energy saving provided by 140 fpm (0.7 m/s) from a ceiling fan would be 5.6°F x 3%/°F = 17%. Thus it was concluded that a ceiling fan may extend the upper limit of the summer comfort envelope from 79°F (26°C) to 85°F (29°C) (the equivalent temperature on any specific situation depends on the velocity of the air on the person). The results suggest that the turbulent and variable characteristics of the air plume of the ceiling fan may be its major comfort-producing feature.


## INTRODUCTION

The new ASHRAE comfort standard, ASHRAE Standard 55-1981 (1), "Thermal Environmental Conditions for Human Occupancy," specifies two comfort zones--

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one for winter and one for summer. Moreover, extension of the upper limit of the summer comfort zone of 79°F (26°C) to 82°F (28°C) is suggested if the air velocity is increased from 50 fpm (0.25 m/s) to 160 fpm (0.8 m/s).

While many different air-moving devices are available, the increased popularity of the ceiling fan, coupled with its low operating costs, made it the obvious choice for study. Specifically this study was concerned with three questions. First, was the ceiling fan capable of extending the summer comfort envelope; second, were the claims of the ceiling fan manufacturers valid that 82°F (28°C) with a fan provides the same amount of comfort as 75°F (24°C) without a fan; and third, could the ceiling fan be considered as an energy-saving device?

## METHOD

Subjects. The subjects were 256 college students (128 men and 128 women) ranging in age from 18 to 22 years. Each had lived within the continental limits of the United States for at least six months prior to the test, and each was paid $12 for participating. No one served as a subject more than once during the study.

Design. Four temperature conditions were studied: 76 FET* (24 CET*), 79 FET* (26 CET*), 82 FET* (28 CET*), and 85 FET* (29 CET*); the relative humidity was 50% for all temperatures. The subjects were tested in groups of eight--four men and four women. Each pair (one man and one woman) was subjected to five air velocity conditions throughout the three-hour exposure. The subjects moved to different locations in the room each half-hour. During the first hour, the fan was not used; this "still air" condition represented the control velocity, $V_o$ (10 fpm or 0.06 m/s). The other four velocities, as measured at 43 in. (109 cm) above the floor were: $V_1$ (30 fpm or 0.15 m/s); $V_2$ (50 fpm or 0.25 m/s); $V_3$ (90 fpm or 0.46 m/s); and $V_4$ (200 fpm or 1.02 m/s). The sequence of exposure to the various velocities was randomized for each pair of subjects within the total group of eight subjects. Thus, for the first hour of exposure, the subjects were exposed to the "still air". Then, for the subsequent half-hour periods, they were exposed to each of the remaining four velocities.

Air Velocity Determinations. Fig. 1 shows the test room layout and work-stations. The arrangement provided two locations at each of four different velocities whose nominal values were 30, 50, 90 and 200 fpm (.15, .25, .46, and 1.02 m/s). Air velocities were measured at 4 in. (10 cm), 24 in. (61 cm), and 43 in. (109 cm) from the floor, the same heights as specified in ASHRAE Standard 55-1981. However, the high velocity gradients with the ceiling fan prohibited further compliance with the standard (which emphasizes average velocities in the room). At some locations the velocity varied from 200 fpm (1.02 m/s) to 50 fpm (0.25 m/s) within just a few feet. In order to accurately estimate the velocity to which the subjects were exposed, the velocities were measured 6 in. (15.5 cm) back from the edge of the table (the approximate location of the subject during the tests).

The measurements were taken using an anemometer that is directional in nature. Smoke was released at each location to determine the direction of the airflow for orientation of the anemometer during the measurements. Data were collected for 80 seconds at each location and recorded on a digital storage oscilloscope. Means and standard deviations then were determined using 10 values from the recording equally spaced in time over the 80 seconds. These measurements were repeated at each location to verify the values calculated. Tab. 1 presents the results of the initial 10 measurements.

The standard deviation is quite large for most locations -- typically 30% to 50% of the mean. The air movement was "gusty" in nature at most locations and the direction varied along with the gustiness. This nature of the air movement made it difficult to measure and average the air velocity accurately and tolerances of ±20% should be assigned to the numbers in Tab. 1. The locations of the subjects were selected on the basis of the velocity at the 43 in. (109 cm) level since it is likely that the velocity at this level has the largest impact on thermal comfort. On this basis, the nominal velocity conditions were:

| Locations | Nominal Velocity fpm | Nominal Velocity m/s |
|---|---|---|
| $V_{1a}$, $V_{1b}$ | 30 | 0.15 |
| $V_{2a}$, $V_{2b}$ | 50 | 0.25 |
| $V_{3a}$, $V_{3b}$ | 90 | 0.46 |
| $V_{4a}$, $V_{4b}$ | 200 | 1.02 |

Apparatus and Facilities. All of the tests took place in the KSU-ASHRAE chamber. This temperature-humidity controlled environmental chamber is 24 ft. (7.3 m) long and 12 ft. (3.7 m) wide and has a ceiling height of 10 ft. (3.0 m). Fig. 1 shows the seating arrangement of the eight subjects; the subjects were seated at tables either individually or with another person (Fig. 2). Task lighting was provided with a 15 W fluorescent lamp at each work station; area lighting was provided by "ultralume" fluorescent lamps located behind a valance on the two 24-ft. (7.3-m) walls.

A 52-in. diameter (132-cm) ceiling fan was centered on the 12-ft. (3.7-m) wall and mounted 6 ft. (1.8 m) from that wall (Fig. 1); the blades of the fan were 18 in. (46 cm) from the ceiling.

Each subject was provided with a short-sleeve shirt, trousers, socks, and sandals. When worn with briefs (males) or panties and bra (females), this ensemble measures 0.5 clo.

Four criteria were measured. The first ballot, the thermal sensation response, was measured on a nine-category rating scale. The second ballot, thermal comfort, consisted of seven bi-polar adjective-pairs arranged in a semantic-differential scale format. Both of these ballots were reproduced on computer cards (Fig. 3).

The third subjective scale measured air plume quality. The scale as administered during the test contained 34 adjective-pairs (Fig. 4). The fourth ballot measured temperature preference, that is whether the subject desired the temperature to be warmer and by how many degrees, cooler and by how many degrees, or left unchanged (Fig. 5).

Procedure. When the subjects reported for the test, their oral temperatures were taken. If the reading was 98.6°F ±0.5°F (37°C ±0.3°C), the subjects disrobed, dressed in the clothing ensemble provided, and were seated in the pre-test room (temperature 72°F [22°C]/40% rh).

After the subjects were oriented and fully understood the procedures, they entered the chamber. After one-half hour, the subjects voted and then moved to a different work station. At the end of the second half-hour, they voted again and moved to a different work station; at this time the fan was turned on. Then for the four remaining half-hour periods they followed the same procedure (i.e., voting and moving to a new work station). At the end of three hours, when the test was completed, the subjects left the chamber, changed into their own clothes, were paid $12, and were dismissed.

## RESULTS

Each of the votes was treated by analysis of variance. Tab. 2 presents the resulting F-ratios for the four dependent measures.

Thermal Sensation. As expected, temperature was significant. From a Duncan's Multiple Range Test (DMRT) (Tab. 3), the responses to each temperature were significantly different from one another. Consistent with previous research, the mean thermal sensation of the men (4.9) was significantly above that of the women (4.5). The Temperature by Sex interaction was significant.

Air velocity also had a significant F-ratio. Tab. 4 demonstrates that, in effect, four air velocities are being studied, since the mean thermal sensations resulting from exposure to $V_2$ (50 fpm [.25 m/s]) and $V_3$ (90 fpm [0.46 m/s]) are the same. Since the thermal sensation in the $V_o$ condition was different from the thermal sensation in the $V_1$ condition, Tab. 4 also demonstrates a possible placebo effect of $V_1$. Under the $V_o$ condition the fan was off. However, under the $V_1$ condition, the velocity is extremely low (30 fpm [.15 m/s]) and probably was unable to be perceived. Yet the subjects could see the fan turning which may have accounted for the reported cooler thermal sensation.

Thermal Comfort. The comfort ballots were scored by assigning values ranging from nine for the most favorable of the adjective-pairs to one for the least favorable. Then the rating for each adjective-pair was multiplied by its respective loading. These seven products (one for each adjective-pair) were summed. From this sum, 4.270 was subtracted and the result was multiplied by 2.92. The resulting value constituted the thermal comfort vote in the form of a percent (Fig. 6). The responses on the thermal comfort ballot also were treated by analysis of variance. As was the case in the thermal sensation votes, the F-ratio associated with the temperature also was significant for the thermal comfort votes (Tab. 5).

Maximum comfort was at 82 F (28 C). However the difference between 82 F (28 C) and 85 F (29 C) was not significant. Both of these conditions were more comfortable than 79 FET* (26 CET*) which, in turn, was more comfortable than 76 FET (24 CET*).

The F-ratios associated with all of the sources involving velocity also were statistically significant. Tab. 6 shows the similarity in the comfort response under the $V_2$ and $V_3$ velocities. However, rather than discussing each of these separately, the mean thermal comfort votes are presented graphically in Fig. 7.

Fig. 7 involves two families of curves. One encompases the control velocity, $V_o$, and the placebo velocity, $V_1$. At 76 FET* (24 CET*), thermal comfort under the $V_o$ and $V_1$ conditions is similar. However, at the other three temperatures the means for the $V_1$ conditions are consistently above the mean comfort at the control velocity, $V_o$. The second family of curves includes the velocities of $V_2$, $V_3$ and $V_4$. These velocities contribute to cool discomfort at 76 FET* (24 CET*) and 79 FET* (26 CET*). However, they are responsible for similar (and indeed optimal comfort) at 82 FET* (28 CET*) and 85 FET* (29 CET*).

Air Plume Quality. To develop the scale, approximately 20 persons made a list of adjectives that could be used to describe the quality of the air plume. This procedure yielded 34 different bi-polar adjective-pairs, and these constituted the experimental form of the Air Plume Quality Scale. This scale, presented in Fig. 4, was administered at each time the other scales were administered. Each adjective-pair was scored by assigning a value of 9 to the most favorable of the adjectives and a value of 1 to the least favorable. Then, using the procedures developed by Rohles and Milliken (2) involving analysis of variance and factor analysis, a scale containing 12 adjective-pairs was identified (22 pairs were discarded due to lack of statistical significance). The adjectives-pairs, which constituted the final Air Plume Quality Scale, were scored in the same manner as the thermal comfort responses (Fig. 8). The resulting air plume quality score then was subjected to an analysis of variance. Temperature was statistically significant at $p < .0003$. The mean APQ scores were subjected to a Duncan Multiple Range Test. Tab. 7 shows maximum comfort at 82 F (28 C) and the air plume felt the same at 82 FET* (28 CET*) and 85 FET* (29 CET*). Both of these means were significantly greater than the mean APQ at 79 FET* (26 CET*), which in turn was greater than the mean APQ at 76 FET* (24 CET*). These findings are similar to those obtained with the thermal comfort votes (Tab. 5).

The mean APQ votes at each temperature were plotted for each of the five separate velocities (Fig. 9). The similarity between these curves and the thermal comfort curves is apparent. The correlation between TC and APQ was 0.89, even though only three of the seven adjective-pairs that appeared in the thermal comfort scale also appeared in the APQ scale; these were "acceptable-unacceptable", "comfortable-uncomfortable", and "pleasant-unpleasant". The quality of both the air plume and thermal comfort was greatest at 82 FET* (28 CET*) and 85 FET* (29 CET*) with velocities $V_2$, $V_3$, and $V_4$.

Temperature Preference. The mean preferred change in temperature (°F) was treated by analysis of variance. Temperature, sex, and velocity were statistically significant. Tab. 8 shows the DMRT. The means differed significantly from one another at $p < .05$, but, more important, the extent to which the subjects at 76 FET* (24 CET*) wanted the temperature to be warmer (+1.65°F) was essentially the same as the extent they wanted it to be cooler (-1.70°F) at 85 FET* (29 CET*). A similar finding may be seen at 79 FET* (26 CET*) and 82 FET* (28 CET*). Tab. 8 also provides evidence that the condition of thermal neutrality, that is, where neither an increase nor a decrease in temperature was preferred, was between 79°F (26°C) and 82°F (28°C) -- if we assume the optimal condition for comfort to exist in a range of ±1 degree F from zero change.

Sex also was significant at $p < .0001$. Tab. 9 shows that men preferred the temperature to increase by 0.4°F, whereas the women preferred the temperature to decrease by 0.4°F. However, when the mean degrees change in preferred temperature was analyzed separately for men and women, greater differences were found at the various temperatures.

Velocity also was significant at $p < .0001$. Tab. 10 presents the means. These results are similar to those of the APQ score and thermal comfort scores. They demonstrate the similarity between the responses under the $V_2$ and $V_3$ velocity conditions.

Fig. 10 presents the means graphically. If one assumes the optimal condition for comfort to exist in a range of ±1 degree F from zero change in the preferred temperature, under still air conditions, $V_o$, the preferred temperature was 76 FET* (24 CET*). At $V_1$, the preferred temperature was 79 FET (26 CET*). In turn, 82 FET* (28 CET*) was the preferred temperature when the velocities were $V_2$, $V_3$ and $V_4$; however, the velocities had to be at the $V_3$ and $V_4$ levels at 85 FET* (29 CET*). This is the first time this version of the temperature preference scale has been used. As such, it introduces a new dimension to comfort research.

## DISCUSSION

ASHRAE Standard 55-1981 places the upper limit of the summer comfort envelope at 79 FET* (26 CET) for "still air" (velocity ≤ 50 fpm or .25 m/s). It points out, however, that the zones may be extended to 82 FET* (28 CET*) if the air velocity is increased to 160 fpm (0.80 m/s). The data from the present study validate this criterion. Moreover, the study demonstrates that if the movement of the air in the occupied space is brought about by a ceiling fan developing mean velocities of 200 fpm (1.0 m/s), the summer comfort zone can be extended to as high as 85 FET* (29 CET*). This conclusion is presented graphically in Fig. 11 which adds to the graph in the ASHRAE Standard the extension suggested from this study.

The rationale for suggesting this extension is based on the finding that the average preferred change in temperature at 76 FET* (24 CET*) in still air is approximately the same as at 85 FET* (29 CET*). One important point which should not be overlooked in this conclusion is that the 160 fpm (.80 m/s) maximum velocity of the ASHRAE Standard was established on the basis of a study at Kansas State University (Rohles, Woods, and Nevins, 1974). In their research, a room with a perforated ceiling was built inside the KSU-ASHRAE chamber. This provided a uniform flow of air with little if any variability. In contrast, the air movement developed by the ceiling fan has a highly variable velocity, and the ceiling fan velocities from one point in time to another are random. These facts also are pointed out by McIntyre (3). In addition, the findings of this study should be considered in light of McIntyre's research in which his subjects were asked to select the speed of an overhead fan to optimize their thermal comfort (4). He found preferred air velocities at 86°F (30°C) as high as 400 fpm (2.0 m/s) and general acceptability of 82°F (28°C) with an air speed of 200 fpm (1 m/s). However, McIntyre had his fan above a louvre which may have been responsible for producing an air plume that differed from the one we had in our study. Burton et al (5) also studied ceiling fans and had similar results. However, he did observe that "if the subjects were ranked in order of their preferred fan speeds, they were sorted approximately in reverse order of their ages, with the younger subjects choosing higher air speeds."

The ASHRAE Standard also states that "loose paper, hair and other light objects may start to be blown about at air movements of 160 fpm (0.8 m/s)." However in the air plume generated by the ceiling fan, this observation was not supported. Again, this may have been due to the randomness of the velocity, and in fact, this random characteristic is believed to be one of the main features contributing to the popularity of the ceiling fan. Thus, to state the mean velocity of the air without an accompanying measure of variability may lead to a false interpretation of its comfort-producing qualities. Fig. 12 represents the mean velocities and their standard errors directly under a 52-in. (132-cm) diameter ceiling fan at a height of 43 in. (109 cm) above the floor at a various fan radii (26 in.; 67 cm) from the center. Immediately apparent is the "low velocity" at the fan's center and the maximum velocity that occurs at the end of the blades, a distance of one radius from the center. In addition, the artist's conception of the velocity profiles (as observed on the digital storage oscilloscope) demonstrates a randomness in the velocity of the air plume that resembles white noise. It should be added that these measurements were made in the test chamber when it was empty of furniture and people -- both of which added even greater variability and randomness to the air plume velocity.

One of the questions implicit in this research is, does one feel the same at high temperatures with a fan as lower temperatures without a fan? The results demonstrate that one does--if the velocity is greater than 90 fpm (0.46 m/s). Three types of feelings were measured--thermal sensation, thermal comfort, and temperature preference. If the mean thermal sensation response is examined at 76 FET* (24 CET*), the value is 4.9 on a scale in which 5.0 is "neutral." At 79 FET* (26 CET*) with low velocity air, $V_1$ (30 fpm [0.15 m/s]), 82 FET* (28 CET*) with a velocity of 50 fpm (0.25 m/s) air and 85 FET* (29 CET*) with air velocities ranging from 90 fpm to 200 fpm (0.45 m/s to 1.02 m/s), the thermal sensation responses are approximately 4.7, 4.7 and 4.9, respectively. Similarly, under still air conditions, the level of comfort at 76 FET* (24 CET*) is approximately 70%. In contrast, it is 76%, 74%, and 75% at 79 FET* (26 CET*), 82 FET* (28 CET*) and 85 FET* (29 CET*), respectively. In terms of temperature preference, at 76 FET* (26 CET*) without a fan, the mean degree change preferred in temperature was -0.3°F (-0.17°C). At 79 FET* (26 CET*), 82 FET* (28 CET*), and 85 FET* (29 CET*) the mean number of degrees change that the subjects preferred was +0.3°F (.17°C), -0.4°F (-.22°C) and -0.9°F (-0.5°C) respectively.

These values (Tab. 11) provide conclusive evidence from three different subjective dimensions that the ceiling fan can make one feel the same at high temperatures as one does at low temperatures in still air.

The logical question to this conclusion is why the ceiling fan provides this comfort. The answer probably lies in the quality of the air plume it generates. The random velocity that characterizes the air plume of the ceiling fan may contribute to its preference over other air-moving devices.

Another feature of the ceiling fan (that their manufacturers are quick to point out) is their role in saving energy. One manufacturer claims that using "a ceiling fan in the summer makes 82°F (28°C) feel like 75°F (24°C) for a savings of more than 30 percent." This amounts to approximately 4.3% savings per degree F (30 ÷ 7 = 4.3). The February 1981 issue of Appliance magazine (6) makes even greater claims. It states that 82°F (28°C) with a fan provides the same amount of comfort as 74°F (23°C) in still air. It also points out that these differences can represent an energy savings of 8% per °F.

Using the findings presented in Fig. 10 on the preferred degree change in temperature and the finding that a 75% comfort level is experienced at 85°F (29°C) with a ceiling fan and also at 79°F (26°C) in still air, the authors conclude that the energy demand with a fan is equal to a difference of 3.3°C (29.4°C-26.1°C) or 6°F (85°F-79°F). The National Bureau of Standards (7,8) estimates a 3% energy demand per degree F or 5.4% per °C, which translates this difference into an energy savings of 18%. With this approach, therefore, the ceiling fan obviously will reduce the demand on energy.

There are several very important assumptions in this generality however. The 6°F (3.3°C) temperature difference is based on the results at $V_3$ (90 fpm [.46 m/s]) and $V_4$ (200 fpm [1.0 m/s]). Averaging the velocity of 90 and 200 fpm, this is 145/6 or 24 fpm/°F. Rounding off, one can assume a 25 fpm velocity will permit an increase in air temperature of 1°F; in SI units, this is .22 m/s/°C or 4.5°C per 1 m/s. A fan generates air velocities of these magnitudes only over a relatively small area near the fan. This level of energy savings is possible only if sufficient fans are available to generate velocities comparable to $V_4$ at locations commonly occupied by people. Otherwise the equivalent temperature difference is less and the savings also are likely to be less. The exact savings are going to vary significantly from the above estimate due to differences in climate, house construction, air-conditioning systems, and how well the occupants utilize the fan. Additionally, the energy consumed by the fan is not included in the above estimate. This consumption will be about 75-150 watts for most fans when they are running at maximum speed. At an electricity cost of 5 cents/Kwh, a fan costs less than a penny an hour (.15 x 5 = 0.75 cents/h) to operate.

Finally, special comment appears appropriate on the methodology employed in this study. At the outset, this was the first study in which the thermal sensation and thermal comfort ballots have been printed on computer cards. In this procedure the thermal comfort and thermal sensation votes were obtained from the subjects using optical mark reading (OMR) cards. The conventional recording method used in previous studies involved transferring votes from the ballot to a coding sheet and then manually keypunching the information from the coding sheets to the computer cards. In contrast, the OMR cards enabled machine reading of the votes directly from the ballots themselves. The OMR method reduces the possibility of errors that the conventional method could encounter during the manual transformation of the data to machine readable form. Using a software controlled keypunch, the OMR cards were pre-punched before the test with the test number, subject number, vote number, and the sex of the subject. This pre-punching uniquely identified each card with the corresponding vote. Once the ballots were executed, they were ready to be analyzed. This modification proved to be successful, time saving, and a source of error reduction.

In previous comfort research the thermal preference ballot has been used with considerable success. On those occasions, the subjects were asked to indicate whether they would prefer the temperature to be warmer, cooler, or remain the same. In this study the amount of change in degrees that the subject preferred was added to the question. This addition proved to be a wise decision, since it provided an additional quantifiable dimension to the comfort response. In fact, together with the nine-category thermal sensation response and the response of the empirically derived thermal comfort scale, it increases the reliability and confidence that can be placed in the subjective evaluation of thermal environment.

In conclusion, the popularity that the ceiling fan currently enjoys should continue. The January 7, 1982, issue of the Wall Street Journal estimated that the sales of ceiling fans reached $700 million in 1981; yet there are still many questions that require answers. For example, how effective is the fan for winter use? Many of the fans have reversible motors or blades that will permit an upward throw of air, but the effectiveness of this mode of operation, as compared to the conventional downward mode, is not known. In addition, several models are equipped with rheostats to permit varying speeds. The fan used in this experiment had four blades yet there are models with five and even six blades.

During the course of this study, the authors were asked what distance from the ceiling a fan should be placed when operating in a high bay area of 30 feet. Also, how many fans would be required in an area of varying dimensions? In fact, the industrial setting has not been explored at all. The question of fan size as it relates to space has not been answered; nor have the effects of furniture and occupant density been addressed. Coupled with these items are the questions relating to the occupant's physical activity, posture (standing, sitting or lying down), and age. In addition, how valid is the comparison of fans and air conditioning when one considers that air conditioning lowers the humidity as well as the temperature, whereas fans do not change either the temperature or humidity? Thus, even though the ceiling fan appears to be entering into our lives again, it has generated a host of questions that go beyond the nostalgia that has contributed so much to its present popularity.

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TABLE 1
Mean Velocities and Standard Deviations at the Eight Work-Stations

| Work-Station | Vel. @ 4 in. (10 cm) fpm | (m/s) | Vel. @ 24 in. (61 cm) fpm | (m/s) | Vel. @ 43 in (109 cm) fpm | (m/s) |
|---|---|---|---|---|---|---|
| $V_1$-a | 97±16 | (.49±.08) | 43±24 | (.21±.12) | 32±18 | (.16±.09) |
| $V_1$-b | 145±63 | (.74±.32) | 42±33 | (.21±.17) | 33±22 | (.17±.11) |
| $V_2$-a | 145±61 | (.74±.31) | 79±25 | (.40±.13) | 45±29 | (.23±.15) |
| $V_2$-b | 165±44 | (.84±.22) | 77±26 | (.39±.13) | 48±20 | (.24±.10) |
| $V_3$-a | 72±28 | (.37±.14) | 40±21 | (.20±.11) | 86±33 | (.44±.17) |
| $V_3$-b | 137±71 | (.88±.36) | 77±37 | (.39±.19) | 91±27 | (.46±.14) |
| $V_4$-a | 143±79 | (.73±.40) | 213±94 | (1.08±.48) | 183±70 | (.93±.36) |
| $V_4$-b | 92±46 | (.47±.23) | 199±66 | (1.01±.33) | 283±44 | (1.21±.22) |

TABLE 2
Significance of the F-Ratios for the Sources of Variance for the Four Dependent Measures*

| Source | TS | TC | APQ | TP |
|---|---|---|---|---|
| Temp (T) | .0001 | .0001 | .0003 | .0001 |
| Sex (S) | .0001 | ns | .0389 | .0001 |
| T x S | .0279 | ns | ns | .0084 |
| Velocity (V) | .0001 | .0001 | .0001 | .0001 |
| V x T | ns | .0001 | .0001 | ns |
| V x S | ns | .0001 | .0005 | ns |
| V x T x S | .0255 | .0004 | .0004 | ns |

*TS--Thermal Sensation; TC--Thermal Comfort; APQ--Air Plume Quality; TP--Temperature Preference; ns--not significant at $p < .05$.

TABLE 3
Mean Thermal Sensation Votes at the Four Temperature Conditions

| Temperature FET* | (CET*) | Mean | Grouping** |
|---|---|---|---|
| 85 | (29.4) | 5.4 | A |
| 82 | (27.8) | 5.1 | B |
| 79 | (26.1) | 4.4 | C |
| 76 | (24.4) | 4.0 | D |

**Means with different letter designations differ from one another $p < .05$.

TABLE 4
Mean Thermal Sensation Votes for the Five Velocity Conditions

| Velocity | Mean | Grouping* |
|---|---|---|
| $V_o$ | 5.8 | A |
| $V_1$ | 5.0 | B |
| $V_2$ | 4.4 | C |
| $V_3$ | 4.4 | C |
| $V_4$ | 4.0 | D |

* Means with the same letter designation do not differ from one another at $p < .05$.

TABLE 5
Mean Thermal Comfort Votes for the Four Temperature Conditions

| Temperature FET* (CET*) | Mean Thermal Comfort Vote (%) | Grouping** |
|---|---|---|
| 82 (27.8) | 70 | A |
| 85 (29.4) | 68 | A |
| 79 (26.1) | 64 | B |
| 76 (24.4) | 57 | C |

**Means with the same letter designation do not differ from one another at $p < .05$.

TABLE 6
Mean Thermal Comfort Votes for the Five Velocity Conditions

| Velocity | Mean Thermal Comfort Vote (%) | Grouping* |
|---|---|---|
| $V_1$ | 70 | A |
| $V_2$ | 66 | B |
| $V_3$ | 66 | B |
| $V_o$ | 62 | C |
| $V_4$ | 59 | D |

*Means with the same letter designations do not differ from one another at $p < .05$.

TABLE 7
Air Plume Quality Scores for the Four Temperature Conditions

| Temperature FET* (CET*) | Mean APQ (%) | Grouping** |
|---|---|---|
| 82 (27.8) | 69 | A |
| 85 (29.4) | 67 | A |
| 79 (26.1) | 64 | B |
| 76 (24.4) | 59 | C |

**Means with the same letter designations are not significantly different at $p < .05$.

TABLE 8
Mean Degrees Change in Preferred Temperature for the Four Temperature Conditions

| Temperature FET* | (CET*) | Mean °F Change in Preferred Temp. | Grouping** |
|---|---|---|---|
| 76 | (24.4) | 1.7 | A |
| 74 | (26.1) | 0.9 | B |
| 82 | (27.8) | -1.0 | C |
| 85 | (29.4) | -1.7 | D |

**Means with the same letter designations are not significantly different from one another at $p < .05$.

TABLE 9
Mean Degrees Change in Preferred Temperature by the Men and Women for the Temperatures Combined

| Sex | Mean, °F | Grouping* |
|---|---|---|
| Male | +0.4 | A |
| Female | -0.4 | B |

*Means with different letter groupings differ from one another at $p < .05$.

TABLE 10
Mean Degrees Change in Preferred Temperature for the Five Velocity Conditions

| Velocity | Mean °F | Grouping* |
|---|---|---|
| $V_4$ | 1.2 | A |
| $V_3$ | 0.6 | B |
| $V_2$ | 0.4 | B |
| $V_1$ | -0.5 | C |
| $V_o$ | -2.0 | D |

*Means with the same letter designations are not significantly different from one another at $p < .05$.

TABLE 11
A Comparison of the Three Subjective Responses at Four Different Temperature and Fan Conditions

| Thermal Condition | TS | TC | TP (°F chg) |
|---|---|---|---|
| 76 FET* (24.4 CET*) no fan | 4.9 | 69% | -0.3 |
| 79 FET* (26.1 CET*) fan: 30 fpm or 0.15 m/s | 4.7 | 76% | +0.3 |
| 82 FET* (27.8 CET*) fan: 50-200 fpm or 0.25-1.02 m/s | 4.7 | 74% | -0.4 |
| 85 FET* (29.4 CET*) fan: 50-200 fpm or 0.25-1.02 m/s | 4.9 | 75% | -0.9 |

TS--Thermal Senstion; TC--Thermal Comfort; TP--Thermal Preference

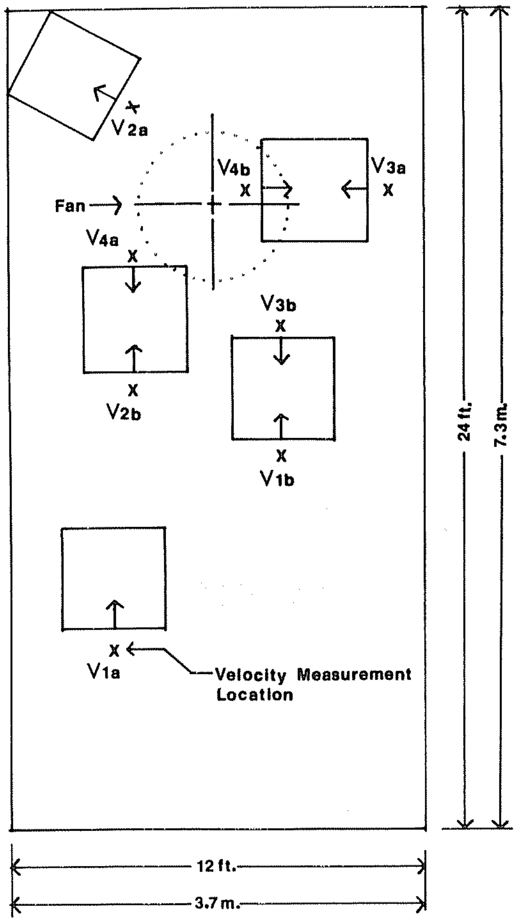



Figure 1. Orientation of the subjects with respect to the fan; the four velocities are identified as $V_1$, $V_2$, $V_3$, and $V_4$; the two work-stations for each velocity are identified as a and b; the arrows show the direction that the subjects were facing

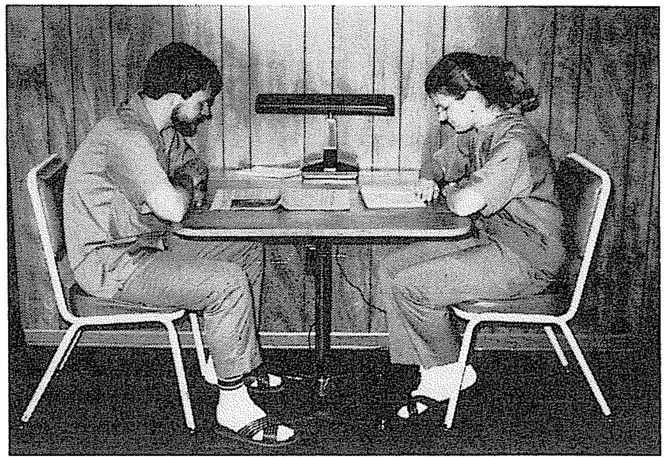


Figure 2. Work-station showing the lighting and the dress of two subjects

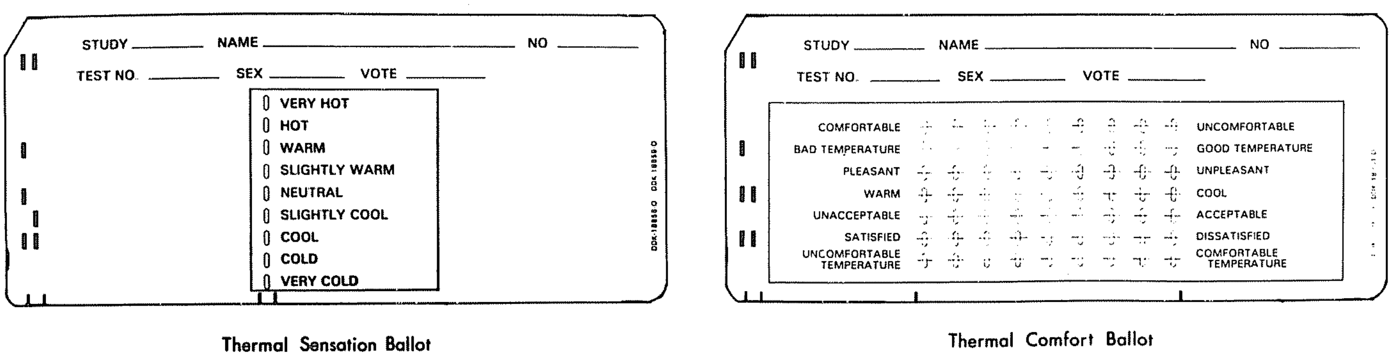



Figure 3. Thermal sensation and thermal comfort ballots as printed on computer cards

# AIR PLUME QUALITY SCALE

| | | |
|---|---|---|
| relaxing | ___:___:___:___:___:___:___:___:___ | tense |
| unacceptable | ___:___:___:___:___:___:___:___:___ | acceptable |
| tear inducing | ___:___:___:___:___:___:___:___:___ | non-tear inducing |
| gusty | ___:___:___:___:___:___:___:___:___ | constant |
| refreshing | ___:___:___:___:___:___:___:___:___ | unrefreshing |
| favorable | ___:___:___:___:___:___:___:___:___ | unfavorable |
| exciting | ___:___:___:___:___:___:___:___:___ | soothing |
| bad circulation | ___:___:___:___:___:___:___:___:___ | good circulation |
| dislike | ___:___:___:___:___:___:___:___:___ | like |
| damp | ___:___:___:___:___:___:___:___:___ | non-damp |
| fresh | ___:___:___:___:___:___:___:___:___ | stale |
| breezy | ___:___:___:___:___:___:___:___:___ | calm |
| clean | ___:___:___:___:___:___:___:___:___ | dirty |
| good odor | ___:___:___:___:___:___:___:___:___ | bad odor |
| noisy | ___:___:___:___:___:___:___:___:___ | quiet |
| bad direction | ___:___:___:___:___:___:___:___:___ | good direction |
| appealing | ___:___:___:___:___:___:___:___:___ | unappealing |
| satisfying | ___:___:___:___:___:___:___:___:___ | annoying |
| useless | ___:___:___:___:___:___:___:___:___ | useful |
| impressive | ___:___:___:___:___:___:___:___:___ | unimpressive |
| distinctive | ___:___:___:___:___:___:___:___:___ | ordinary |
| functional | ___:___:___:___:___:___:___:___:___ | non-functional |
| good ventilation | ___:___:___:___:___:___:___:___:___ | poor ventilation |
| uncomfortable | ___:___:___:___:___:___:___:___:___ | comfortable |
| humid | ___:___:___:___:___:___:___:___:___ | non-humid |
| drafty | ___:___:___:___:___:___:___:___:___ | non-drafty |
| unpleasant | ___:___:___:___:___:___:___:___:___ | pleasant |
| dry | ___:___:___:___:___:___:___:___:___ | not dry |
| gentle | ___:___:___:___:___:___:___:___:___ | brisk |
| subduing | ___:___:___:___:___:___:___:___:___ | stimulating |
| repelling | ___:___:___:___:___:___:___:___:___ | inviting |
| good | ___:___:___:___:___:___:___:___:___ | bad |
| invigorating | ___:___:___:___:___:___:___:___:___ | non-invigorating |
| turbulent | ___:___:___:___:___:___:___:___:___ | smooth |

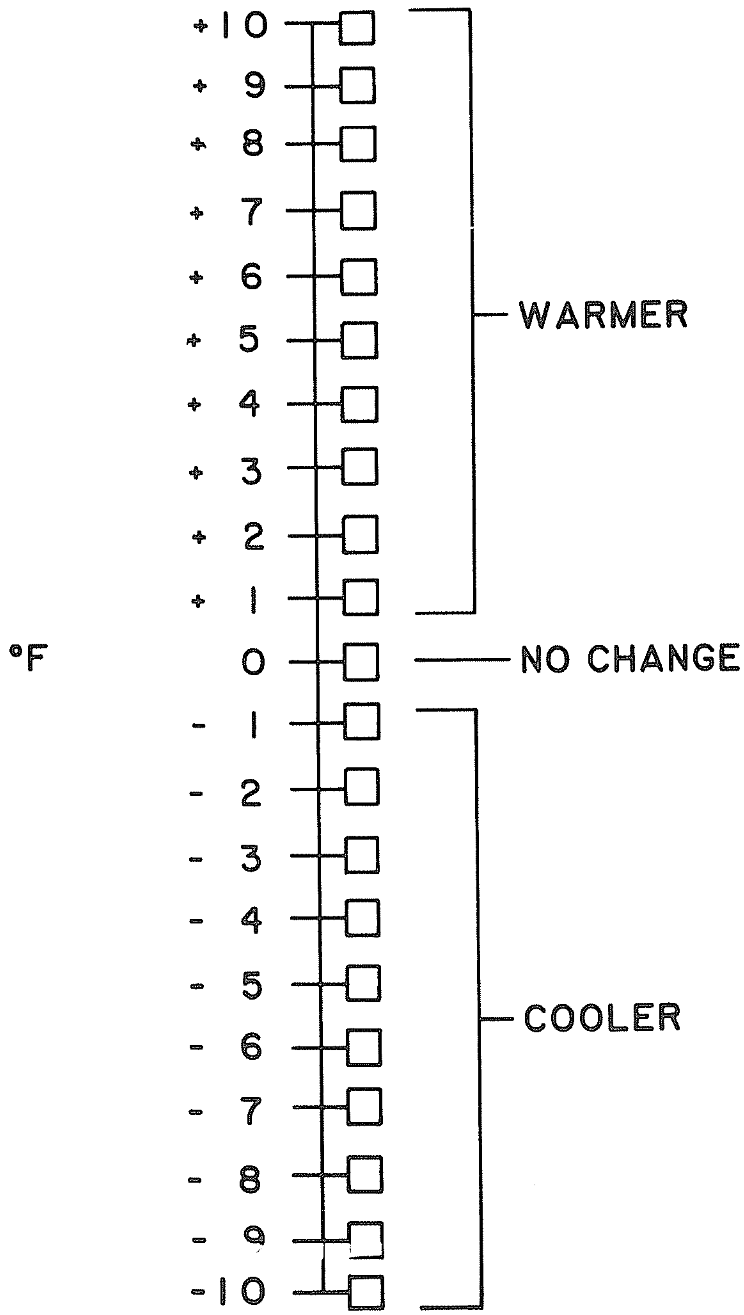
Name_______________________ Subj. No.__________ Test__________

Vote__________ Study__________ Code__________ Code__________

*Figure 4. Air plume quality scale as administered*

TEMPERATURE PREFERENCE

We wish to know if you would like the temperature in this room to be WARMER (and by how many degrees), COOLER (and by how many degrees), or remain the same. Your decision should be indicated on the scale below. For example, if you want the temperature to remain the same, put an X in the box labeled NO CHANGE; if you want the temperature to be 3 degrees F WARMER, mark the box opposite +3; for 5 degrees WARMER, mark +5; if you want the temperature to be 5 degrees COOLER; mark -5, etc.




Name____________________ Subj. No.________ Test________

Vote________ Study__________ Code ________ Code ________

*Figure 5. Thermal preference ballot*

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| comfortable | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | uncomfortable |
| bad temperature | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | good temperature |
| pleasant | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | unpleasant |
| cool | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | warm |
| unacceptable | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | acceptable |
| uncomfortable temperature | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | comfortable temperature |
| satisfied | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | dissatisfied |

Numbers in cells are the values assigned to the ratings; loadings are as follows: comfortable-uncomfortable, .555; bad temperature-good temperture, .693; pleasant-unpleasant, .628; cool-warm, .579; unacceptable-acceptable, .521; uncomfortable temperature-comfortable temperature, .726; satisfied-dissatisfied, .568; the loading sum = 4.270. Thermal Comfort is determined by the following formula:

$$\text{THERMAL COMFORT (\%)} = [\Sigma\ (\text{Rating x loading}) - 4.270]\ 2.92$$

*Figure 6. Thermal comfort ballot scoring*

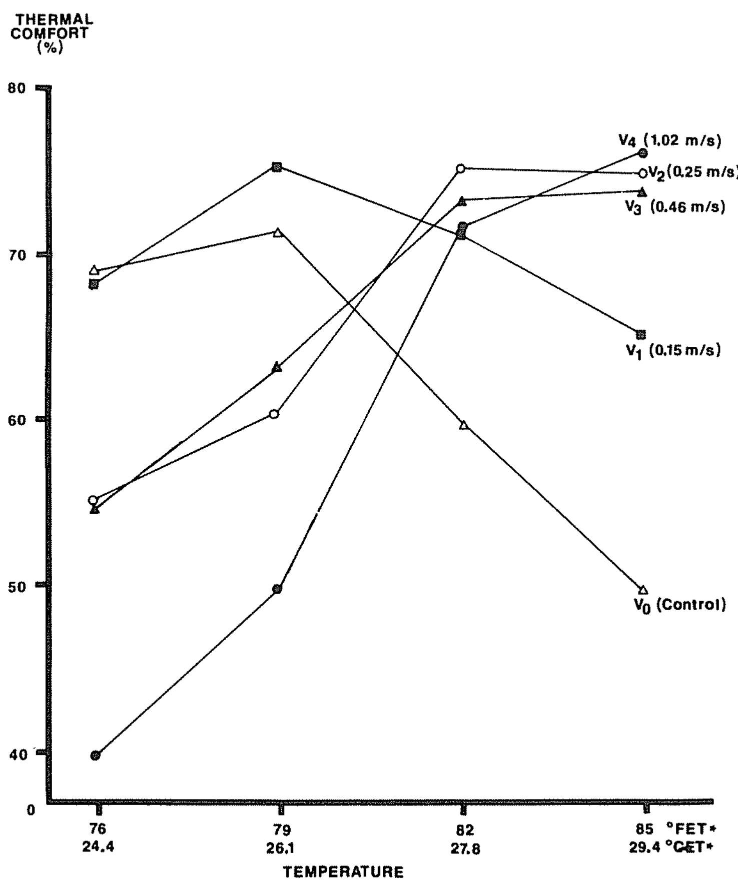



*Figure 7. Mean thermal comfort votes at the four temperatures for each of the five velocities*

| | | |
|---|---|---|
| relaxing | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | tense |
| unacceptable | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | acceptable |
| refreshing | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | unrefreshing |
| favorable | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | unfavorable |
| dislike | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | like |
| damp | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | nondamp |
| appealing | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | unappealing |
| satisfying | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | annoying |
| uncomfortable | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | comfortable |
| unpleasant | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | pleasant |
| repelling | 1 : 2 : 3 : 4 : 5 : 6 : 7 : 8 : 9 | inviting |
| good | 9 : 8 : 7 : 6 : 5 : 4 : 3 : 2 : 1 | bad |

Numbers in the cells are the values assigned to the ratings; loadings are as follows: relaxing-tense, .843; unacceptable-acceptable, .871; refreshing-unrefreshing, .694; favorable-unfavorable, .910; dislike-like, .842; uncomfortable-comfortable, .872; unpleasant-pleasant, .909; repelling-inviting, .731; good-bad, .868; the sum of the loadings = 10.068. Air Plume Quality (APQ) is determined by the following formula:

$$APQ\ (\%) = [\Sigma\ (\text{rating} \times \text{loading}) - 10.068]\ 1.242$$

*Figure 8. Air plume quality ballot scoring*

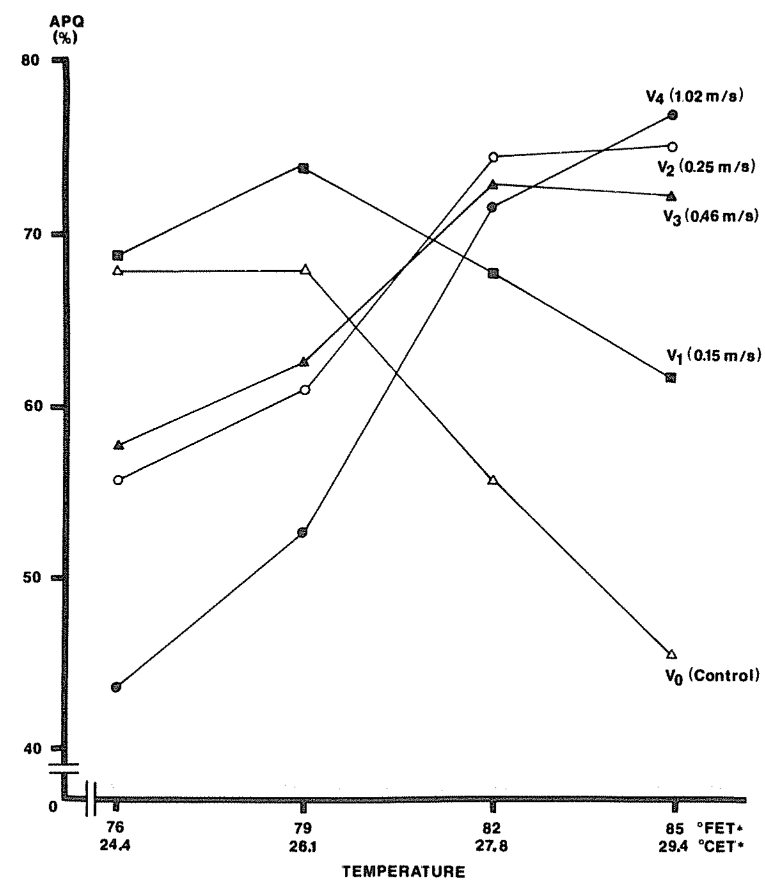



*Figure 9. Mean air plume quality scores at four temperatures for each of the five velocities*

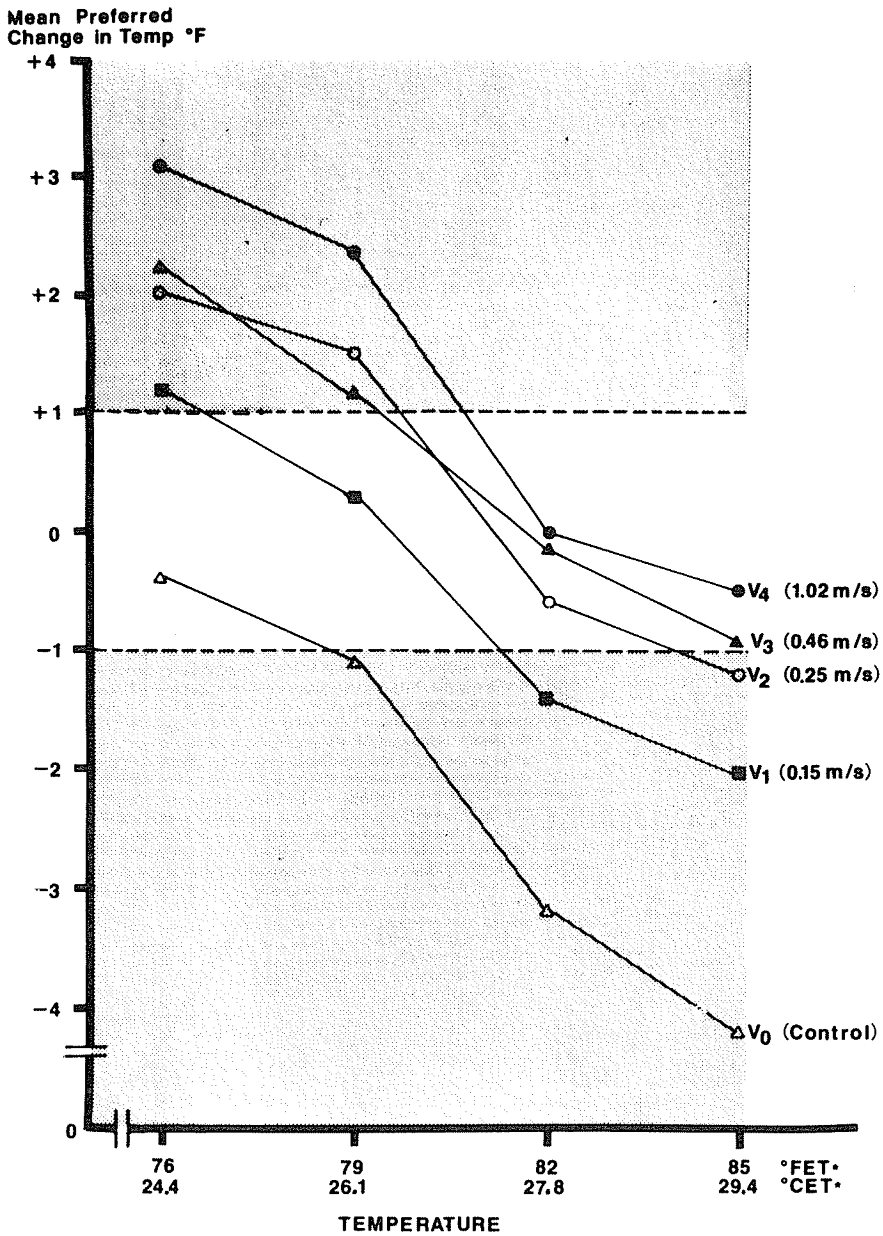



*Figure 10. Mean degrees change in preferred temperature at the four temperature conditions for five velocities*

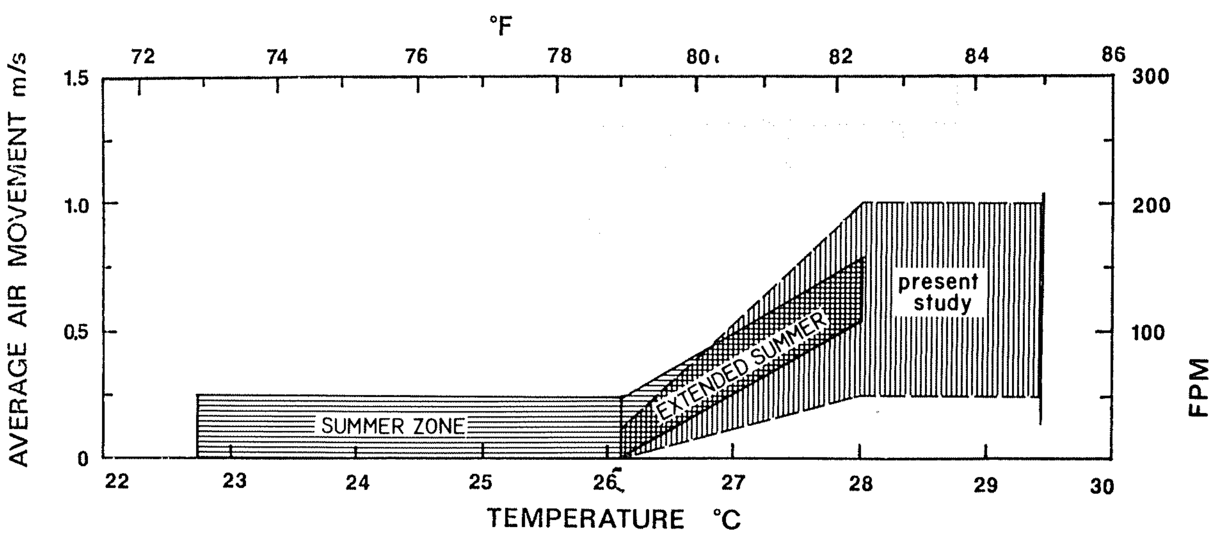



*Figure 11. Added extension of the summer comfort zone based on data from the present study*

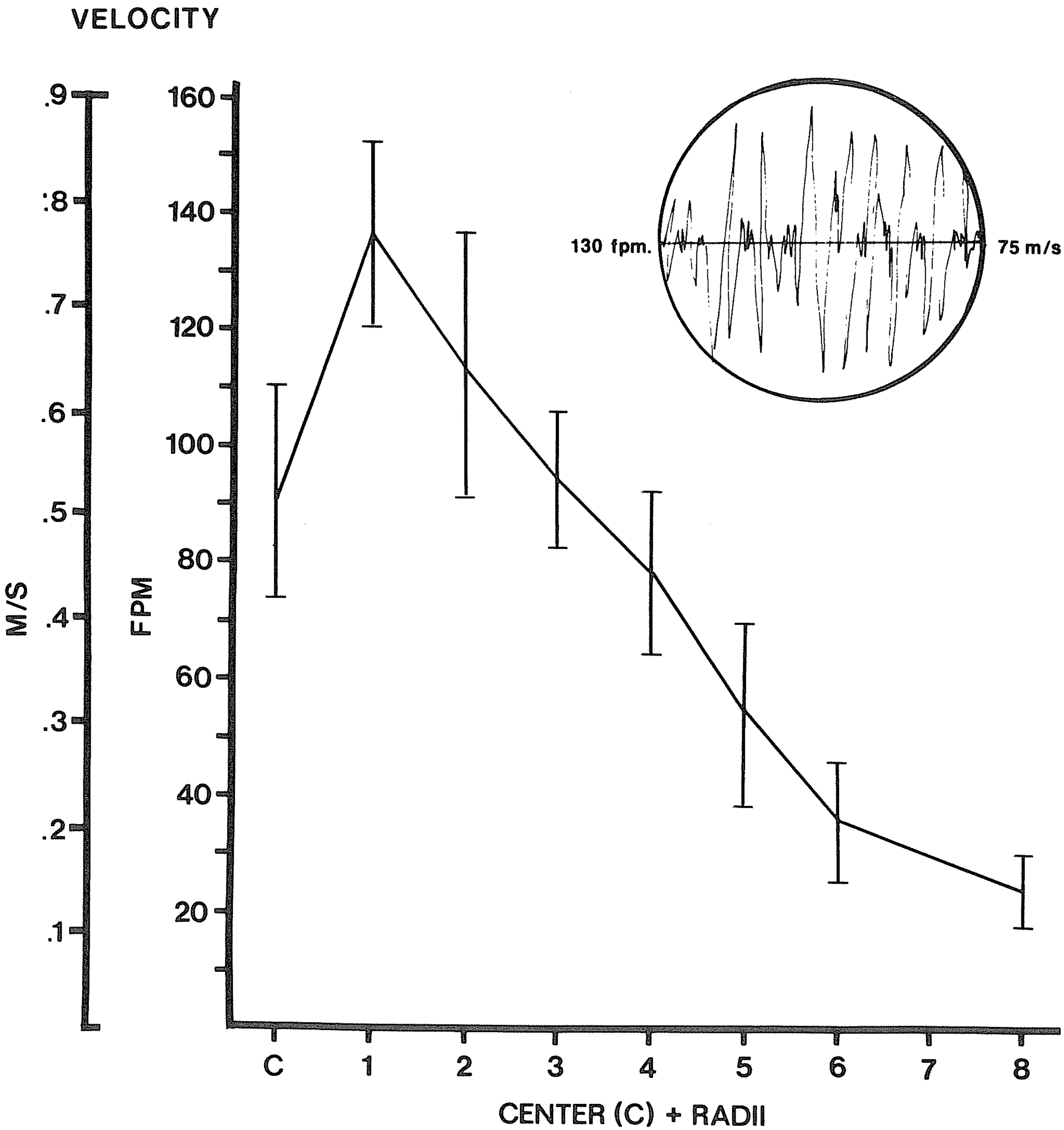



*Figure 12. Mean air velocities ± 2 standard errors at different radii from the center of the ceiling fan*

# DISCUSSION

B.W. Olesen, Lab. of Heating & Air Conditioning, Technical Univ. of Denmark, Lyngby: In the present study, the air velocity distribution has not been very uniform and it may be difficult to estimate a velocity that is representative for the subjects' exposure at the eight different work stations. The authors have decided to let the velocity 1.1 m above floor level represent the exposure on the subjects. But when studying Tab. 1, it is clear that the air velocity in the 1.1 m level is not at all representative for the total exposure. There are great differences between the three levels (0.1, 0.6, and 1.1 m). Instead, it is recommended to use a mean value of the three levels:

$$v_{mean} = (v_{0.1} + v_{1.1})/3$$

The mean values estimated from Tab. 1 are then,

| Location | m/s | Authors' values |
|---|---|---|
| $v_{1a}$, $v_{2a}$ | 0.32 | 0.15 |
| $v_{2a}$, $v_{2b}$ | 0.48 | 0.25 |
| $v_{3a}$, $v_{3b}$ | 0.46 | 0.46 |
| $v_{4a}$, $v_{4b}$ | 0.90 | 1.02 |

This explains why no significant differences were found between conditions 2 and 3. In future studies, it is recommended that you have a specific question about the sensation of local thermal comfort, especially at head and feet level, due to draft.

B.W. Jones: The use of the 43 in. (1.1 m )level to represent the exposure was primarily for convenience in selecting locations for test subject work stations. The process was rather tedious as the introduction of the work station significantly altered the local velocity field. It was essentially a trial and error process and a single level was much easier to work with than multiple levels. It is agreed that the velocities at other levels are also important. However, it is doubtful that a simple mean is the best approach, since air movement over the upper body is likely to have a greater effect on thermal comfort than will air movement over the lower body. Additional research to determine how velocities at different heights affect thermal comfort would be useful.

N.A. Buckley, Broan Mfg. Co., Hartford, WI: Were the reported air velocity measurements taken with an instrument that was directionally sensitive? Why did you choose to jump in velocity from 90 to 200 fpm, particularly in view of the 160 fpm velocity related to Standard 55?

Jones: The anemometer used is relatively insensitive to direction as long as the air does not strike it at more than a $45^{o}$ angle. Smoke tracers were used in conjunction with the anemometer to determine the best orientation at each measurement location. The velocity gradients were very steep between 100 and 200 fpm (0.5-1.0 m/s). We found it difficult to locate areas large enough for test subjects that could be reliably characterized by intermediate velocities.