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Authors

Bauman, Fred
Zhang, H.
Arens, Edward A
et al.

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LOCALIZED COMFORT CONTROL WITH A DESKTOP TASK CONDITIONING SYSTEM: LABORATORY AND FIELD MEASUREMENTS

F.S. Bauman, P.E.
Member ASHRAE

H. Zhang

E.A. Arens, Ph.D.
Member ASHRAE

C.C. Benton

ABSTRACT

This paper reports the results of recently completed laboratory and field measurements investigating the thermal performance of an occupant-controlled desktop task conditioning system. The laboratory experiments were performed in a controlled environment chamber configured to resemble a modern office space with modular workstation furniture and partitions. Velocity and temperature distributions were measured throughout the test chamber for a range of test conditions to investigate the effects of supply volume and direction, supply outlet size, and heat load levels (both uniform and nonuniform) in the space. Comfort model predictions are presented to describe the degree of environmental control and range of occupant comfort levels produced in the workstations. Individual desktop units in side-by-side workstations having significantly different heat load levels could be adjusted to maintain close to comfortable conditions, demonstrating localized comfort control.

The field study was performed in a small demonstration office containing two permanent data acquisition systems capable of monitoring in detail the thermal and energy performance of the office, including four installed desktop task conditioning units. Portable measurement methods were also used to assess the thermal comfort of the workers occupying the office. Initial results from the field study demonstrate the occupant response and use patterns of the desktop system, typical energy use patterns, and the effect of the desktop system on local air velocities and thermal comfort within the workstations.

INTRODUCTION

Recently an increasing amount of attention has been paid to air distribution systems that individually condition the immediate environments of office workers within their workstations. As with task lighting systems, the controls for these systems are partially or entirely decentralized and under the control of the occupants. Typically, the occupant has control over the speed and direction and, in some cases, the temperature of the incoming air supply. The systems have been variously called "task conditioning," "localized thermal distribution," and "personalized air-conditioning" systems. These task conditioning systems provide supply air and (in some cases) radiant heating directly into the workstation, either through a raised-access floor system or in conjunction with the workstation furniture and partitions.

The primary types of task conditioning systems at this time are (in rough chronological order):

- *Floor-Based*: The earliest such systems were widely developed and used in South Africa and Europe (Sodec 1984; Spoormaker 1990; Sodec and Craig 1990). Air is either drawn from an underfloor plenum by local variable-speed fans or forced through the subfloor plenum by the central air handler and delivered to the space through floor-level supply grilles.
- *Desktop-Based:* There are several types of desktop system designs, some with air emerging from grilles on the back of the desk surface (Barker et al. 1987) and others with the air emerging from freestanding directable nozzles at the back of the work surface (Sodec 1984). The desktop system that is the subject of this paper consists of freestanding supply nozzles, but also has additional environmental control features, as described below. Another desktop system supplies air at the desk's front edge directly facing the seated occupant and is only now coming onto the market (Wyon 1992).
- *Partition-Based:* Japanese researchers have characterized the performance of several such systems, with the air emanating from linear diffusers positioned either at mid-panel height (just above desk level) or from a band just below the top of the panel (SHASE 1991). In some of these systems, radiant panels supplement the environmental control, either within the kneespace of the desk or mounted in vertical partitions of the workstation.

Task conditioning systems have the potential to affect many of the ways in which modern offices perform, as described briefly below.

1. The *thermal comfort* of the occupants is perhaps the area of greatest potential improvement in that individual differences or preferences can be accommodated. However, the comfort is obtained by imposing environments that are often thermally asymmetrical, with air

Fred S. Bauman is a research specialist, **Zhang Hui** is a visiting research associate, **Edward A. Arens** is a professor, and **Charles C. Benton** is an associate professor with the Center for Environmental Design Research, Department of Architecture, University of California, Berkeley.

movement and radiation directed on some parts of the body and not on others. In order to optimize such system designs, a new understanding is needed of the thermophysiological effects of such asymmetrical thermal environments.

2. *Ventilation efficiency and air quality* also have the potential to be improved over conventional uniformly mixed systems in that the fresh supply air can be delivered at breathing level and near the occupant.
3. *Energy use* can be raised or lowered depending on system design and operation. In this regard, the most important aspect of installing and operating an energy-efficient task conditioning system is the level of sophistication with which it is integrated with the design and operation of the building's central heating, ventilating, and air-conditioning (HVAC) system. The influence of occupancy sensors can be particularly strong; by shutting off individual workstations when they are unoccupied, substantial savings can be realized. It is also possible that energy savings can be obtained from conditioning only the smaller volume of the occupied workstations. Offsetting these advantages are inefficiencies associated with the small electric motors powering the fans.
4. *Occupant satisfaction and productivity* can also be increased as a result of improved thermal comfort and control over the environment. The financial implications of such improvements have the potential to be very large. However, proving and quantifying such effects is a difficult undertaking.

Should there be substantial improvement in any of the above areas, the task conditioning technology is likely to grow significantly.

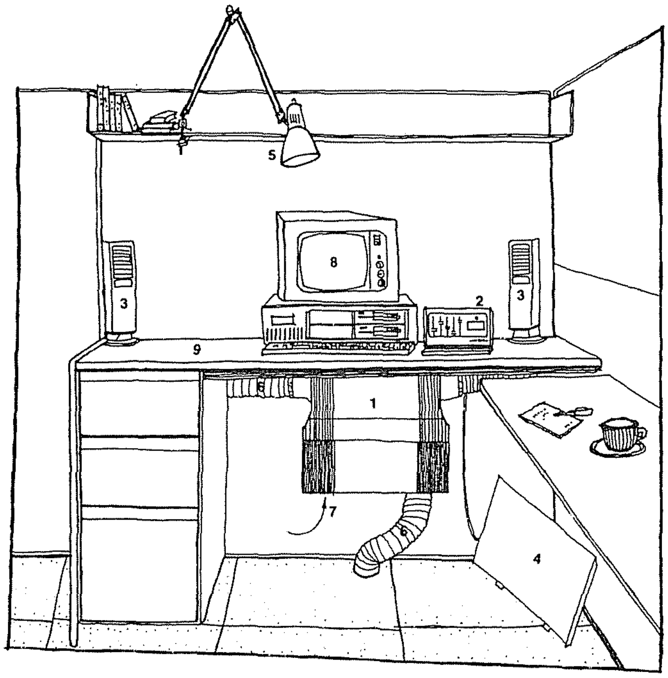
This paper reports the results of recently completed laboratory and field measurements investigating the thermal performance of a desktop task conditioning system. The system is described below.

PRESENT INVESTIGATION

Figure 1 shows a sketch of a typical installation of the desktop task conditioning system selected for investigation in the current study. Each unit uses a self-powered mixing box that is hung in the back or corner of the knee space of the desk and connected by flexible duct to two supply nozzles on the top of the desk. The supply nozzles may be rotated 360° in the horizontal plane and contain outlet vanes that are adjustable $\pm 30°$ in the vertical plane. The mixing box uses a small variable-speed fan to pull supply air from a zero or very low pressure plenum either under the floor (as indicated) or from flexible ducts in the office partitions supplied from the ceiling. A second fan pulls air from the knee space through a mechanical prefilter. Both supply air and recirculated room air are drawn through an electrostatic air filter. The relative fractions of supply air and recirculated air are controlled by dampers on each of these two lines. The main supply line damper is never allowed to close completely, thus ensuring the delivery of fresh ventilation air at all times.

The unit has a desktop control panel containing adjustable sliders that control the speed of the air emerging from the nozzles, its temperature (produced by adjusting the ratio of supply to recirculated air), the temperature of a 200-W radiant heating panel located in the knee space, the dimming of the occupant's task light, and a white noise generator in the unit that issues a rushing sound through the supply nozzles. The control panel also contains a motion-detector-based occupancy sensor that shuts the unit off when the workstation has been unoccupied for a few minutes. The control panel is connected to a microprocessor-based programmable controller contained inside the main unit located under the desk. The controller receives the incoming setpoint information from the control panel and provides the necessary output signals to control the operation of all components.

Each desktop unit is capable of providing approximately 40 to 150 cfm (20 to 70 L/s) of air. Even when its internal fans are turned off, the system is designed to deliver 40 cfm (20 L/s) to satisfy minimum ventilation requirements. In our laboratory, the maximum outlet



1 desktop supply module
2 desktop control panel
3 desktop supply nozzle
4 radiant heating panel
5 task light
6 flexible supply duct
7 recirculated room air
8 personal computer
9 desk

Figure 1 *Desktop task conditioning system.*

velocity measured at the face of the 2.3 by 4 in. (58 by 100 mm) supply vent varied between 6.5 and 24.5 fps (2 and 7.5 m/s) over the same range of airflows described above. In operation, 55°F (13°C) is provided by a variable-air-volume HVAC system, with desk-level outlet temperatures in the range of 65°F (18°C). Under typical operating conditions, the Archimedes number (Ar) of each outlet ranges between 3.5 and 50 $\times$ 10^{-4}, indicating that the force of inertia from the jet dominates the buoyancy forces in the room. (Ar is defined here as Ar = $g \cdot d_0 \cdot \Delta T_0 / [v_0^2 \cdot T_r]$, where g is the acceleration due to gravity [m/s^2], d_0 is the outlet diameter [m], v_0 is the outlet velocity [m/s], ΔT_0 is the temperature difference at jet entry [K], and T_r is the room temperature [K].)

Figure 2 shows a detailed contour plot of the characteristic velocity distribution produced by two desktop supply nozzles. The nozzles are located at the back corners of the desk (5 ft wide by 30 in. deep [1.52 m by 0.76 m]) and deliver air toward a focal point near the center of the front edge of the desk. The velocity contours were measured under the following conditions: (1) intermediate fan speed setting on the control panel ($\approx$ 90 cfm [43 L/s]), (2) horizontal air delivery, (3) isothermal conditions (supply air temperature equals room air temperature), and (4) all measurements taken at the same height as the supply outlets (40 in. [1.02 m] above floor). The results show that the two supply jets are quite focused and provide an average air speed of about 3.3 fps (1 m/s) at the work location in front of the desk.

The first large installation (370 units) of the desktop system was recently completed in a newly designed office building occupied by an insurance company in West Bend, Wisconsin. The building was fully occupied in July 1991 and has provided a rare field research opportunity to study the impact of the desktop task conditioning system on productivity. Using an established computer-based method for measuring the productivity of the insurance company's employees, researchers have tracked the productivity of more than 100 employees before and after they moved into the new building. The study has concluded that the desktop system does have a positive impact on worker productivity, although the magnitude of this impact is still under analysis (Kroner et al. 1992).

During the past two years, laboratory experiments have been completed in our controlled environment chamber (CEC) to investigate the thermal and ventilation performance of the desktop task conditioning system in a partitioned office configuration. Results indicate that the units can be controlled to produce a wide range of thermal conditions, allowing office workers the opportunity to fine-tune the local workstation environment to their individual comfort preferences (Bauman et al. 1991a; Arens et al. 1991). Under some operating conditions, the units were able to provide true task ventilation (i.e., increased ventilation at the location of the occupant), with lower ages of air at the breathing level in the workstation compared to that of

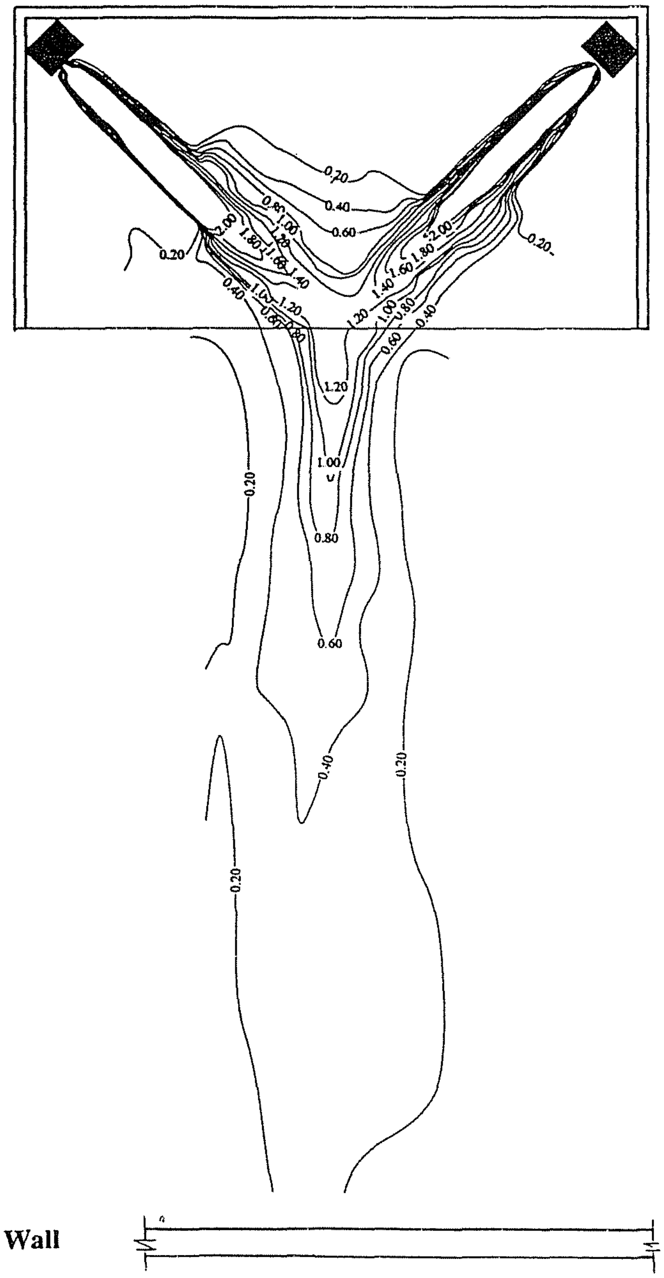


Figure 2 *Velocity distribution (m/s) from desktop system.*

the air leaving the room through the return grille (Faulkner et al. 1993). Preliminary modeling studies of energy use have concluded that installations may use more or less energy compared to a conventional air distribution system, depending primarily on operating strategies (Heinemeier et al. 1991; Seem and Braun 1992). A recent research report describes the results of a survey of the industry perspective on task conditioning systems and also presents recommendations to improve task conditioning system performance (Bauman et al. 1992).

The primary objectives of the laboratory experiments described in the current paper are outlined below.

1. Test the desktop system under a range of operating conditions to identify optimal thermal and ventilation performance characteristics. Previous laboratory experiments had shown that the desktop system was capable of producing uncomfortably high air velocities at its maximum airflow setting. The current series of experiments focused on low to moderate airflow settings to improve the local thermal comfort conditions. In addition, a second, larger supply nozzle design was fabricated and tested for comparison with the original smaller nozzle design. The larger nozzle allowed a higher volume of air to be supplied (improving ventilation performance) while decreasing the corresponding supply inlet velocities (reducing the effects of cool drafts on the occupant). This paper presents the thermal performance and comfort results. The ventilation performance results are described in a related paper (Faulkner et al. 1993).
2. Test the desktop system in an office environment to study its ability to provide localized comfort control. Measurements were made in side-by-side workstations (1) with and without desktop systems and (2) with both uniform and unequal heat loads to demonstrate how the desktop system can be adjusted in response to local comfort requirements.

Recently, results of field measurements of the performance of the desktop system have become available. This field study provides some of the first data on occupant response and use patterns of the desktop system. Initial results from this ongoing study (Bauman et al. 1993) are presented later in the paper.

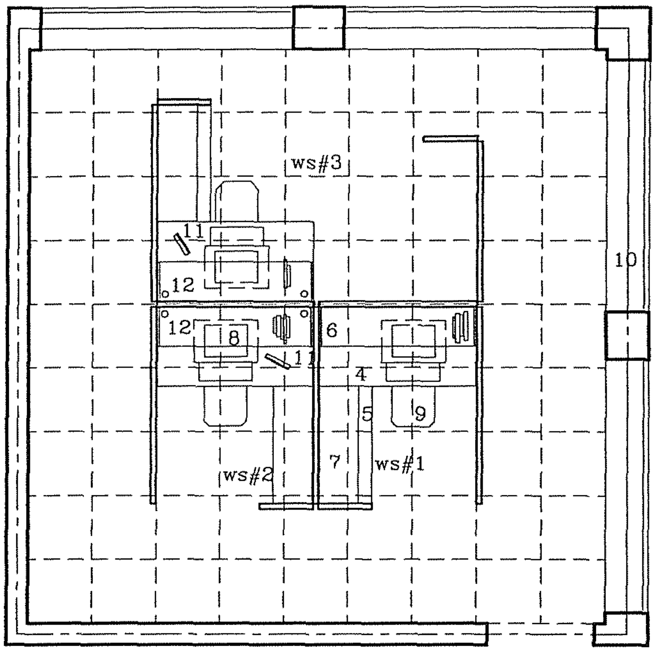
LABORATORY EXPERIMENTAL METHODS

Controlled Environment Chamber

The desktop system was tested in a controlled environment chamber (CEC) measuring 18 ft by 18 ft by 8 ft, 4 in. (5.5 m by 5.5 m by 2.5 m) and located in a university laboratory. The CEC is designed to resemble a modern office space while still allowing a high degree of control over the test chamber's thermal environment (Bauman and Arens 1988). To study the performance of the desktop system in an office environment, a modular workstation configuration, shown in Figure 3, was installed in the CEC. Solid partitions (65 in. [1.65 m] tall) were set up to produce two small 60 in. by 75 in. (1.5 m by 1.9 m) workstations and one double-sized 120 in. by 75 in. (3.0 m by 1.9 m) workstation. The arrangement of the furniture, including desks, side tables, and overhead storage bins, is also shown in the figure. A desktop system was installed in both workstation #2 (WS#2) and workstation #3 (WS#3). For comparison, workstation #1 (WS#1) contained no desktop unit. Conditioned air was provided to each of the two desktop units through separate supply lines ducted through the subfloor plenum to the mixing boxes under the desks. During all tests, the total volume of supply air to the test chamber was delivered through these two desktop units, and air was returned at ceiling level through a single ducted, perforated return grille.

The CEC air distribution system also allows a separately conditioned airflow to be provided within the plenum wall construction of the two exterior chamber walls and between the inner and outer window panes in the area called the annular space. During most tests, airflow through the annular space maintained the temperature of the interior window pane at approximately the average indoor air temperature.

Heat loads were provided to simulate typical office load distributions and densities. Overhead lighting fixtures had a total power rating of 500 W (1,700 Btu/h). Energy balance tests indicated that only a small fraction ($\approx$ 100 W [340 Btu/h]) of the overhead lighting load contributed to the room load. Personal computers containing small internal cooling fans and monitors ($\approx$ 90 W [310 Btu/h] total) were placed on each of the three desktops. Each workstation had a 75-W (256 Btu/h) task light above the desk. A second 75-W light bulb was located at the 1.1-m level near the edge of the desk to simulate the sensible heat load from a typical office worker. The experimenter and computer-based data acquisition system also added approximately 260 W (885



ws#1	– workstation #1	7	– overhead storage
ws#2	– workstation #2	8	– computer
ws#3	– workstation #3	9	– chair
4	– desk	10	– annular space
5	– sidetable	11	– desktop control pannel
6	– bookshelf	12	– desktop supply nozzle

Figure 3 Controlled environment chamber plan.

Btu/h) to the total room load during these tests. During some tests, a higher heat load was produced by placing a 200-W (680-Btu/h) electric radiant heater on the floor under one or more of the desks to represent larger computer processing units.

Instrumentation and Equipment

Detailed air velocity and temperature measurements within the test room were accomplished by using a light-weight sensor rig fabricated of aluminum tubing that allowed a vertical array of sensors to be positioned at desired measurement heights and moved around the room to map out a grid of selected measurement locations. At each location in the room, air velocity and temperature were measured at six heights: 4 in. (0.1 m); 2 ft (0.6 m); 3 ft, 7 in. (1.1 m); 5 ft, 7 in. (1.7 m); 6 ft, 7 in. (2.0 m); and 7 ft, 9 in. (2.35 m). The 0.1-m, 0.6-m, and 1.1-m levels correspond to recommended measurement heights for seated subjects, and the 0.1-m, 1.1-m, and 1.7-m levels correspond to heights recommended for standing subjects, as specified by ASHRAE (1981). Velocities were measured with spherical-element omnidirectional anemometers having a range of 0 to 700 fpm (0 to 3.5 m/s), and temperatures were measured with shielded thermistor temperature probes. All sensors were calibrated prior to testing. The measurement error of the anemometers was estimated to be ± 4 fpm (± 0.02 m/s) over the range 0 to 80 fpm (0 to 0.4 m/s) and ± 8 fpm (± 0.04 m/s) at higher velocities. The measurement error of the temperature sensors was ± 0.2°F (± 0.1°C). Temperature and velocity sensors were sampled 50 times over a 90-second measurement period.

To determine mean radiant temperature within each workstation, an array of three globe temperature sensors was positioned at the front edge of each desk. Constructed using a 1.5-in. (38-mm) diameter table tennis ball, as described by Benton et al. (1990), the globe temperature sensors recorded temperatures at the 0.1-, 0.6-, and 1.1-m heights. Additional details of the measurement equipment, sensor calibration, and data acquisition system are described by Bauman et al. (1991b).

Test Procedures

To investigate the performance of the desktop system under operating conditions that produced lower air velocities, the original system design was modified by fabricating larger supply nozzles to replace the smaller nozzles obtained from the manufacturer. The size of the larger outlet was 3.1 in. by 9.1 in. (78 mm by 230 mm), three times the size of the original smaller outlet. The reduced air velocities produced by the large nozzle in comparison to the small nozzle are demonstrated in Figure 4. The figure shows the measured isothermal centerline velocity profiles from a single large and small nozzle with a supply volume of 26 L/s (55 cfm). Both resemble the characteristic profile of a free jet.

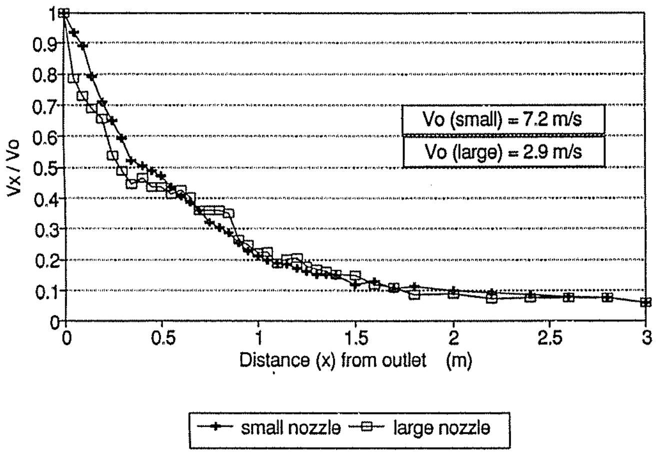


Figure 4 *Centerline velocity profile: Isothermal conditions, supply volume = 55 cfm.*

Since the desktop system is designed to supply air for cooling purposes only, during all experiments the test chamber was controlled to represent the interior zone of an office building. To more clearly demonstrate the local cooling effects of the desktop system, during most tests the average room air temperature was maintained near the upper limit of the ASHRAE-specified comfort zone (ASHRAE 1981) (see Table 1). Two types of tests were carried out, as described below.

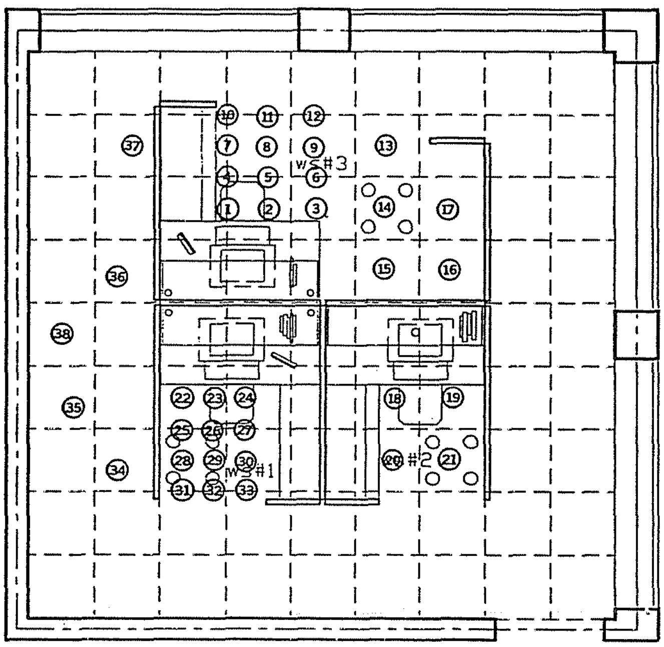
1. Under steady-state conditions, thermal conditions were measured with the previously described sensor rig in all three workstations and at points in the surrounding area. Figure 5 shows the 38 measurement locations used for these tests. As indicated, these measurements focused on determining the conditions within each of the three workstations. A finer grid of points (approximately one-foot intervals) was used in front of the desks in WS#2 and WS#3 to provide greater detail of the velocity and temperature distributions produced by the desktop system.

   To conduct these experiments, the electrical heat sources in the room were turned on in the morning and allowed to warm up the room until the expected average room temperature was reached. After completing the warmup, the mechanical system was turned on; the desktop system supply air volume, temperature, and direction were adjusted to their selected setpoints; and conditions in the room were allowed to further stabilize. During the tests, typical control of the supply air temperature entering the room through the supply nozzles was to within ± 1°F (± 0.5°C) of the desired setpoint. Due to the close proximity of the desktop supply nozzles to the occupant, supply air temperature setpoints were normally close to 65°F (18°C) (see Table 1).
2. To investigate the degree of control and range of comfort conditions that an office worker could produce

TABLE 1
Test Conditions

Test #	Supply Air Volume (cfm) WS#2	Supply Air Volume (cfm) WS#3	Supply Air Temp. (°C) WS#2	Supply Air Temp. (°C) WS#3	Heat Load (W) WS#1	Heat Load (W) WS#2	Heat Load (W) WS#3	Heat Load (W) Total	Room Temp. (°C)	Return Air Temp. (°C)	Return Air Volume (cfm)	Nozzle Position	Nozzle Size
1	47	48	18.4	20.4	272	264	446	1,240	26.7	27.6	48	toward	small
2	79	79	17.0	17.1	197	464	446	1,370	25.6	27.2	84	toward	small
3	81	83	17.0	17.0	197	464	446	1,370	26.3	28.0	84	toward	large
4	105	94	16.6	16.6	197	464	446	1,370	24.1	25.7	136	toward	large
5	88	91	17.9	18.1	197	464	446	1,370	24.1	26.4	88	straight	small
6	84	87	17.8	17.9	197	464	446	1,370	23.8	25.2	85	straight	large
7	83	85	17.8	17.8	75	215	185	735	23.0	24.2	86	toward	large
8	82	82	17.7	17.8	75	215	185	735	22.7	23.9	86	toward	small
9	54	51	18.9	18.8	397	189	446	1,290	27.3	28.8	88	toward	large
10	52	49	19.4	19.2	397	264	446	1,370	26.6	27.9	90	toward	large
11	34	53	20.8	18.6	197	264	446	1,170	25.6	26.7	68	toward	large
12	56	29	19.1	19.5	197	464	446	1,370	27.4	27.8	65	toward	large
13	53	50	18.5	18.2	0	464	446	1,170	26.2	26.9	92	toward	large
14	52	49	19.1	18.8	397	464	446	1,570	28.3	29.1	89	toward	large
15	36	29	19.7	19.2	0	189	171	620	25.1	26.1	59	toward	small
16	53	54	18.9	18.4	397	464	446	1,570	27.2	28.4	80	toward	small
17	54	51	18.3	18.0	397	464	446	1,570	25.8	27.6	92	straight	small
18	52	48	17.9	17.6	0	464	446	1,170	25.5	26.6	93	toward	large
19	49	40	18.1	17.8	0	464	446	1,170	25.6	27.0	90	toward	small
20	*	*	*	*	197	464	*	*	25.2	27.0	*	*	small
21	*	*	*	*	197	464	*	*	25.6	26.9	*	*	large
22	*	*	*	*	0	*	446	*	25.2	26.9	*	*	small
23	*	*	*	*	0	*	446	*	25.7	26.9	*	*	large

* variable



ws#1--workstation #1
ws#2--workstation #2
ws#3--workstation #3

Figure 5 Measurement locations.

in the local workstation, a series of short-term adjustments were made to the desktop system control settings, and the resulting thermal conditions were monitored with the sensor rig placed at a single position in front of the desk to represent the normal working location (position #24 in WS#2 or position #1 in WS#3, as shown in Figure 5). These controllability tests studied changes in airflow setting, supply nozzle direction, and local heat load.

Overall thermal conditions in the chamber were allowed to reach steady state before beginning the test. Table 2 lists the seven control setting combinations that were investigated. During these tests, supply volume per unit was varied from 30 cfm to 90 cfm (14 L/s to 42 L/s), heat load per workstation was changed by turning a 200-W radiant heater on or off, producing a high heat load or a medium heat load, and the supply nozzles were oriented toward the occupant or straight ahead parallel to the sides of the desk. Controllability tests were performed with both small and large nozzles in WS#2 and WS#3. As measurements were being made in one workstation, an equivalent amount of air was also supplied by the desktop system in the adjacent workstation to help maintain overall thermal conditions in the chamber close to equilibrium. Each setting listed

TABLE 2
Desktop System Controllability Test: Control Settings in WS#3

No.	Supply Volume (cfm)	Heat Load (W)	Nozzle Direction
1	30	446	toward
2	90	446	toward
3	50	446	toward
4	50	446	straight
5	50	246	straight
6	50	246	toward
7	30	246	toward

in Table 2 was maintained for 16 minutes before changing the system to the next control setting. Sensor readings were recorded at exactly two-minute intervals throughout the test.

Table 1 presents a complete list of the average conditions maintained during each of the 23 tests of the desktop system. The tests investigated the following ranges of test parameters: (1) supply volume per unit from 29 cfm to 105 cfm (14 L/s to 50 L/s); (2) supply air temperature from 16.6°C to 20.8°C (62°F to 69°F); (3) heat load per workstation from 0 to 460 W (0 to 1,570 Btu/h); (4) average heat load density from 24 W/m^2 to 51 W/m^2 (7.8 Btu/h·ft^2 to 16.8 Btu/h·ft^2); (5) uniform and nonuniform heat load distributions; (6) average room temperature from 22.7°C to 28.3°C (73°F to 83°F); (6) small and large supply nozzles; and (7) nozzles pointed toward the occupant and straight ahead parallel to the sides of the desk.

The above test procedures are similar to those used and described by Bauman et al. (1991b) and Arens et al. (1991).

LABORATORY RESULTS

Due to the large amount of experimental data, a limited number of tests have been selected from Table 1 for presentation and discussion. The emphasis of the data presented here is on the local thermal conditions within each workstation. For brevity, average conditions at a given height in a workstation are defined as the velocity or temperature calculated by averaging the measured values from the measurement locations in front of the desk. Referring to Figure 5, average conditions in WS#1 are based on the four points 18-21; WS#2 is based on the twelve points 22-33; and WS#3 is based on the twelve points 1-12.

Nozzle Size and Supply Volume

Figures 6a and 6b present average velocity and temperature results within each workstation for tests 2, 3, 15, 18, and 19. Due to instability in the anemometer located at the

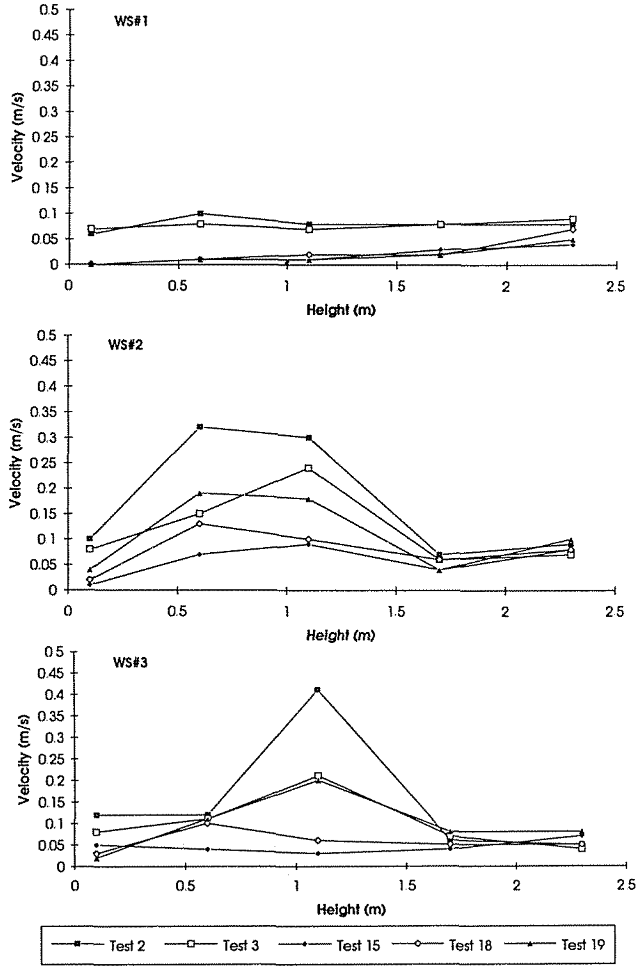


Figure 6a Nozzle size and supply volume: Average velocities.

2.0-m height, velocity measurements are not reported at that height. The selected tests cover both small and large nozzle sizes and supply volumes in the range of 29 to 83 cfm per workstation, representing minimum to mid-range design flow rates for the desktop system. The observations are as follows:

1. As expected, the desktop systems in WS#2 and WS#3 produce higher average velocities and lower average temperatures within the occupied zone for a seated office worker (0.1 m to 1.1 m) during all tests compared to WS#1 (without a desktop system).
2. For tests with the same supply volume, the small nozzles always produce higher velocities in WS#2 and WS#3 compared to the large nozzles. The maximum single-point velocities (measured at the 1.1-m height focal point of the two supply jets in front of the desk) were 1.37 m/s (4.5 fps) in WS#2 and 1.46 m/s (4.8 fps) in WS#3 during test 2 with the small nozzles and highest supply volume.
3. Within the occupied zone for a seated office worker, two characteristic velocity distributions are found in

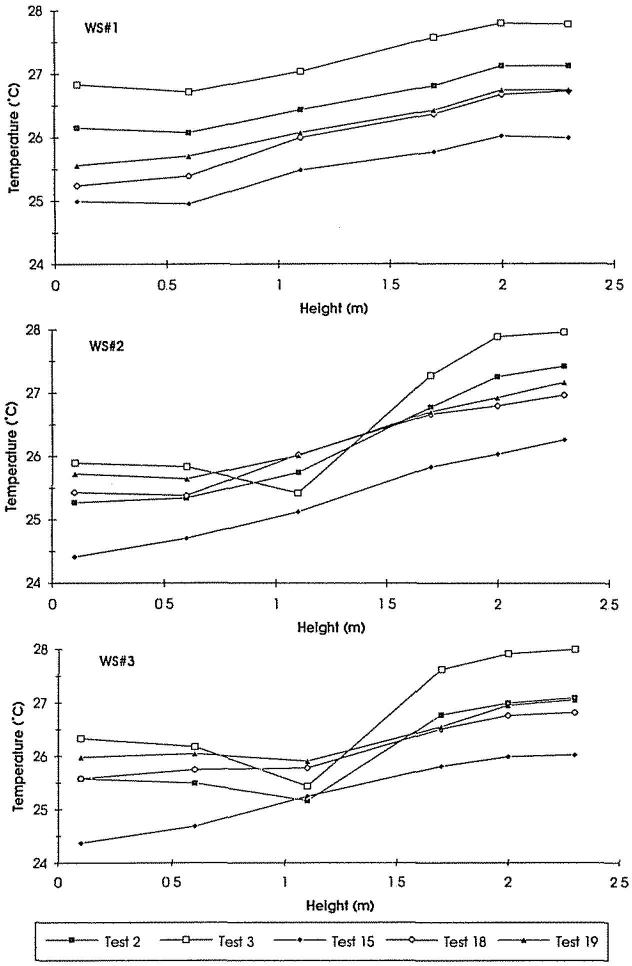


Figure 6b *Nozzle size and supply volume: Average temperatures.*

WS#2 and WS#3. In WS#3, the larger workstation, the maximum velocities occur primarily at the 1.1-m height, the level at which the supply nozzles are focused. In WS#2, the smaller workstation, the single-point data indicated that the supply air jets from the desktop system were focused at a slightly lower trajectory and that the partitions also tended to contain and recirculate the supply air, causing average velocities at the 0.6-m height to be similar in magnitude to velocities at the 1.1-m height. This result was particularly true for the small nozzles.

4. During tests 2 and 3, the desktop system was able to maintain average temperatures at the 0.1- to 1.1-m heights in WS#2 and WS#3 from 0.5°C to 1.6°C below the corresponding average temperatures in WS#1. This result was obtained despite the fact that the heat load in WS#1 was less than half that in WS#2 and WS#3. At the higher supply volume of these two tests, the significant effect of the desktop system's supply air jets on temperatures at the 1.1-m height is quite evident.
5. In WS#1, without a desktop system, velocities for all tests are quite low, although there is a noticeable increase for tests 2 and 3 at the highest supply volume.
6. Average temperatures at the 1.7-m height and above were quite similar in all three workstations during all tests.

Nozzle Size and Controllability

Potentially the most significant performance characteristic of desktop systems is their controllability by individual office workers. Tests were performed to determine the range of thermal conditions that could be achieved in a relatively short length of time (16 minutes) by simply adjusting the desktop system control settings (see Table 2). Tests of this type have been previously reported by Arens et al. (1991) and have demonstrated that the desktop system can be used to control thermal conditions over a wide range. In the current series of tests, the comfort controllability of the original small-nozzle design is compared with that of the large-nozzle desktop system under the same test conditions.

The ISO (1984) computer program for calculating PMV (predicted mean vote) and PPD (predicted percent dissatisfied), based on Fanger's PMV model (Fanger 1970), was used to evaluate the comfort conditions produced by each combination of control settings listed in Table 2. Measurements recorded at the end of each 16-minute test period were used as input to the model along with assumed values of 50% relative humidity, 0.5 clo, and 1.2 met. The model was run for two sets of data for each test condition: (1) data recorded at the 1.1-m level, representing the head/neck region of a seated person, and (2) data averaged for the 0.1-, 0.6-, and 1.1-m levels, representing a whole-body average for a seated person. While the effect of localized cooling of the head/neck region, the area most sensitive to draft discomfort, on whole-body comfort is not well quantified at this time, the desktop system (with its head-level supply air jets) is expected to produce comfort conditions that fall between the limits calculated from the above two data sets.

Figures 7a and 7b show the predicted PMV values from the controllability tests in WS#3 for both small and large nozzles (tests 20 and 21 in Table 1). The seven combinations of control settings are listed in Table 2. Figure 7a shows results based on 1.1-m data and Figure 7b shows results based on whole-body average data. The observations are as follows:

1. Predicted PMV values for both small and large nozzles are seen to cover a wide range of comfort conditions, particularly when based on the data at the 1.1-m height. As expected, the coolest conditions occur for control setting no. 2 with the highest supply volume. The warmest conditions occur at high heat load with either minimum supply volume (setting no. 1) or when

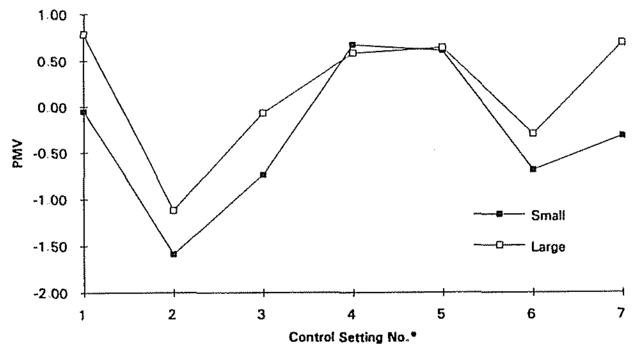


Figure 7a *Comfort controllability: 1.1-m data in WS#3.*

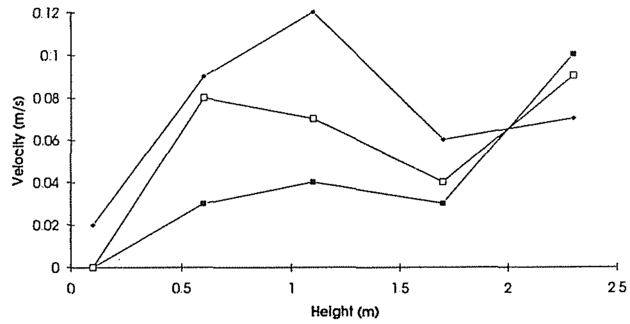


Figure 8a *Localized comfort control: Average velocities for test 11.*

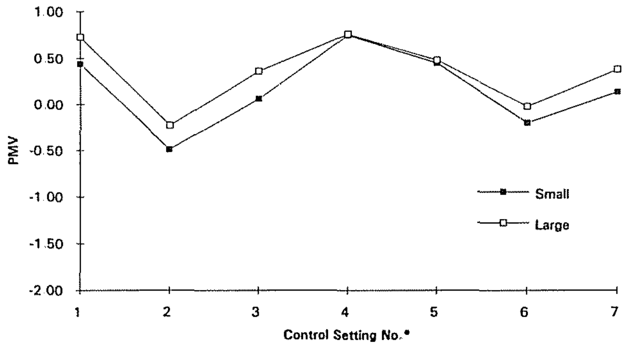


Figure 7b *Comfort controllability: Average data in WS#3.*

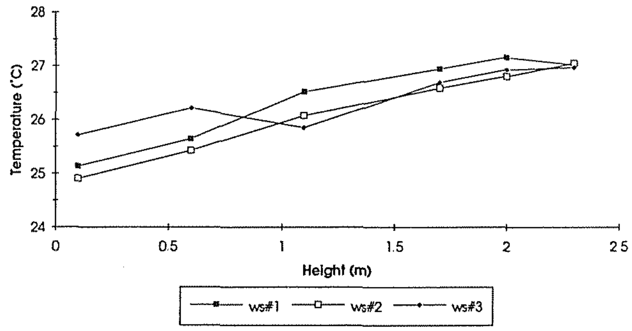


Figure 8b *Localized comfort control: Average temperatures for test 11.*

the nozzles are not directed toward the measurement location (setting no. 4).

2. When the nozzles are directed toward the work location in front of the desk (setting nos. 1, 2, 3, 6, and 7), the higher velocities generated by the small nozzles always produce cooler comfort conditions compared to those produced by the large nozzles. For these five control settings, PMV values based on 1.1-m data for the small nozzles are 0.4 to 1.0 below the corresponding PMV values for the large nozzles (Figure 7a). Similarly, PMV values based on average data for the small nozzles are 0.2 to 0.3 below those for the large nozzles (Figure 7b).
3. When the nozzles are directed straight ahead (setting nos. 4 and 5), both nozzle sizes produce the same comfort conditions at the work location in front of the desk.

Localized Comfort Control

In the same way that the desktop system can be adjusted by office workers to satisfy their individual comfort preferences, it can also be used to satisfy localized cooling requirements created by nonuniform heat loads that are commonly found in office environments. Figures 8a and 8b present average velocity and temperature results within each workstation for test 11. During this test, WS#3 had a relatively high heat load that was nearly 150 W (510 Btu/h) greater than that in WS#2 and more than 200 W (680 (Btu/h) greater than that in WS#1. To demonstrate the localized control of thermal conditions in side-by-side workstations, two desktop systems with large nozzles were adjusted to deliver 53 cfm (25 L/s) of 18.6°C (65°F) temperature air to WS#3 and 34 cfm (16 L/s) of 20.8°C (69°F) temperature air to WS#2. In both workstations, the supply air was directed toward the work location in front of the desk. The observations are as follows:

1. Even at the low supply volumes of test 11, representing minimum, or close to minimum, airflow conditions for the desktop system, average velocities are increased in both WS#2 and WS#3 compared to WS#1. The maximum single-point velocity at the focal point of the two supply nozzles was 0.51 m/s (100 fpm) in WS#3 with the higher supply volume and 0.33 m/s (65 fpm) in WS#2 with the lower supply volume.
2. In Figure 8b, the highest temperatures at the 0.1-m and 0.6-m heights occur in WS#3 due to the high heat load. This trend is reversed at the 1.1-m height because of the cooling effect of the higher velocity desktop level airflow. Temperatures in WS#2 are maintained slightly below those in WS#1 at all heights and very close to those in WS#3 at the 1.1 m height and above.

The localized comfort control provided by the desktop system during test 11 was further analyzed by comparing the calculated PMV and PPD values for each workstation. Table 3 lists the comfort model (ISO 1984) predictions for measurements taken at the work location in front of the desk in each workstation. Results are shown for both 1.1-m data and data averaged for the 0.1-, 0.6-, and 1.1-m levels, as described above, representing the limits within which whole-body comfort conditions are expected to fall.

As expected, WS#1, without a desktop system, experienced the warmest conditions. Despite the unequal heat loads in WS#2 and WS#3, the desktop systems are shown to be able to control comfort conditions over a range that is predicted to include very nearly thermal neutrality (PMV = 0) in both workstations. Due to the higher desktop system air supply and higher heat load in WS#3, the range of PMV values in WS#3 (−0.51 to 0.29) is greater than the range in WS#2 (−0.29 to −0.05).

To demonstrate the spatial variability of comfort conditions produced by the desktop system, Figure 9 presents a contour plot of PMV values from conditions measured during test 2. The calculations are based on data averaged for the 0.1-, 0.6-, and 1.1-m levels at individual measurement locations throughout the test chamber. During test 2, approximately 80 cfm (38 L/s) was supplied through small nozzles turned toward the work location in both WS#2 and WS#3. A high heat load density of 46 W/m^2 (15.2 Btu/h·ft^2) was used, producing an average room temperature of 25.6°C (78°F), near the upper end of the ASHRAE-specified comfort zone (ASHRAE 1981).

The significant impact of the desktop system's air supply on local comfort conditions within WS#2 and WS#3 is clearly visible. At the focal point of the supply nozzles in front of the desk, PMV values are less than −0.3. Despite the warm average room temperature during this test, the desktop system is able to maintain PMV values of less than or equal to 0.3 over the approximately 4 ft (1.2 m) square area in front of each desk. Comfort conditions throughout the surrounding area of the test chamber (region d) are near the upper limit of the acceptable PMV range. Conditions in WS#1 with concentrated heat loads, but without a desktop system, are even warmer.

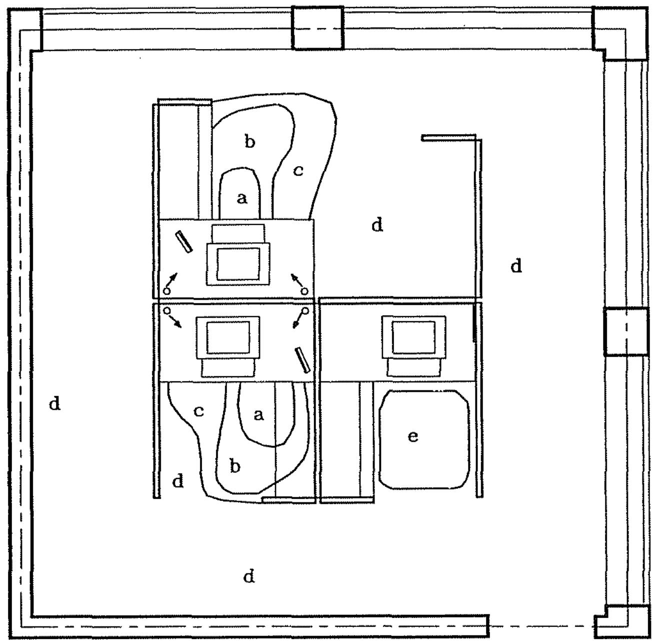


Figure 9 *Thermal comfort contours from desktop system.*

FIELD EXPERIMENTAL METHODS

In spring 1992, a small 1,600-ft^2 demonstration office was set up by a California utility company to allow advanced office technologies to be installed, tested, and demonstrated. As shown in Figure 10, within the eight-workstation office, the desktop system was installed in a cluster of four partitioned workstations and, for comparison, a second cluster consisted of four identical workstations without desktop systems. The partition arrangement in each cluster forms a central core area that proved to be convenient for installing the desktop system air supply duct and the workstation monitoring networks. This central core was extended to the ceiling, forming a hollow column through which the air supply duct was run down from the ceiling to serve the four desktop units. The central column above the conventional workstation cluster looked identical but contained no ductwork.

The office was heavily instrumented and two permanent data acquisition systems were installed, allowing thermal and energy performance to be monitored in detail. With the

TABLE 3
PMV and PPD Results for Test 11

	WS#1		WS#2		WS#3	
	1.1 m	Average	1.1 m	Average	1.1 m	Average
PMV	0.73	0.41	-0.29	-0.05	-0.51	0.29
PPD (%)	16.1	8.5	6.7	5.0	10.3	6.7

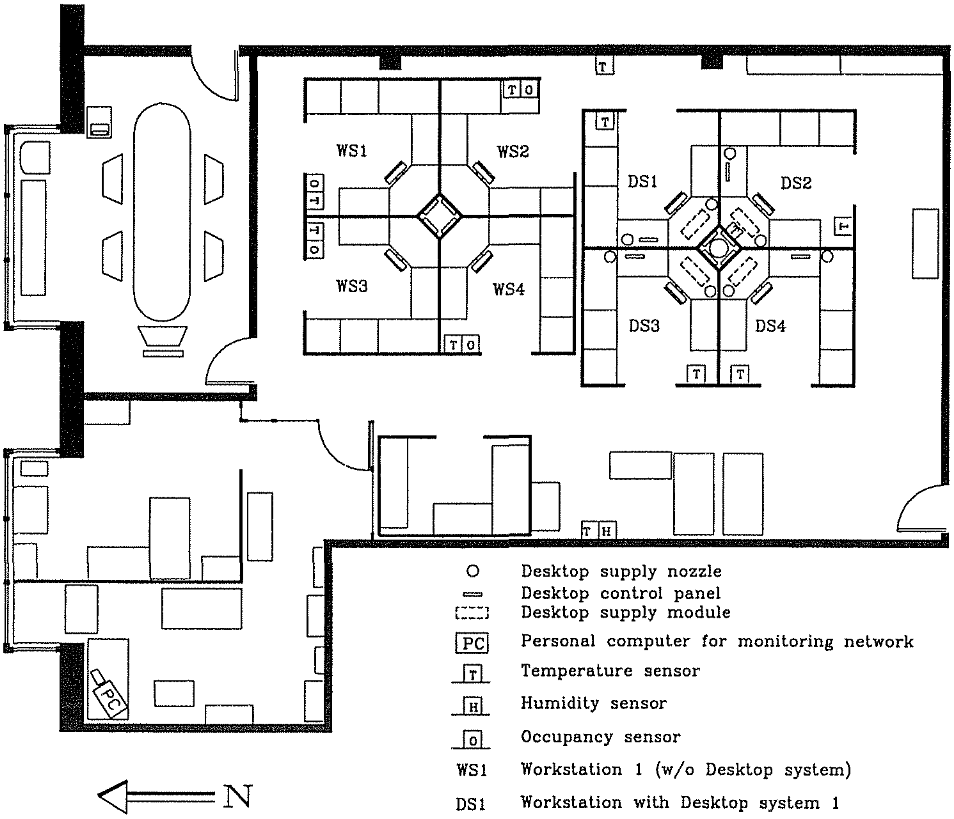


Figure 10 Demonstration office floor plan.

selection of the desktop system as the initial advanced technology to be evaluated in the demonstration office, we were presented with a unique opportunity to directly monitor the occupant use patterns and performance characteristics of individually controlled desktop units by utilizing a network communication capability provided by these units. Each desktop unit contains a microprocessor-based programmable controller. The controller receives the incoming setpoint information from the desktop control panel and provides the necessary output signals to control the operation of all system components. The controller utilizes a communication link allowing multiple controllers to be networked together and to be connected to a single host microcomputer.

Within each desktop unit, the controller allows the status of several control parameters and two temperature sensors to be monitored. These include (1) discharge air temperature setpoint, (2) radiant panel setpoint, (3) fan speed setpoint, (4) task light setpoint, (5) occupancy sensor status, (6) discharge air temperature, and (7) workstation air temperature. By adding three more controllers, the data acquisition network was expanded to monitor (1) room air temperatures and humidity, (2) air temperatures and occupancy status in the four conventional workstations, and (3) supply and return air conditions (temperature, humidity, and volume) in the office's HVAC system. Figure 10 shows the locations in the office where sensors were attached to partitions and walls to monitor room air temperatures, occupancy status, and humidity.

The desktop system's monitoring network is controlled by software executed from the host computer. The software enables data to be collected and stored, and collected data from the files to be displayed on the computer monitor using color graphic images. Full details of the monitoring network are described by Bauman et al. (1993).

Portable measurement methods were used to assess the thermal comfort of the eight office workers occupying the office. A second-generation physical measurement system was developed in 1991 and used for the current study. The system design was based on an earlier version that had been developed and used for a field study of thermal comfort in 10 San Francisco Bay area office buildings (Schiller et al. 1988; Benton et al. 1990). The new thermal measurement cart takes advantage of recent technological developments in data acquisition hardware and transducers by packaging these in a frame smaller and more maneuverable than the original cart design. The new cart, like its predecessor, collects a complete set of detailed measurements characterizing the local thermal environment using an automated approach. We collected data for air temperature, relative humidity, air velocity, globe temperature, and radiant asymmetry to satisfy the requirements of *ASHRAE Standard*

55-1981 (ASHRAE 1981) and ISO Standard 7726 (ISO 1985).

The portable measurement system also included a laptop-based subjective survey that was administered to the office worker before each workstation visit. The survey asks questions relating to current thermal sensation, current satisfaction with the environment, recently used methods to make changes to the local thermal environment (e.g., turn on fan, turn on heater), current emotions, current clothing, and recent activity levels.

The field measurement protocol closely followed that developed in our previous thermal comfort field work. While a physical measurement is collected at a particular workstation, the field worker looks for potentially available subjects to take the subjective survey. Having found the next subject, the field worker enters the subject's identification number into the laptop computer and places it on the subject's desk. While the subject takes the survey, the field worker retrieves the cart from the previous workstation and moves it to the vicinity of the subject taking the survey. When the survey is completed, the field worker removes both the laptop computer from the subject's desk and the subject's chair from in front of the desk. The cart is then placed in the location and orientation of the subject's chair and the measurement period is initiated by flipping a switch on the cart. During the next five minutes, the cart collects physical data at the workstation while the field worker searches for the next available subject. For additional details, see Benton et al. (1990).

FIELD MEASUREMENT RESULTS

The first thermal comfort study of the demonstration office was performed from April 30 to May 1, 1992, and consisted of 39 workstation visits (approximately five visits to each of the eight subjects). Results of the physical measurements found quite uniform temperatures throughout the office. Within the occupied zone (0.1 m to 1.1 m), the overall average air temperature was 23.0°C with a maximum of 23.6°C and a minimum of 22.2°C.

To study the effect of the desktop system on local air velocities, velocity data recorded by the portable measurement system were grouped by workstations with and without desktop systems and compared for each of the three measurement heights (0.1 m, 0.6 m, and 1.1 m). While average velocities at the 0.1-m level were very similar (0.06 m/s [12 fpm] in workstations with a desktop system vs. 0.07 m/s [14 fpm] in workstations without), the average velocity in workstations with desktop systems was noticeably higher at the 1.1-m height (0.18 m/s [36 fpm]) and somewhat higher at the 0.6-m height (0.12 m/s [24 fpm]) compared to the average velocity in workstations without desktop systems (0.10 m/s [20 fpm] and 0.08 m/s [16 fpm], respectively). Looking at all single-point data from the 0.6- and 1.1-m heights (78 total measurements), there were 17 occurrences that exceeded the ASHRAE winter comfort limit of 0.15 m/s (30 fpm) and five occurrences that exceeded the ASHRAE summer comfort limit of 0.25 m/s (50 fpm) (ASHRAE 1981). All 17 of the measured velocities exceeding 0.15 m/s occurred in workstations containing desktop systems. This demonstrates that the desk-mounted supply nozzles have a significant impact on air movement at heights near desk level. Of note is that in the recently revised version of ASHRAE Standard 55 (ASHRAE 1992), air velocities greater than the previously specified limits will be allowed if the occupant has direct control over the local airflow. This change in the comfort standard is intended to accommodate such occupant-controlled systems as the desktop task conditioning system.

Although the small number of subjects (eight) in this study prevents statistically significant conclusions from being drawn, a few comparisons between subjective survey responses for the two groups of subjects (one with desktop systems, one without) are worth mentioning. Average thermal sensation results based on the ASHRAE Thermal Sensation Scale (seven-point scale with -3 = cold, 0 = neutral, and $+3$ = hot) found that subjects with desktop systems were very nearly neutral (0.02) compared to the slightly warmer thermal sensation (0.44) for subjects without desktop systems. A six-point general comfort scale (1 = very uncomfortable, 2 = moderately uncomfortable, 3 = slightly uncomfortable, 4 = slightly comfortable, 5 = moderately comfortable, and 6 = very comfortable) found an average result of 5.3 for those with a desktop system compared to 4.6 for those without.

The Fobelets and Gagge (1988) two-node comfort model was used to calculate the standard comfort indices (PMV, PMV*, DISC, TSENS, ET*, SET*, and HSI) for each workstation visit. The average ET* predicted by the model for all workstation visits was 23.0°C. When compared to the comfort zone described in *ANSI/ASHRAE 55-1981*, 31% of the workstation visits produced ET* values below the minimum specified limits for summer conditions. Given the warm inland climate around the demonstration office building, located east of San Francisco, this result suggests that the office was slightly overcooled at the time of this field study (early May). Observations by field researchers and comments from the study participants also supported the assessment that the office was cooler than necessary.

The potential for improved local comfort and ventilation using the individually controlled desktop system should allow conditioning requirements in parts of the surrounding office to be relaxed (e.g., thermostat setpoint could be raised). Due to the cooler ambient conditions maintained in the demonstration office, there was little need for an office worker to fine-tune the environment except under extreme conditions (e.g., increased activity level). Results from the desktop system monitoring network confirmed this finding, as the desktop control panels were used only sparingly.

Figure 11 shows an example of the occupant use pattern of the desktop system in one workstation between

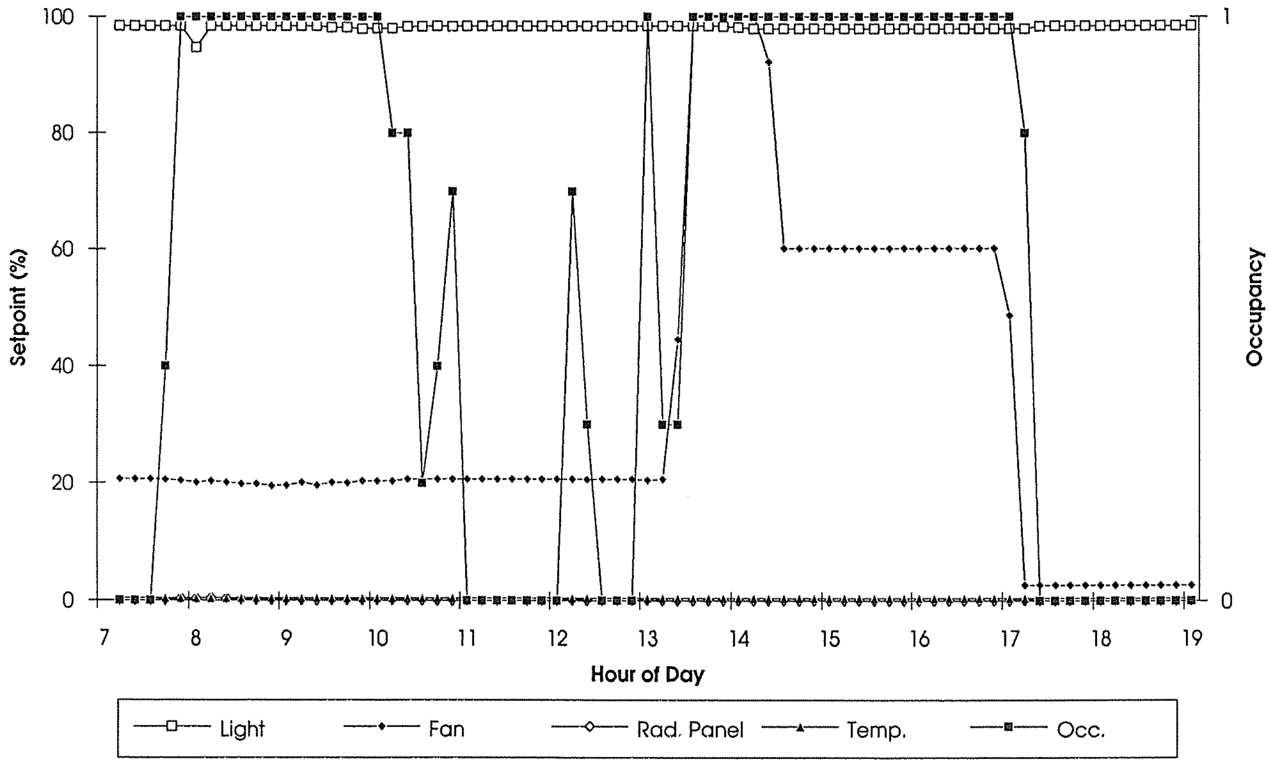


Figure 11 *Desktop system setpoints and occupancy: April 30, 1992.*

the hours of 7 a.m. and 7 p.m. on April 30, 1992. In the figure, setpoint positions (0% to 100%, where 0% is the minimum position of the lever on the control panel and 100% is the maximum position) for the task light, fan, radiant panel, and discharge temperature are shown on the left axis, while occupancy (0 = unoccupied, 1 = occupied) is shown on the right axis. Ten-minute average data are presented, which in this example show a greater frequency of use of the desktop system's controls than was commonly observed. In Figure 11, the light is turned on 100% all day long. The fan setpoint stays at 20% during the morning and jumps to 100% when the occupant returns from the lunch hour having played basketball. After another hour, the fan is turned down to 60% until the end of the day. Radiant panel and temperature controls are unused all day long. The short occupancy peaks during the later morning and noontime hours may in fact be due to visits to the workstation by office workers other than the occupant.

Monitoring of the total electrical plug load from all four desktop systems allowed their energy use patterns to be investigated. To illustrate the effect of the occupancy sensor on energy use, Figure 12 presents the total power in watts used by all four desktop systems compared to the total occupancy in the four workstations with desktop systems. Ten-minute average data are shown between the hours of 7 a.m. and 7 p.m. As expected, the pattern of energy use closely follows the occupancy pattern, demonstrating the operation of the occupancy-sensor-controlled desktop unit.

A quick calculation shows the energy-saving capability of such a system. Between the hours of 8 a.m. and 6 p.m., the average desktop system power use for the four units is 337 W when controlled by the occupancy sensors. This represents a 31% savings over the maximum power use of 488 W during the same period, an estimate of the worst-case scenario for energy use. Total building energy performance will depend on how well the desktop task conditioning system is integrated with the central HVAC system.

CONCLUSIONS

Laboratory and field measurements were made to investigate the performance of a desktop task conditioning system. The laboratory experiments were carried out in a controlled environment chamber configured to resemble a modern office space with modular partitioned workstations. Detailed tests were conducted to study the effects of supply nozzle size, supply volume, supply direction, supply temperature, heat load density and distribution, and average room temperature. The field measurements were performed in a small demonstration office space with eight occupants—four in workstations with desktop systems and four in workstations without them. Measurements were made to assess the thermal characteristics of the office, as well as the occupants' thermal comfort and use patterns of the desktop system. The major findings are summarized below.

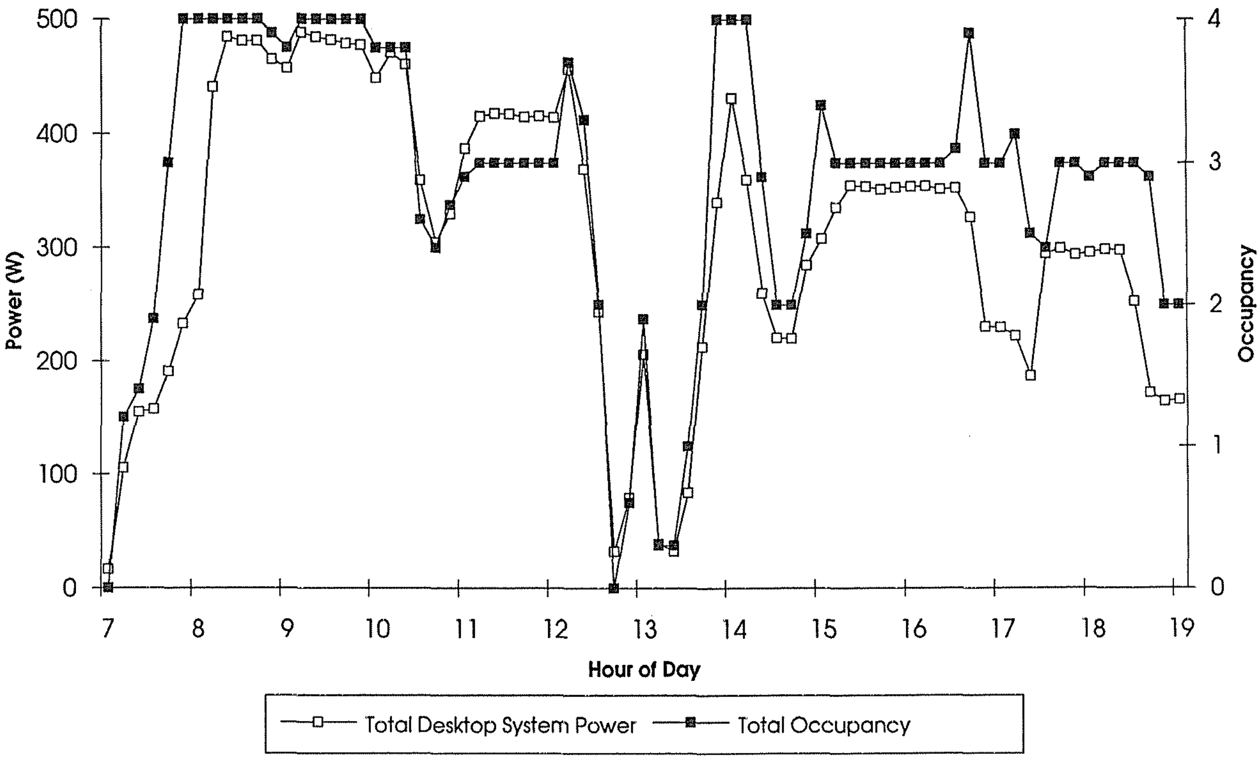


Figure 12 Desktop system power and occupancy: April 30, 1992.

1. During the laboratory experiments, primarily by adjusting the volume and trajectory of the supply air from the desktop system, local thermal conditions could be controlled over a wide range.
2. Even at relatively low air supply rates, individual desktop units in adjacent workstations having significantly different heat load levels could be fine-tuned to maintain nearly comfortable conditions, demonstrating localized comfort control.
3. Under warm average room air temperature conditions, the local cooling effect of the desktop system was able to maintain average temperatures in the occupied zone (0.1-m to 1.1-m heights) of one workstation from 0.5°C to 1.5°C (1°F to 3°F) below the corresponding temperatures in an adjacent workstation without a desktop system. This result was achieved with only a moderate supply air volume (approximately 50% of maximum).
4. The desktop system was shown to deliver lower velocity air using a larger supply nozzle compared to the original smaller nozzle design at the same supply volume. The larger nozzles reduced the potential for draft discomfort while maintaining improved (task) ventilation performance at moderate to high air supply volumes (see Faulkner et al. 1993).
5. In the field study, noticeably higher average velocities were measured at the 1.1-m and 0.6-m heights in workstations with desktop systems compared to workstations without such systems.
6. Due to the comfortably cool ambient conditions that were maintained in the field study office, the office workers adjusted their desktop control panels only occasionally. They had little need to fine-tune their local environment, except under rare conditions.
7. The pattern of energy use for the desktop system closely followed the occupancy pattern, demonstrating the operation of the occupancy-sensor-controlled desktop unit. By turning off the desktop system whenever the workstation is unoccupied, the occupancy sensor has the potential to significantly reduce the amount of energy used by the task conditioning system.

Future work on task conditioning systems is needed to address the following issues:

1. Testing of other task conditioning systems to provide more performance data to the building engineering community.
2. Field monitoring projects of operational task conditioning systems to demonstrate occupant response, thermal comfort, indoor air quality, and energy use implications under a wider range of environmental conditions.
3. Improved integration of the design and control of task conditioning systems with the building's central HVAC

system. Overall, building energy performance is closely related to the sophistication with which this integration occurs.

4. Investigation of worker productivity issues related to these systems.
5. Quantification and optimization of comfort control using task conditioning systems with human subject studies in laboratories.
6. Development of new task conditioning system designs that are less expensive and easier to install and maintain.

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DISCUSSION

C.Y. Shaw, Senior Researcher, Institute for Research in Construction, National Research Council of Canada, Ottawa, ON: Have you compared your desktop task-conditioning system with a desktop fan? Does the improvement justify the cost of installing such a system?

F.S. Bauman: We have recently completed a series of human subject tests in our controlled environment chamber in which subjects adjusted the local air speed provided by three different systems (desk fan, desktop task conditioning system, and floor-based task conditioning system) in order to maintain comfort at elevated room air temperatures. When asked to state their preference among the three systems, the results were fairly evenly divided. This, of course, is based only on thermal and mechanical (airflow annoyance) considerations.

Your question is a good practical one and is difficult to answer explicitly. We do know that task conditioning systems can provide improved ventilation efficiency by supplying fresh air at the breathing level. From an energy standpoint, local task conditioning fan units that are incorporated into the building's air distribution system should allow corresponding reductions in the central fan size. With separate desktop fans, however, the central fans would have to be sized to condition the entire space, regardless of the existence of desktop fans. The subject desktop system of this paper also contains an occupancy sensor that provides additional energy savings, although presumably this same technology could be applied to any electrical device. Recent field research by others on this same desktop system has indicated that these systems may have a positive influence on worker productivity, although the magnitude of this effect is difficult to quantify. There may even be a preference among office workers for a "high-tech" approach to task conditioning as opposed to simply placing a fan on their desk; perhaps a recirculating fan/filter system (already available from some partition manufacturers) integrated into the partition systems with individual controls would be an acceptable solution in some cases. Placing a value on these benefits of the desktop task conditioning system over a simple desktop fan is not easy, particularly for the more subjective occupant-related issues. It is clear, however, that if the provision of task conditioning can be positively correlated to even a small improvement in worker productivity, the economics will strongly justify wider use of systems that allow occupants to have greater control over their workplace environment.

John Mentzer, SSOE, Troy, MI: With a 60°F to 65°F supply air temperature through the subject system, how is the space humidity level maintained? In a space with a high frequency of occupant absence from dedicated workstations, how does the system deal with non-occupant-based fixed loads, such as general lighting, while occupancy sensors have the respective fans off?

Bauman: In its current configuration, the subject system is designed to be integrated with a conventional variable-air-volume (VAV) system that provides 55°F air at less than 0.1 in. WG static pressure to a pressure-regulating VAV box or an underfloor plenum, depending on the air distribution system layout. The 60°F to 65°F supply air temperature at the desktop outlet is obtained because of the temperature rise that naturally occurs due to heat transfer from the underdesk fan unit, the occupied space, and the underfloor plenum. With conventional cooling coil temperatures, humidity can be maintained at acceptable levels in the normal way by lowering the dew point of an adequate portion of the supply air to achieve the desired space humidity conditions when it is mixed with recirculated air from the system.

In most applications of task conditioning systems, some provision in the design would need to be made to accommodate ambient loads such as general lighting, as you mention. One approach is to design the air distribution system as a task/ambient air-conditioning system (similar to a task/ambient lighting system) in which two types of space conditioning are provided. (1) Conditioned air is supplied under automatic control to maintain minimum ambient comfort conditions in areas not controlled by local supply units, including areas in which significant numbers of local units have been turned off due to occupancy control. This could be done with a conventional ceiling-based system or with a floor-based (non-occupant-controlled) system. (2) Conditioned air is supplied through local supply units under occupant control to satisfy their individual comfort preferences.

The particular task conditioning system of this study has a design feature that sets a nonzero minimum supply volume (~ 4 cfm/unit) even when the local fan is turned off. This is intended to satisfy minimum ventilation rates, although 40 cfm seems unnecessarily high. In the field study reported in the paper, this minimum air supply often resulted in unoccupied workstations being overcooled, so improvements to the recommended system operation can hopefully be made as more experience is obtained.

Carl H. Jordan, Consulting Mechanical Engineer, Berkeley, CA: Please explain the procedures for calibration of temperature sensors. Did you follow ASHRAE's standard for temperature measurements (which standard)?

Bauman: Each temperature sensor was calibrated by placing it in an ice bath and adjusting its output (if necessary) to the manufacturer's reference voltage level. A subsequent side-by-side comparison of all sensors and an in-house high-quality laboratory thermometer at room temperature found agreement to within $\pm 0.2°F$ ($0.1°C$). In our test procedures, we positioned the sensor rig (containing all temperature and velocity sensors) at the desired measurement location in the room and waited 15 to 30 seconds before sampling all sensors over a 90-second measurement period. Preliminary tests had determined the length of this period as the minimum sampling time that still produced acceptable repeatability between consecutive measurements at the same position. Measurement heights corresponded to those recommended by ASHRAE Standards 55-1992 and 113-1990.

Ted N. Carnes, ASC, Richardson, TX: In which direction does a person at the workstation prefer the nozzles to be pointed? I would like to have this information in addition to airflow and temperatures.

Bauman: Due to the fact that the desktop task conditioning system is designed to be individually controlled by a person at the workstation, the possible answers to your question are as varied as the different comfort preferences found among building occupants. However, a few general statements can be made. As mentioned in the paper, during the field study the ambient space temperature (maintained by a separate ceiling-based air distribution system) tended to be on the cool side. We found in many instances under these conditions that the workers turned the nozzles away from them. On other occasions, when the worker was warm and wanted immediate cooling (i.e., after playing basketball during lunch), they pointed the nozzles directly at them. Under average conditions, workers will not tolerate a strong jet of air in their face for a substantial length of time. Based on our experience, people prefer to have only a light breeze on their face, or they turn the nozzles so that they provide good circulation within the workstation without blowing directly on them. Clearly, more information is needed on occupant use patterns, as they will be strongly dependent on the system design and operation (e.g., What ambient space temperature is maintained? What is the outlet supply temperature? What are the local heat loads in the space?).