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AIR MOVEMENT, VENTILATION, AND COMFORT IN A PARTITIONED OFFICE SPACE

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ABSTRACT

Results are presented from a research project to investigate the effects of office partition design on air movement, worker comfort, and ventilation in workstations. The objectives of the study were to evaluate the comfort and ventilation conditions produced by a conventional ceiling supply-and-return air distribution system in workstations separated by (1) solid partitions of different height (75 in. [1.9 m], 65 in. [1.65 m], 42 in. [1.1 m], and 0 in. [partitions removed]) and (2) partitions containing a gap positioned at the bottom of the partition. The project consisted primarily of experiments performed in a full-scale controlled environment chamber (CEC) in which a typical modular office environment was set up. The range of partition configurations and environmental parameters investigated included (1) partition height, (2) solid vs. airflow partitions, (3) airflow gap size, (4) supply air volume, (5) supply/room temperature difference, (6) supply diffuser location, (7) heat load density, (8) workstation size, and (9) cooling vs. heating mode. Under steady-state conditions, multipoint measurements were made of air velocities, air temperatures, and radiant (globe) temperatures to characterize the key environmental variables affecting thermal comfort, and tracer gas methods using multipoint sampling locations were employed to determine the ventilation performance within the test chamber.

The results indicated that variations in solid partition height produce only small differences in overall thermal and ventilation performance. Results also showed that while the existence of an airflow opening at the bottom of office partitions can, in some cases, produce slight increases in air velocities near the floor, there are no significant improvements in comfort conditions or ventilation efficiency within the workstations compared to results obtained for solid partitions. Test parameters that were found to have a more substantial impact on air movement and comfort included heat load density and distribution, supply air temperature, and supply diffuser location.

INTRODUCTION

Recent developments in office design, function, and technology make it increasingly difficult for conventional centralized HVAC systems to satisfy the environmental preferences of individual office workers. Valuable data from several recent occupant surveys of large office buildings more precisely define the range of environmental factors that are critically related to the interdependent relationships between a building and its occupants (Harris 1980; Brill 1984; Woods et al. 1987; Dillon and Vischer 1987; Baillie et al. 1987; Schiller et al. 1988).

In today's typical open-plan office building, the design and layout of workstation furniture and partitions can play important roles in determining the nature of many of these environmental factors, including thermal and airflow conditions, noise and spatial privacy, and the functionality of the workplace. Workstations are frequently separated by partitions that may, under certain conditions, divert the flow of air between conventional, ceiling-mounted supply diffusers and return registers so that the workstations themselves are not well ventilated. The workstations are also often reconfigured to accommodate changing tenant needs, affecting the HVAC system's ability to meet the loads for which they were designed. Modern offices also have large amounts of heat-generating equipment (computers, printers, etc.) whose loads may vary considerably from workstation to workstation. Finally, with the growing awareness of the importance of the comfort, health, and productivity of office workers, the increased demand among employers and employees for a high-quality work environment cannot always be met by conventional approaches to HVAC and office design.

Standards for maintaining comfortable indoor thermal environments have been developed by ASHRAE (1981) and ISO (1984). Both of these standards specify a zone of relatively uniform conditions within which no more than 20% of the occupants are expected to be dissatisfied. Although 20% is in itself a fairly large number, a recent field study in office buildings suggests that the dissatisfac-

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tion level for environments maintained within the comfort zone may in fact be substantially higher (Schiller et al. 1988). In addition, this study and others (Croome and Rollason 1988; Harris 1989) have found that lack of air movement is one of the most common complaints in office environments, although the low air movement rates are mandated by the standards.

There is understandably a great deal of concern in the building engineering community over the potentially detrimental effects of office partitions on air movement, comfort, and air quality. "Airflow" partitions, or partitions that have been raised off the floor, thereby providing a gap for additional air movement between adjacent workstations, have been introduced as one possible means for improving airflow conditions. The currently available literature, however, provides only a few reports describing the effects of partitions (both solid and "airflow") on air movement in office environments.

Hart and Int-Hout (1981) tested the influence of 1.5 m (5 ft) vertical acoustical screens placed at various locations with respect to continuous linear diffusers in an open-plan office. They measured the ASHRAE-defined Air Diffusion Performance Index (ADPI) (ASHRAE 1990) and found relatively good performance and circulation for all configurations tested. Public Works Canada (PWC) performed air circulation tests in a building in Calgary, Alberta, using a variety of flow visualization techniques to provide a qualitative assessment of supply air movement from the diffusers (PWC 1983). It was concluded that the mechanical system was "generally performing adequately"; however, operating and layout characteristics of the air-distribution system and the positioning of partitions in the office created some areas where negligible air movement was observed. Based on its observations in the building, PWC recommended that partitions be raised slightly off the floor to provide good air circulation. Subsequently, PWC documented in detail its methods for evaluating air circulation in buildings (Tilley 1988). Huvinen and Rantama (1987) tested ventilation efficiency in a 3-m by 3-m (10-ft by 10-ft) partitioned office space within a larger open-plan office. Theoretical predictions and a limited amount of experimental data showed little difference between 1.5 m (5 ft) and 2.0 m (6.6 ft) high partitions. In the same study, significant differences in air circulation were predicted by the model when a gap was provided at the bottom of the partitions, but no experimental data were presented to verify this result. The results were strongly dependent on the inlet/outlet configuration and the control objectives of the mechanical system. In a recent publication, Nguyen (1990) reported on full-scale testing of ventilation effectiveness for office partitions of two different heights (48 in. and 62 in.) that were raised above the floor by 3, 6, 9, and 12 inches. Although it was concluded that "depending on the type of diffuser, raising the partitions above the floor at a certain elevation does provide a better fresh air exchange rate," and "the height of partitions... has an impact on air exchange rates and air velocities," there were not enough experimental data reported from which to accurately understand the rationale behind these statements.

In the current study, a series of detailed laboratory experiments were carried out to investigate the effects of office partition configurations and environmental control parameters on thermal and ventilation conditions within workstations. The range of partition and environmental parameters investigated included partition height, solid vs. airflow partitions, airflow gap size, supply air volume, supply/room temperature difference, supply diffuser location, heat load density, workstation size, and cooling vs. heating mode. The current effort did not include modeling by either detailed numerical or simplified methods in order to address the fundamentals of the airflow conditions under study. Future work is planned in this area.

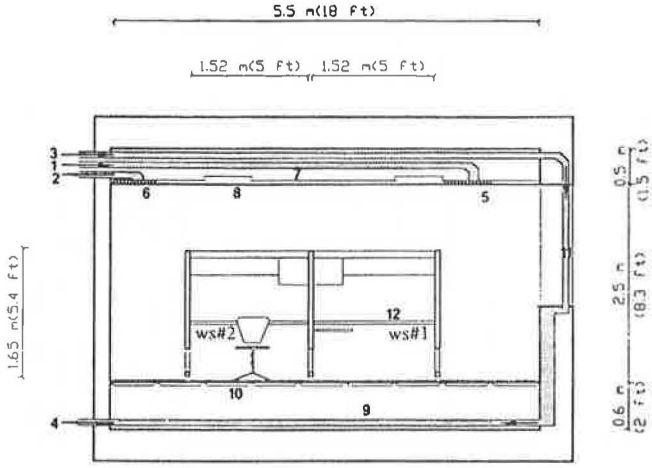
The overall objectives of this study were (1) to evaluate the conditions under which partition designs can improve or degrade air movement, ventilation performance, and worker comfort, and (2) to evaluate the effects on air movement, ventilation performance, and worker comfort of providing an opening at the bottom of partitions.

EXPERIMENTAL METHODS

Controlled Environment Chamber

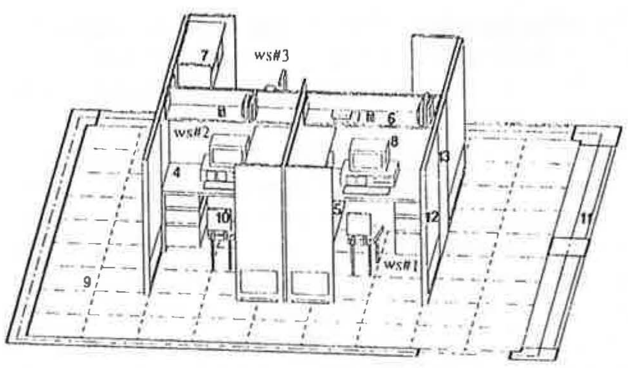
All experiments were performed in a controlled environment chamber (CEC) measuring 18 ft by 18 ft by 8 ft, 4 in. (5.5 m by 5.5 m by 2.5 m) and located in a university laboratory. The CEC is designed to resemble a modern office space while still allowing a high degree of control over the test chamber's thermal environment (Bauman and Arens 1988). The floor is fully covered with carpet tiles, the finished gypboard walls are heavily insulated and painted white, triple-pane windows in the two exterior walls provide a view to the outside, the suspended ceiling contains patterned acoustical tile, and six 2 ft (0.6 m) square recessed dimmable lighting fixtures are mounted in the ceiling. As shown in Figure 1a, a raised-access floor system provides a 2 ft (0.6 m) high subfloor plenum, and the suspended ceiling provides a 1.5-ft (0.5-m) ceiling plenum.

A typical modular office configuration was installed in the test chamber. As shown in Figure 1b, the partitions 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. (3.05-m by 1.9-m) workstation. The arrangement of furniture, including desks, side tables, and overhead storage bins, is also shown in the figure. The base-case partition configuration used during a large majority of the tests consisted of medium-height (65-in. [1.65-m]) airflow partitions. Figure 1b also shows the locations of the airflow and solid partitions; airflow partitions were



1 ceiling supply
2 ceiling return
3 annular space supply
4 annular space return
5 ceiling supply diffuser
6 ceiling return register
7 ceiling plenum
8 ceiling luminaire
9 sub-floor plenum
10 access floor system
11 window plenum
12 workstation furniture

Figure 1a *Section: controlled environment chamber.*

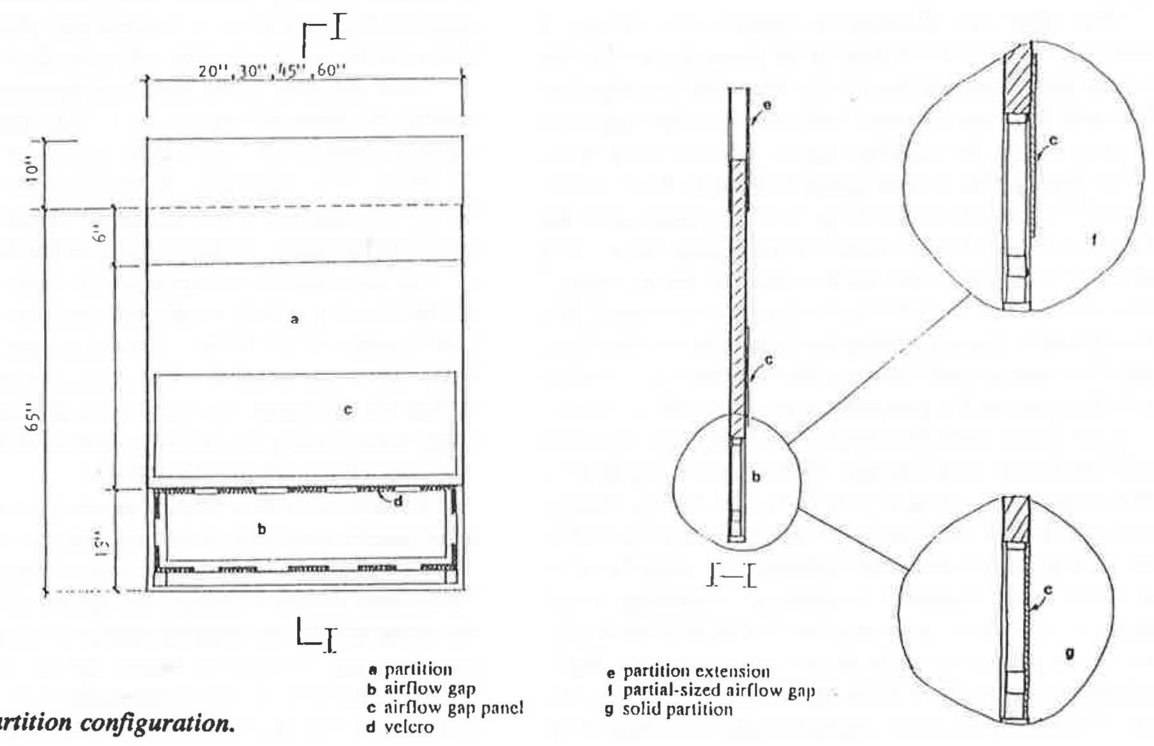


ws#1 workstation #1
ws#2 workstation #2
ws#3 workstation #3
4 desk
5 sidetable
6 bookshelf
7 overhead storage
8 computer
9 access floor panel
10 chair
11 annular space (window)
12 airflow/solid partition
13 solid partition

Figure 1b *Chamber schematic.*

installed everywhere except along the 30-in. (0.76-m) sides of the desks, where the desk support would completely block any airflow gap. In order to take advantage of airflow partitions placed along the back of each desk and side table, all modesty panels (vertical panel on backside of desk) were removed.

To aid the experimental method for comparing the performance of solid vs. airflow partitions, replacement panels for each airflow gap were fabricated out of 1/4-in. foam core. Velcro strips placed on the back of each panel allowed it to be positioned to completely cover the airflow gap (forming a solid partition) or to be easily secured to the fabric of the partition to produce a full-sized or partial-sized airflow gap (Figure 2). Also shown in Figure 2 are 10-in. (0.25-m) extension panels that were designed and fabricated to fit on top of the 65-in. partitions, thereby increasing the overall partition height to 75 in. (1.9 m). These extension panels allowed us to quickly convert the office configuration from medium-height to tall partitions, improving the comparability of measurement results obtained under similar thermal conditions.



a partition
b airflow gap
c airflow gap panel
d velcro
e partition extension
f partial-sized airflow gap
g solid partition

Figure 2 *Partition configuration.*

The extension panels also allowed us to investigate the effects of the airflow gap in 75-in.-tall partitions.

The CEC's reconfigurable air distribution system permits ducted or plenum air to be supplied to and returned from the test chamber at any combination of ceiling and floor locations. Figure 1a shows the airflow configuration used during the tests reported here, consisting of a conventional ducted ceiling supply-and-return air distribution system. Figure 3 describes the various locations of the supply diffuser(s) and return register used during the tests in relation to the nine-by-nine grid of 2-ft by 2-ft (0.6-m by 0.6-m) suspended ceiling panels. During most tests, supply air was provided through a single perforated lay-in diffuser, positioned near one side of the room at (x = 5, y = 2). At this position, the internal pattern deflectors were adjusted to produce a three-way airflow pattern, away from the adjacent wall, as shown in Figure 3. A single perforated return register was located at (5,9) during all tests. By placing supply and return locations at opposite sides of the room, airflow conditions in the central region of the test chamber were expected to resemble those encountered in open-plan offices, where most workstations are positioned somewhere between supply and return locations. Figure 3 also shows the alternative diffuser locations and blow configurations that were studied during a series of additional tests that will be discussed later. These include (1) a single diffuser at (8,5) with three-way blow away from the window; (2) a single diffuser at (2,8) with two-way blow away from the adjacent corner of the room; (3) a single diffuser at the base-case position (5,2) with three-way blow away from the window; and (4) two diffusers at (2,2) and (8,2) with two-way blow away from the adjacent corners.

The CEC air distribution system also allows a separately controlled 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. Consequently, the exterior walls and windows were not a source of strong natural convective airflow but affected indoor air movement like interior walls. During heating-mode tests in the chamber, cooled air was passed between the windows to simulate cooling effects in the perimeter zone of an office space.

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. During the thermal measurements, a second 75 W

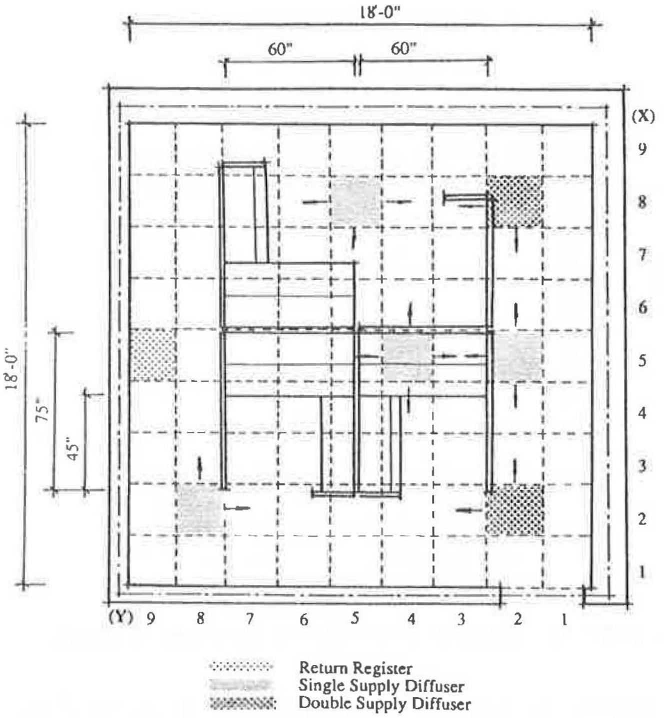


Figure 3 *Supply and return locations.*

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 150 W (510 Btu/h) to the total load during these tests. During the tracer gas measurements, one or two of the three workstations was occupied by a seated mannequin. Electric resistance heating elements wrapped around the mannequin released 75 W in a manner that closely resembled the sensible heat load of an office worker.

Two different office heat load densities were studied during the thermal experiments. The heat sources described above generated a load density of approximately 35 W/m^2 (11 Btu/h·ft^2). A higher load density of 55 W/m^2 (18 Btu/h·ft^2) was produced by placing a 200 W (680 Btu/h) electric radiant heater on the floor under each desk to represent larger computer processing units. Most of the tracer gas tests were performed at the lower heat load density of 35 W/m^2. During a few tests, internal loads were increased by the operation of mixing fans within the chamber. In tests with the chamber heated using warm supply air, the only additional heat gain to the space was from the overhead lights.

Except for a few heating-mode tracer-gas tests, all experiments were carried out under steady-state conditions chosen to represent an interior zone of an office building. To achieve these conditions, the electrical heat sources in the room and the mechanical system were turned on in the morning and allowed to warm up the room until the expected average room temperature for the upcoming experiment was reached (22°C to 28°C [72°F to 82°F] during these tests). After completing the warmup, the supply air volume and temperature were adjusted to their selected setpoints, and conditions in the room were allowed to further stabilize. Typical control of the supply air temperature entering the room was to within $\pm 1.0°C$ (1.8°F) over the test period. Room humidity levels were not controlled during the tests.

Heating mode tests were initiated with a similar cooldown period during which cool air was passed through the annular space until the windows reached a steady minimum temperature ($\approx$13°C [55°F]). After the warm supply air temperature and volume into the test chamber were set and stabilized, the test proceeded under steady-state conditions.

Thermal Measurements

Detailed air velocity and temperature measurements within the test room were accomplished by using a lightweight 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 26 selected measurement locations (Figure 4a). 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). Temperature and velocity sensors were sampled 50 times over a 90-second measurement period. The measurement equipment, sensor calibration, and data acquisition system have been described in detail by Bauman et al. (1991a).

Table 1 lists the thermal measurement test conditions. A total of 39 separate tests were completed, including 6 preliminary tests (P1A-P3B) and 33 final tests (1A-16). The tests investigated the following ranges of test parameters: (1) supply air volume from 54 cfm (0.2 cfm/ft^2) to 320 cfm (1.0 cfm/ft^2); (2) heat load densities of 35 and 55 W/m^2 (11 Btu/h·ft^2 and 18 Btu/h·ft^2); (3) supply air temperature from 12.8°C to 19.5°C (55°F to 67°F); (4) average room temperature from 21.9°C to 28.5°C (71°F to 83°F); (5) return/supply air temperature difference from 5.6°C to 12.3°C (10°F to 22°F); (6) 75-in., 65-in., 42-in., and 0-in. (no partitions) partition heights; (7) solid partitions and full-open (12-in.), 4-in., and 2-in. airflow gaps.

The final tests listed in Table 1 can be divided into two groups according to supply volume: (1) low supply volume in the range of 150 to 180 cfm (0.5 cfm/ft^2) and (2) high supply volume in the range of 280 to 320 cfm (0.9 to 1.0 cfm/ft^2). At the low supply air volume, the throw of the supply diffuser was at the minimum level recommended by the manufacturer to achieve acceptable room air diffusion. The results of these tests are, there-

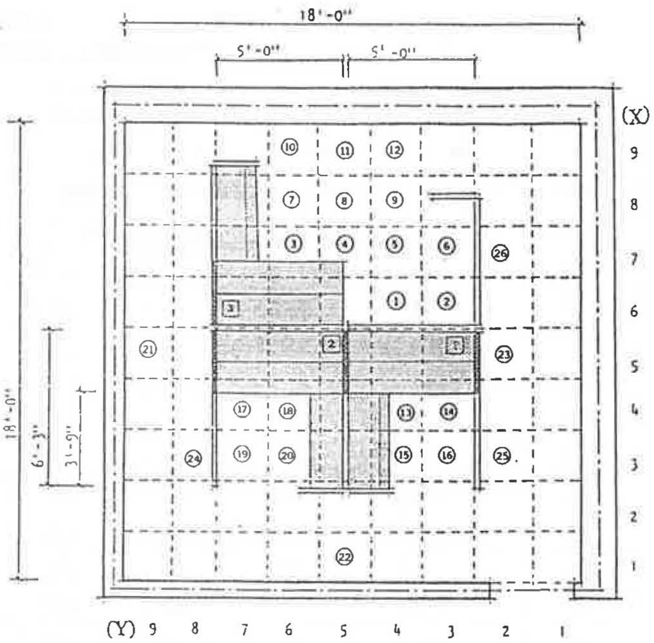


Figure 4a Thermal measurement locations.

fore, indicative of a single VAV diffuser operating at or below its minimum airflow rate. At the higher supply air volume, the duct diameter of the neck leading into the diffuser had to be increased from 6 inches (low volume tests) to 10 inches. This modification kept the noise generated by the diffuser to a level less than NC = 35 and provided a throw within the acceptable range for good room air diffusion. Note that the room temperature reported in Table 1 was measured at a typical wall thermostat location and, due to the effect of the warm adjacent wall, is quite close to the return air temperature. For a given supply air volume, the return air volume was adjusted to maintain a slight overpressure in the test room in relation to the surrounding rooms of the building. Therefore, since the chamber was not perfectly sealed, and due to the relatively high ambient pressure in the surrounding spaces, the return air volume was always less than the supply air volume.

Tracer Gas Measurements

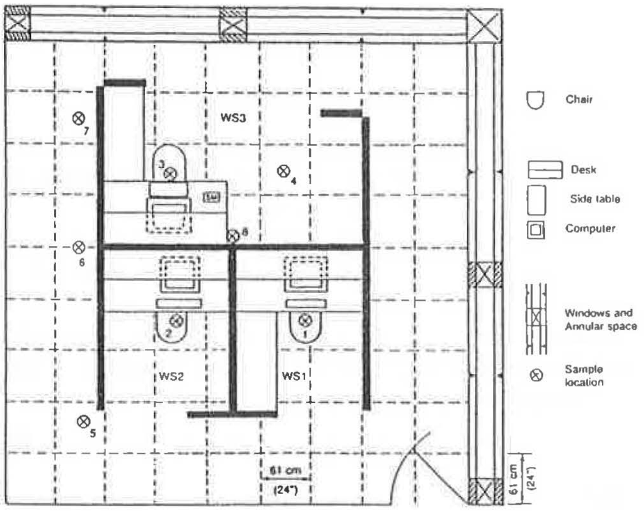
The tracer gas step-up procedure (Sandberg and Sjoberg 1983; Fisk et al. 1988, 1989) was used to study indoor airflow patterns and the spatial variability of ventilation. In this procedure, the supply air was labeled with a tracer gas and the rate of increase of tracer gas concentrations at a location indicated how rapidly the indoor air was replaced with outdoor air that entered the building after the start of tracer gas injection. During the step-ups, a mixture of 1% sulfur hexafluoride (SF_6) in air was injected at a constant rate into the supply airstream. A peristaltic pump drew the tracer/air mixture from a storage bag and directed the mixture through a flowmeter and tubing into the supply duct. The injection rate was

TABLE 1
Thermal Measurement Test Conditions

Test Number	Supply Air Volume (cfm)	Heat Load (W/sq.m)	Return Air Volume (cfm)	Supply Air Temp. (deg C)	Room Temp* (deg C)	Return Air Temp. (deg C)	Partition Height (inches)	Partition Air Gap	Diffuser Location**
P1A	55	55	-	16.5	27.9	-	65	Solid	(5,2)
P1B	54	55	-	16.7	28.5	-	65	Open	(5,2)
P2A	108	35	-	12.8	23.9	-	65	Solid	(5,4)
P2B	107	35	-	13.0	23.8	-	65	Open	(5,4)
P3A	179	55	-	13.9	25.1	-	65	Solid	(5,2)
P3B	180	55	-	14.3	25.8	-	65	Open	(5,2)
1A	152	55	133	13.1	24.2	24.3	65	Solid	(5,2)
1B	152	55	133	13.0	24.2	24.4	65	Open	(5,2)
2A	154	55	143	13.0	24.5	24.7	65	Two inch	(5,2)
2B	154	55	144	13.1	24.1	24.6	65	Solid	(5,2)
3A	157	55	144	18.0	26.4	26.5	65	Solid	(5,2)
3B	152	55	157	18.0	26.0	26.5	65	Open	(5,2)
4A	155	35	133	13.2	21.9	21.6	65	Open	(5,2)
4B	154	35	114	13.3	22.1	21.5	65	Solid	(5,2)
5A	319	55	258	15.3	23.6	23.5	65	Solid	(5,2)
5B	319	55	260	13.3	23.2	23.0	65	Open	(5,2)
6A	300	55	200	13.8	22.0	22.6	65	Comb.	(5,2)
6B	305	55	201	13.4	21.9	22.6	65	Two inch	(5,2)
6C	296	55	201	13.3	22.0	22.6	65	Solid	(5,2)
7A	171	55	143	13.3	25.0	25.1	75	Solid	(5,2)
7B	170	55	143	13.0	25.3	25.2	75	Open	(5,2)
7C	170	55	145	13.1	25.4	25.4	65	Solid	(5,2)
8A	288	55	249	18.2	25.0	24.9	75	Open	(5,2)
8B	288	55	249	17.9	25.2	25.2	75	Solid	(5,2)
8C	289	55	252	18.0	25.5	25.5	65	Solid	(5,2)
9A	291	55	246	18.3	24.3	25.0	65	Solid	(5,2)
9B	292	55	241	18.0	24.5	25.1	65	Four inch	(5,2)
9C	293	55	245	18.0	24.9	25.3	75	Four inch	(5,2)
10A	282	55	236	18.1	24.6	24.9	65	Open	(8,5)
10B	282	55	237	17.9	24.7	25.1	65	Solid	(8,5)
11A	290	55	241	18.1	23.5	25.2	65	Open	(2,8)
11B	291	55	241	18.0	23.5	25.2	65	Solid	(2,8)
12A	320	55	255	19.3	24.3	24.9	65	Open	(5,2)
12B	320	55	258	19.5	24.8	25.5	65	Solid	(5,2)
13A	309	55	259	17.9	24.4	24.7	65	Open	(2,2)(8,2)
13B	309	55	259	18.0	24.6	24.9	65	Solid	(2,2)(8,2)
14	312	55	277	17.0	24.6	24.7	None	-	(5,2)
15	308	55	276	16.9	24.5	24.5	42	Solid	(5,2)
16	169	55	160	13.0	23.7	24.3	42	Solid	(5,2)

*Room temperature measured at height of 4.5 ft on chamber wall

**Refer to Figure 3



Plan view of CEC with workstations denoted WS1, WS2, WS3. Tracer gas was sampled at points 1-4 (0.4 m, 1.1 m, and 2.1 m above floor), at points 5 & 7 (1.1 m above floor), at point 6 (1.1 m above floor and 2.1 m above floor), and at point 8 (2.1 m above floor).

Figure 4b *Tracer gas sampling locations.*

monitored using rotameters calibrated with a bubble flowmeter and was generally stable within ±2%. To ensure thorough mixing of the SF_6 in the supply airstream, an array of small propeller fans was installed downstream of the injection point. These fans were oriented to cause airflow perpendicular to the general direction of flow in the duct. Mixing was confirmed by collection and analysis of air/tracer samples. Air samples were drawn continuously through copper tubes to three gas chromatographs (GCs) equipped with electron capture detectors. During tests 21-25, five samples originated from within the chamber at a subset of the locations illustrated in Figure 4b and four samples originated from the HVAC system. During tests 39-46, one sample originated from within the chamber and four samples originated from the HVAC system. The GCs were capable of analyzing a sample within 1 minute using a 0.38-m-long molecular sieve main column; a backflush column with two sections (0.08 m of 5% phosphoric acid on acid-wash diatomaceous earth support followed by 0.38 m of molecular sieve); carrier gas (5% methane, 95% argon) flow rates of approximately 40 cc/min; and approximately a 12 s backflush time (Harrje 1990). Using this method, the tracer gas concentration was measured every three minutes at each sample location. The time required to perform repeated real-time tracer gas measurements limited the maximum supply air volume (200 cfm) for which reliable test results could be achieved. As a result, as seen in Table 2, most tracer gas tests were performed at a low supply volume of 100 cfm (0.3 cfm/ft^2).

During the tests, bag samplers also directed air/tracer samples at a constant rate into 0.75-L sample bags. Bag sampling commenced at the start of tracer gas injection and continued until tracer gas concentrations were stable (as determined from the periodic measurements of tracer gas concentration in the return duct) at which time syringe samples were collected manually from each location. The 14 (tests 21-25) or 17 (tests 39-46) bag samplers collected samples from the locations within the chamber depicted in

TABLE 2
Tracer Gas Test Conditions

Test Number	Supply Air Volume (cfm)	Supply Air Temp (deg C)	Room Temp (deg C)	Partition Height (inches)	Partition Air Gap	Diffuser Location**	Comments
21	100	13	24	65	Solid	(5,4)	
22W	100	15	24	65	Solid	(5,4)	Mixing fans in CEC
22*	100	15	24	65	Solid	(5,4)	
23	150	25	22	65	Solid	(5,4)	Heating test
24	110	14	24	65	S&O+	(5,4)	
25	110	14	26	65	S&O	(5,2)	
39	210	na	24	75?	Solid	(5,2)	
40	200	18	24	75	Open	(5,2)	
41	55	18	26	75	Solid	(5,2)	
42W	100	13	26	65	Open	(5,2)	Mixing fans in CEC
43	100	25	23	75	Solid	(5,2)	Heating test
45	70	25	22	75	Solid	(5,2)	Heating test
46	100	25	23	75	Solid	(5,2)	Heating test

*Tracer decay, all other tests are tracer stepup
** See Figure 3
+Partition gaps open in WS 1&2

Figure 4b. Air samples were directed to both a sample bag and a GC at some locations; thus, samples were collected and analyzed from 17 unique locations within the CEC. Bag and syringe samples were analyzed using the GCs immediately after completion of the tests. Equipment and procedures are similar to those used previously and described by Fisk et al. (1985, 1988, 1989).

The GCs were calibrated prior to each test using nine total calibration gases with SF_6 concentrations of 0 ppb to 185 ppb. Measurements of tracer gas concentrations were generally repeatable within a couple ppb.

Since the tracer gas measurement methods required the test chamber to be closed and unoccupied throughout each test, tracer gas tests were performed on separate days from the thermal tests described in Table 1. Table 2 lists the tracer gas test conditions. There is no relationship between test numbers for Tables 1 and 2. Gaps in the sequence of test numbers are due to tests with air supplied through floor units or unsuccessful tests. During tests 21-25, the supply diffuser was centrally located at position (5,4) (see Figure 3) and adjusted for a four-way 360° air supply orientation. During tests 39-46, the supply diffuser was located at position (5,2) and adjusted for a three-way 270° air supply orientation with no air directed toward the windows. Test variables included partition height, absence or presence of a 0.3-m (12-in.) gap at the bottom of the partitions, supply flow rate, supply temperature, and internal heat loads. In most tests, the CEC was cooled to offset the internal heat loads. During tests 23, 43, 45, and 46, the windows were cooled, internal heat generation was reduced, and the supply air was used to heat the chamber. To determine measurement precision, tests 22W and 42W were run with fans operating in the chamber to vigorously mix the chamber air. Test 22 was a tracer gas decay (instead of step-up) with the tracer gas concentration uniform at the start of the decay and no tracer injection during the decay.

Tracer Gas Data Analysis

Age-of-air concepts are a common basis for evaluating ventilation efficiency and the spatial variability of ventilation in a ventilated space. The age of air in a sample collected at a specific location is the time that has elapsed since the air entered the building. The reciprocal of the age of air is a measure of a local ventilation rate. Thus, a relatively low age of air indicates a higher rate of ventilation than a relatively high age of air. Equations based on age distribution theory (Sandberg and Sjoberg 1983) were used to calculate the ages of air. We present only the equations for a tracer gas step-up; similar equations for data from tracer gas decays are presented elsewhere (Sandberg and Sjoberg 1983). Using tracer gas concentrations as a function of time, the following equation was employed:

$$A = \int_0^{t_{ss}} [1 - C(t)/C(t_{ss})]\, dt \tag{1}$$

where

A = age of air,
t = time variable set equal to zero at the start of tracer gas injection,
$C(t)$ = tracer gas concentration at time t,
t_{ss} = time when concentrations have stabilized.

The integral is evaluated numerically. Using the tracer gas concentrations in bag and syringe samples, age of air was determined using the equation

$$A = t_{bag}(1 - C_{bag}/C_{syr}) \tag{2}$$

where

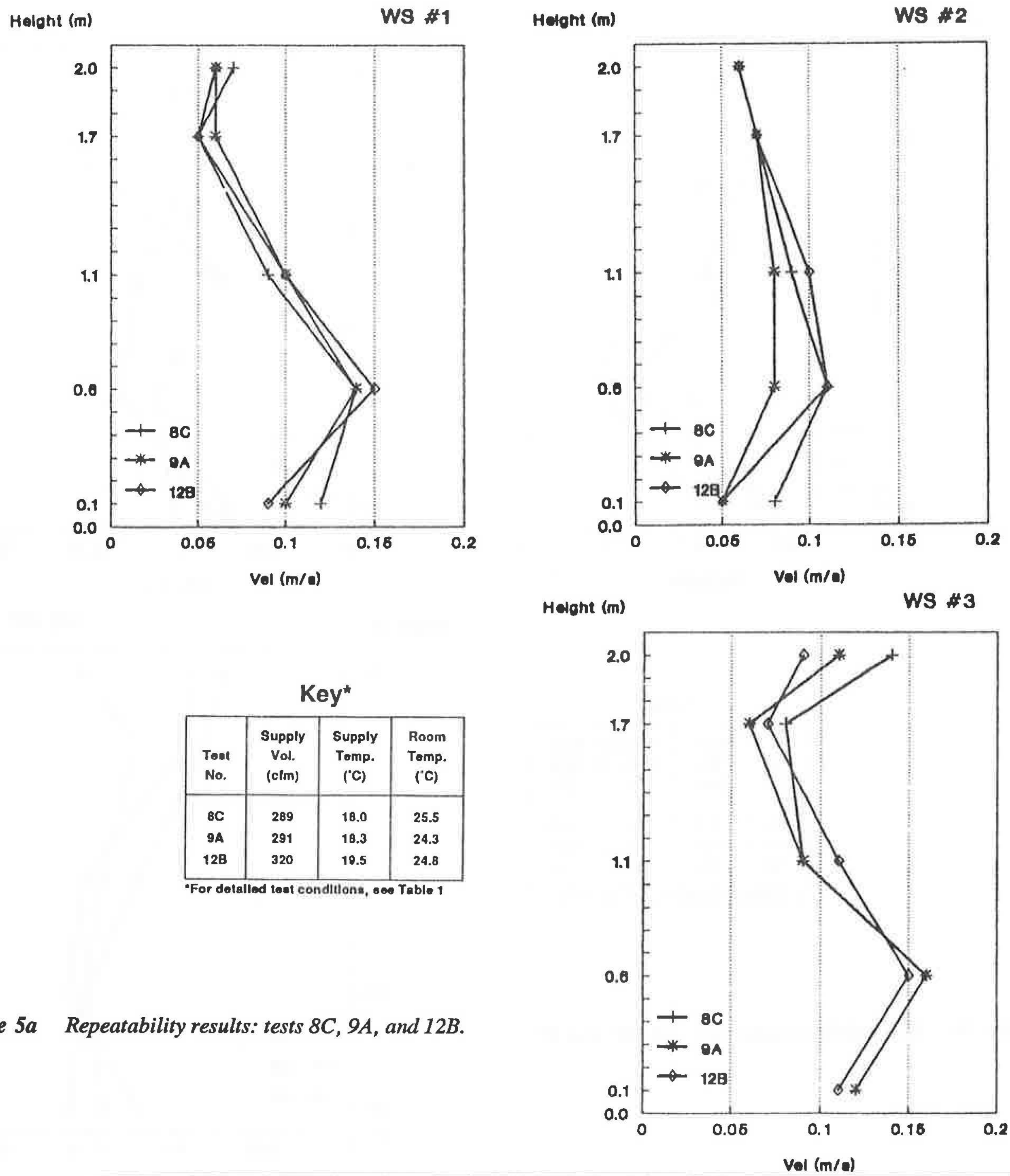
t_{bag} = elapsed time of bag sampling,
C_{bag} = concentration in bag sample,
C_{syr} = concentration in syringe sample.

To indicate the spatial variability in the age of air, we use the age expressed in hours and various ratios based on the ages. For example, the age of air in the return duct divided by the average age of air in all of the workstations at 0.4 m and 1.1 m above the floor yields a ratio that is an indicator of short-circuiting. With short-circuiting, fresh (low age) air does not mix thoroughly with room air before exiting via the return duct. Therefore, values less than unity for this ratio indicate short-circuiting since the age of air in the return is lower than the age in the workstations. When ratios contain an average of the age measured at several locations, we use volume-weighted averages, assuming that each measurement is representative of a volume that extends halfway to adjacent measurement points and/or to the edge of the workstation.

RESULTS AND DISCUSSION

Thermal Measurements

Due to the large amount of experimental data, a subset of tests has been selected from Table 1 for presentation and discussion to demonstrate the effects of each of the major test parameters investigated. A more complete presentation of results is reported by Bauman et al. (1991b). 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 below as the velocity or temperature calculated by averaging the measured values obtained from the four locations directly in front of the desk. Referring to Figure 4a, average conditions in WS #1 are based on points 13, 14, 15, and 16; those in WS #2 are based on points 17, 18, 19, and 20; and those in WS #3 are based on points 3, 4, 7, and 8. For most cases, results are shown only for measurement heights from 0.1 to 2.0



Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)
8C	289	18.0	25.5
9A	291	18.3	24.3
12B	320	19.5	24.8

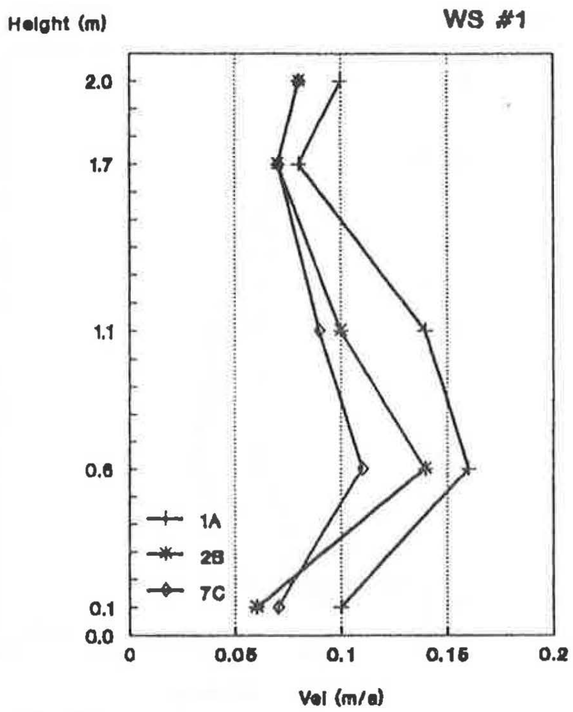
*For detailed test conditions, see Table 1

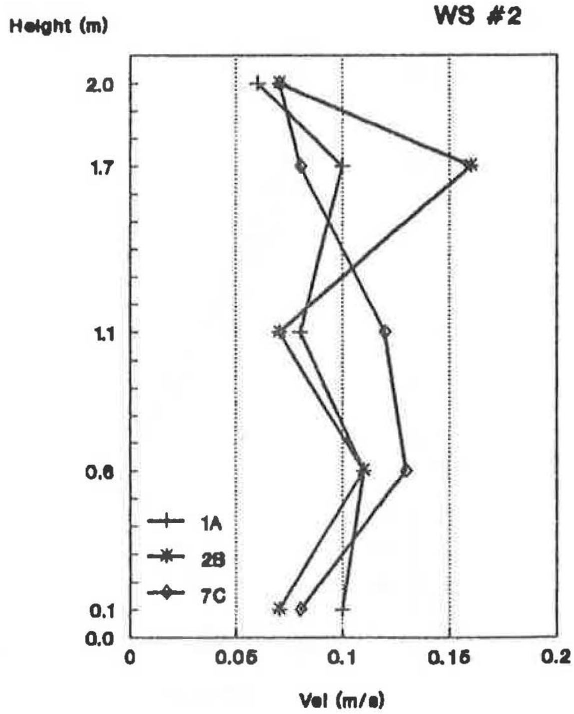
Figure 5a Repeatability results: tests 8C, 9A, and 12B.

m, enabling greater detail to be observed and an improved comparison to be made between separate measurements at lower heights in the room. The higher velocities near the ceiling (2.35 m height) followed a consistent pattern in relation to the location of the supply diffuser.

Based on preliminary test results indicating that the largest effects, if any, of the airflow gap partition would occur at the highest supply air volumes, a base-case set of environmental control conditions was selected and used for a majority of the parametric studies investigating the influence of partition design. These base-case conditions consisted of high supply air volume (0.9 to 1.0 cfm/ft^2); high heat load density (55 W/m^2); supply diffuser location at (5,2) (see Figure 3); and supply air temperature in the range of approximately 17 to 19°C, selected to maintain the average room temperature in the range of approximately 24 to 25°C.

Data Precision A measure of the experimental repeatability of our thermal measurements is indicated in Figures 5a and 5b, each of which presents velocity results from three tests having similar test conditions. Figure 5a shows results from tests 8C, 9A, and 12B for solid 65-in. partitions under base-case conditions. Figure 5b shows results from tests 1A, 2B, and 7C for solid 65-in. par-





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)
1A	152	13.1	24.2
2B	154	13.1	24.1
7C	170	13.1	25.4

*For detailed test conditions, see Table 1

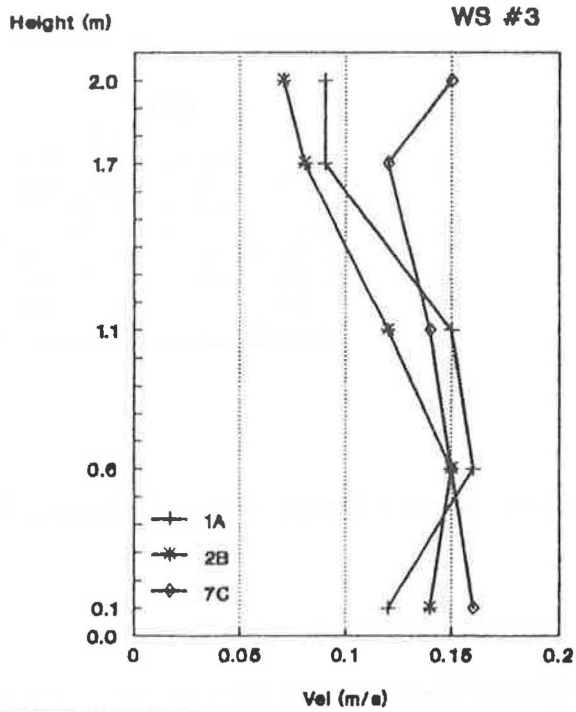
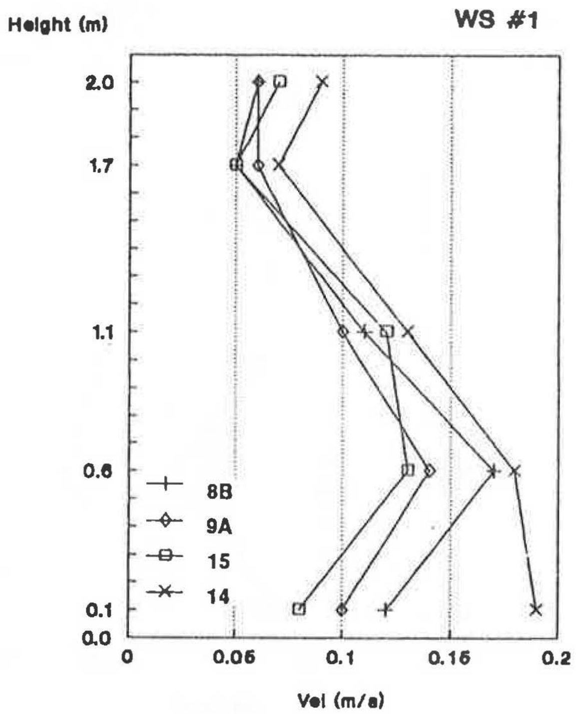
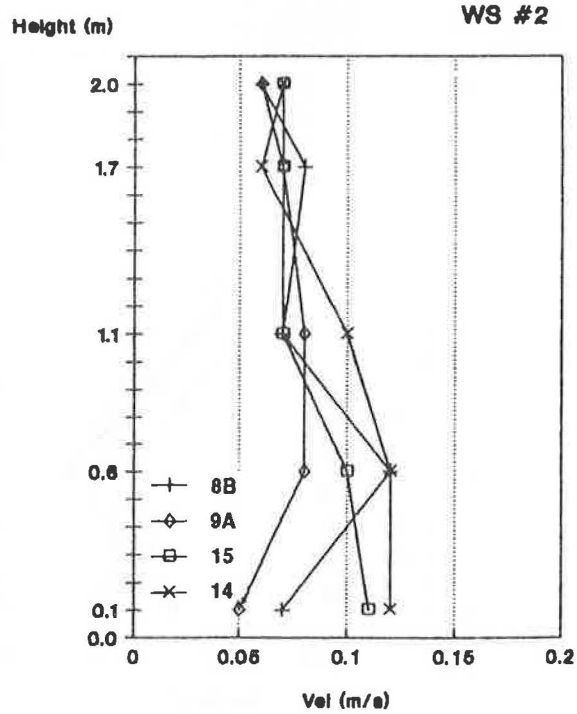


***Figure 5b** Repeatability results: tests 1A, 2B, and 7C.*

titions at low supply air volume ($\approx$ 0.5 cfm/ft^2), high heat load density (55 W/m^2), supply diffuser location at (5,2), and a lower supply air temperature of 13°C. In Figure 5a, at the higher supply air volume, the results are repeatable for all three workstations, with all measured velocities in the occupied zone (0.1 to 1.7 m) falling within 0.03 m/s of each other. This is a good result, as there is some variation in the supply air volume and temperature conditions between the three tests, but the test-to-test variability is only slightly greater than the calibrated accuracy of the anemometers. In Figure 5b, at a lower supply volume and temperature, velocities in the occupied zone are repeatable to within 0.05 m/s, except at the 1.7-m level in WS #2. The wider variation in WS #2 is due to the minimum diffuser throw characteristics described earlier.

Solid Partition Height Figure 6 presents average velocity results for tests 8B, 9A, 15, and 14, corresponding to solid partition heights of 75 in., 65 in., and 42 in. and no partitions, respectively. The tests were performed under base-case conditions and the results are organized by workstation. The observations are as follows:





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Part. Ht. (in.)
8B	288	17.9	25.2	75
9A	291	18.3	24.3	65
15	308	16.9	24.5	42
14	312	17.0	24.6	None

*For detailed test conditions, see Table 1

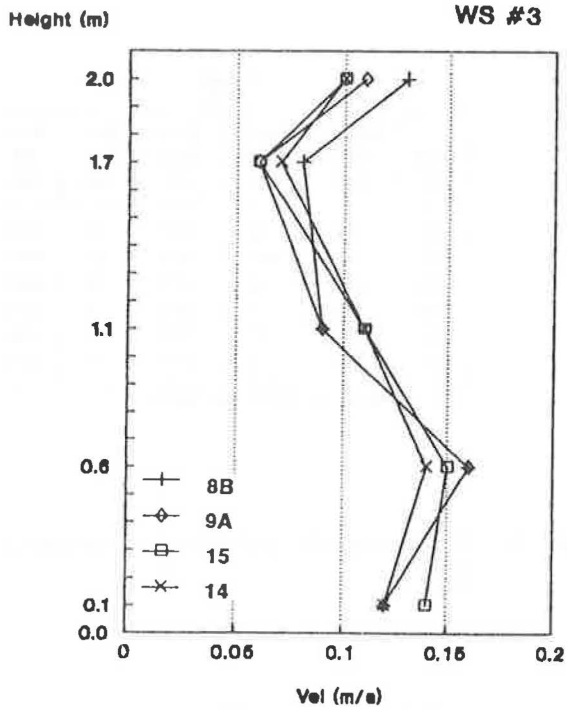
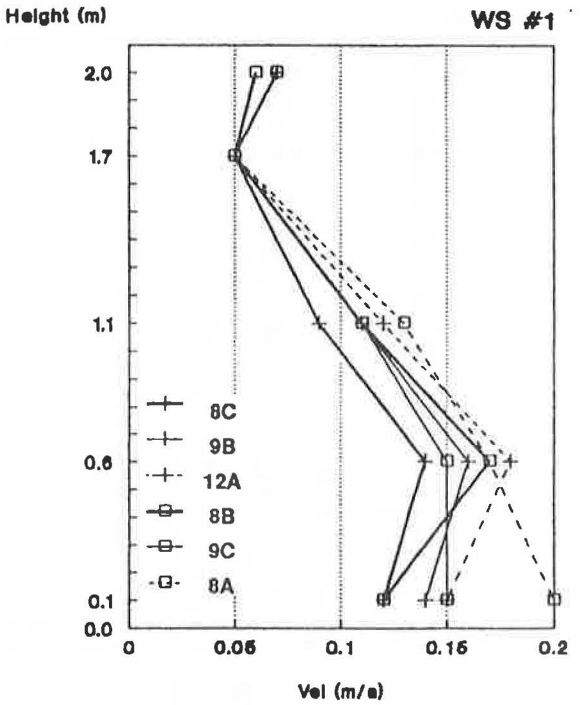
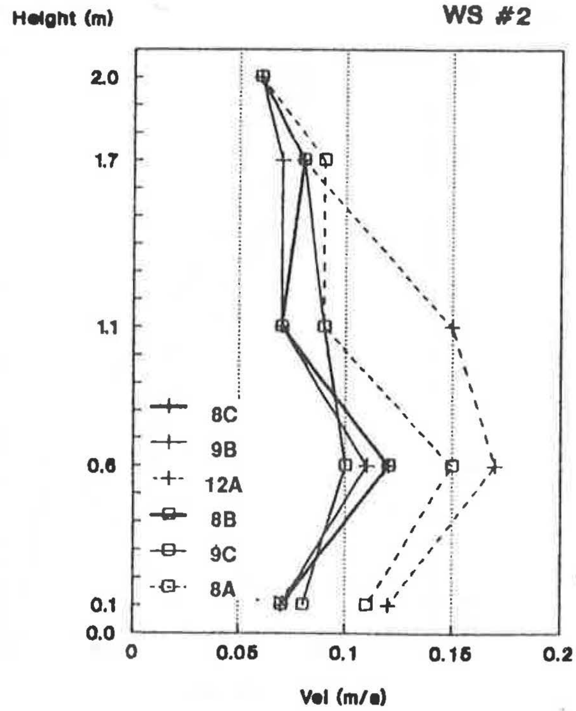


Figure 6 *Solid partition height: velocity effects.*

1. The largest differences between tests occur in WS #1, due to its proximity to the supply diffuser. Within WS #1, the no-partition test shows the highest velocities at all measurement heights, although the differences are only significant at the 0.1-m and perhaps the 0.6-m levels (for comparison, see Figure 5a). The next highest air velocities at these same two heights occurred for 75-in. partitions and decreased with decreasing partition height to their minimum values for 42-in. partitions. The upward entrainment of air by the overhead supply diffuser, combined with the buoyancy-driven airflow produced by the high heat loads within the partitioned workstation, generated these characteristic velocities.
2. In WS #2, the no-partition test again seems to have the highest overall velocities, although this result is not as significant as it was in WS #1. Velocity differences caused by partition height effects are quite small and follow no observable pattern.
3. In WS #3, velocity differences between all four tests





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Part. Ht. (in.)	Part. Air Gap
8C	289	18.0	25.5	65	Solid
9B	292	18.0	24.5	65	4-in.
12A	320	19.3	24.3	65	Open
8B	288	17.9	25.2	75	Solid
9C	293	18.0	24.9	75	4-in.
8A	288	18.2	25.0	75	Open

*For detailed test conditions, see Table 1

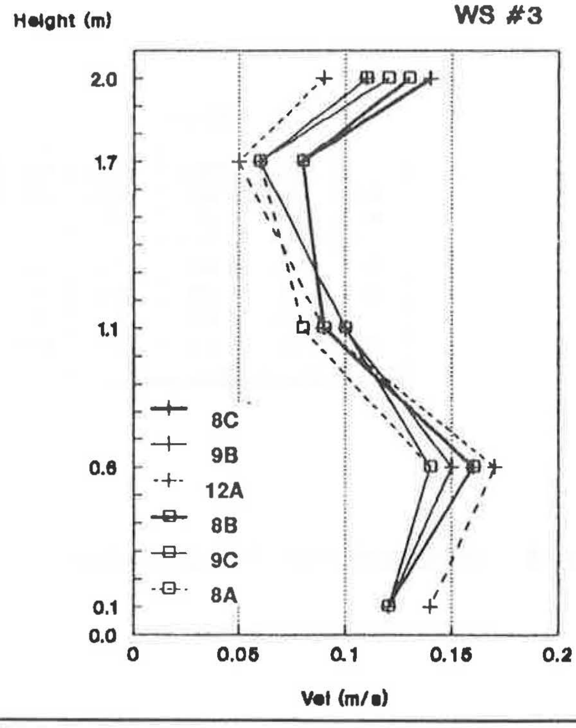
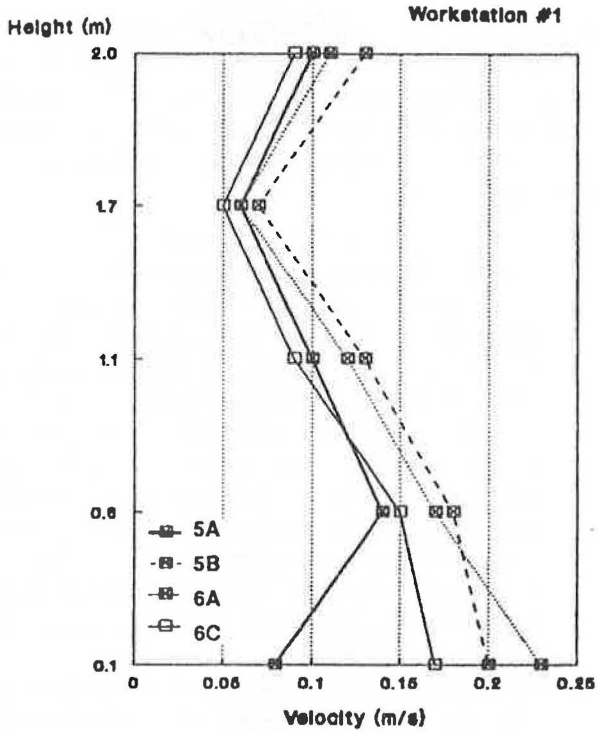


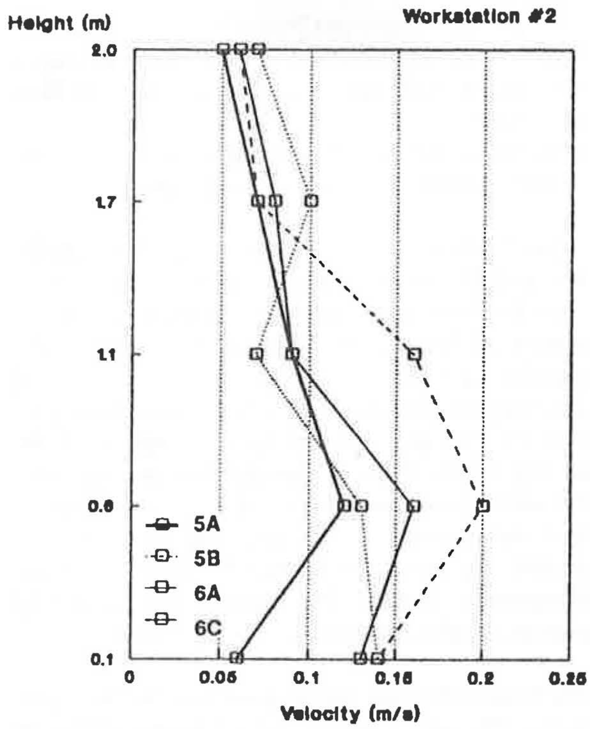
Figure 7a Partition height and gap size: velocity effects.

are insignificant (compare with Figure 5a). This result is not surprising, as the magnitude of partition effects should diminish with increasing workstation size, approaching, in the limiting case, air movement conditions found with no partitions present.

4. In the majority of tests, measured velocities at the 0.6-m level are greater than or equal to velocities near the floor (0.1 m), presumably due to buoyancy effects from the floor heaters located under the desks.

Partition Height and Gap Size Figure 7a presents average velocity results for tests 8A, 8B, 8C, 9B, 9C, and 12A, comparing the effects of solid and airflow partitions for both 65-in. and 75-in. heights under base-case conditions. As previously described, the tall airflow partitions were formed by placing 10-in. extension panels on top of the 65-in. airflow partitions (see Figure 2). Two gap sizes were investigated: (1) full open and (2) a 4-in. gap located above the steel cross-member of the partition—in this case, the 2-in. gap between the floor and the cross-member (typically used for the electrical powerway) was covered. The observations are as follows:





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Part. Air Gap
5A	319	15.3	23.6	Solid
5B	319	13.3	23.2	Open
6A	300	13.8	22.0	Comb.
6C	296	13.3	22.0	Solid

*For detailed test conditions, see Table 1

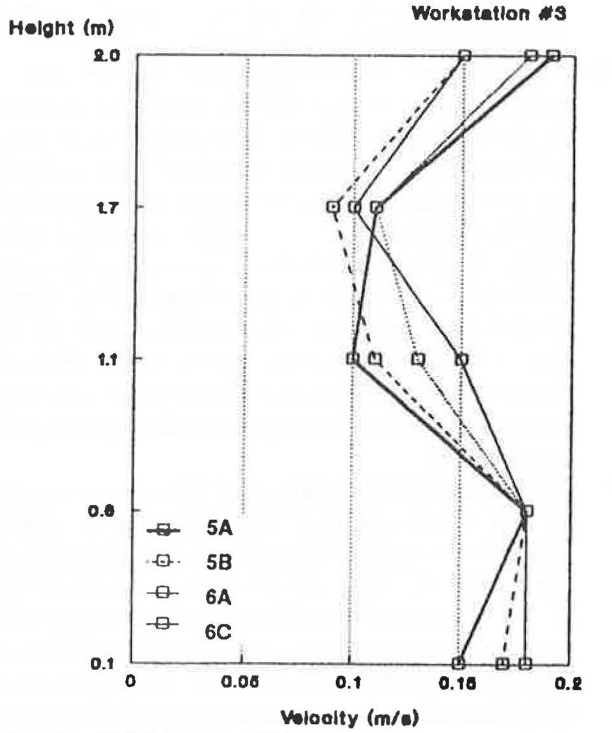


Figure 7b *Partition air gap: velocity effects by workstation.*

1. Overall, the results show only small differences in velocities between solid and airflow partitions, and in most instances, the measured differences are experimentally insignificant.
2. The largest observed effects occur at the 0.1-m level in WS #1 for 75-in. partitions and at the 1.1-m level in WS #2 for 65-in. partitions. For these cases, a 70% to 100%, or 0.08 m/s (16 fpm), increase in velocity was obtained between solid and airflow partitions. Even so, the velocities for solid partitions were so low that all velocities from all tests were still within the comfort zone limits specified by ASHRAE (1981).
3. Except for results for the 65-in. airflow partition in WS #2, measured velocity differences are insignificant at the 0.6-m level and above. This result supports the conclusion that in most instances, airflow partitions appear to provide little or no comfort benefits to an office worker, who will be most sensitive to changes in velocity at the head level (1.1

m). Overall comfort results are discussed later.

4. There are no identifiable effects of the 4-in. gap in comparison with the solid partition for all three workstations.
5. In the large majority of cases, there are no differences between 65-in. and 75-in. partitions.

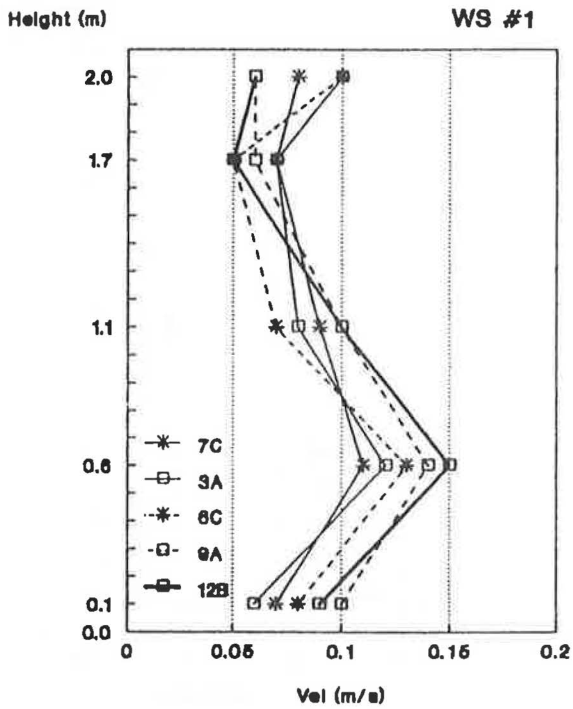
Figure 7b shows average velocity results for tests 5A, 5B, 6A, and 6B. All test conditions were the same as those for the base case, except for a lower supply air temperature of 13 to 15°C. Four different airflow gap configurations for the medium height (65-in.) partitions were investigated: (1) solid; (2) open (12-in. airflow gap); (3) 2-in. airflow gap, formed by covering all of the airflow gap except the 2-in. opening between the floor and the steel cross-member; and (4) a combination, in which all airflow gaps are full open except in the two 60-in. partitions separating WS #1 and WS #2 from WS #3, which remained covered. The results are organized by workstation and the observations are as follows:

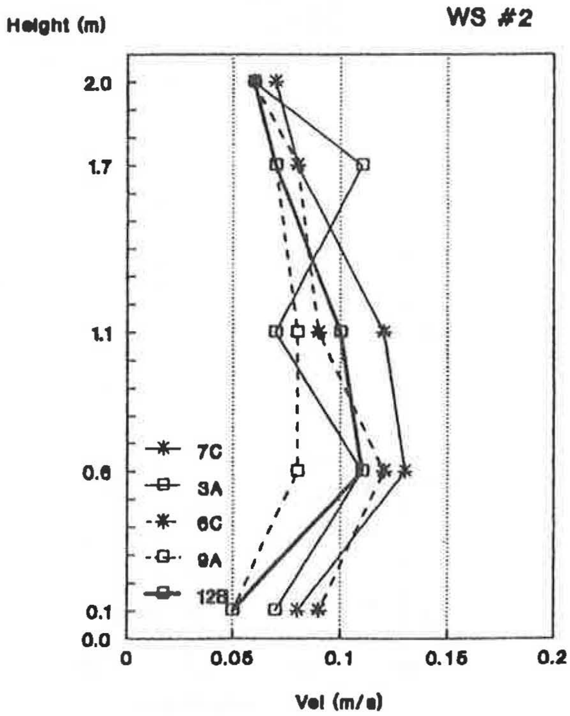
1. The largest observed effect occurs near the floor level in WS #1, where an increase of nearly 200%, or 0.15 m/s (30 fpm), was obtained at the 0.1-m level between the solid and combination partitions.
2. The floor-level velocity differences due to the airflow partition are greatest in WS #1, which is the closest to the supply air diffuser. The magnitude of these differences is also greater than that found in Figure 7a, due in part to the colder supply air temperatures used in tests 5 and 6. Colder supply air temperatures (larger supply air/room air temperature differences) increase the movement of air down to the floor level. Despite the three-way blow configuration of the supply diffuser, some of the supply air was observed to move down the adjacent wall and window surface, across the aisle, and through the airflow gap into the workstation.
3. The slight increase in air motion occurring at the 0.6-m and 1.1-m levels in WS #2 for open airflow partitions is nearly identical to that observed in Figure 7a. This effect is not seen in the larger WS #3, so it appears that airflow partitions may provide small increases in velocity within small workstations. The precise relationship between the magnitude of this effect and distance to the supply diffuser requires further investigation.
4. The smaller 2-in. airflow gap provides the same trends in air movement as the larger 12-in. airflow gap.
5. The larger workstation (WS #3) has slightly higher average air velocities than WS #1 and WS #2 under most test conditions. The one exception is the open airflow partitions for which no discernible difference among the three workstations is observed.

Supply Air Volume and Temperature Figures 8a and 8b present average velocity and temperature results, respectively, for five different combinations of supply air volume and temperature (tests 3A, 6C, 7C, 9A, and 12B). Solid 65-in. partitions, high heat load density, and the base-case diffuser location were used in all five tests. The observations are as follows:

1. In WS #1, there is only a small overall effect on velocity with the maximum difference between all test results at all measurement heights being no greater than 0.04 m/s (8 fpm). At the lower measurement heights, tests using higher air supply volumes (6C, 9A, and 12B) provide slightly higher velocities.
2. In WS #2 and WS #3, the measured velocity differences are also quite small. In most cases, the highest velocities at the lower measurement heights (0.1 to 1.1 m) are provided by the two tests using the lower supply air temperature (tests 6C and 7C). This result is consistent with the improved buoyancy-driven mixing that should result for larger supply/room temperature differences.
3. The ceiling diffuser does a good job of mixing the room air, with no significant stratification measured in any workstation for all test configurations.
4. Temperature differences between tests demonstrate a consistent pattern in all three workstations, with a 3 to 3.5°C (5.5 to 6.5°F) difference between the maximum and minimum results.
5. The lowest temperatures are provided by a high supply air volume combined with a low supply air temperature (test 6C), while the highest temperatures are provided by a low supply air volume combined with a high supply air temperature (test 3A). The other tests (high supply volume with high supply temperature and low supply volume with low supply temperature) produce intermediate temperature results of similar magnitude.

Diffuser Location Figure 9 shows velocity results for a series of tests investigating the effects of four alternative supply diffuser locations for both solid and airflow partitions. Refer to Figure 3 for the diffuser locations tested, which included (1) a single diffuser at (8,5) with three-way blow away from the window (tests 10A and 10B); (2) a single diffuser at (2,8) with two-way blow away from the adjacent corner of the room (tests 11A and 11B); (3) a single diffuser at the base-case position (5,2) with three-way blow away from the window (tests 12A and 12B); and (4) two diffusers at (2,2) and (8,2) with two-way blow away from the adjacent corners (tests 13A and 13B). All tests were performed under base-case conditions, except for changes in the diffuser location. In the figure, results for solid partitions are plotted with a solid line, and results for airflow partitions are plotted with a dashed line. Velocity data for the 2.35-m height are also included due to the strong dependence of airflow near the ceiling on diffuser location. The observations are as follows:





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)
7C	170	13.1	25.4
3A	157	18.0	26.4
6C	296	13.3	22.0
9A	291	18.3	24.3
12B	320	19.5	24.8

*For detailed test conditions, see Table 1

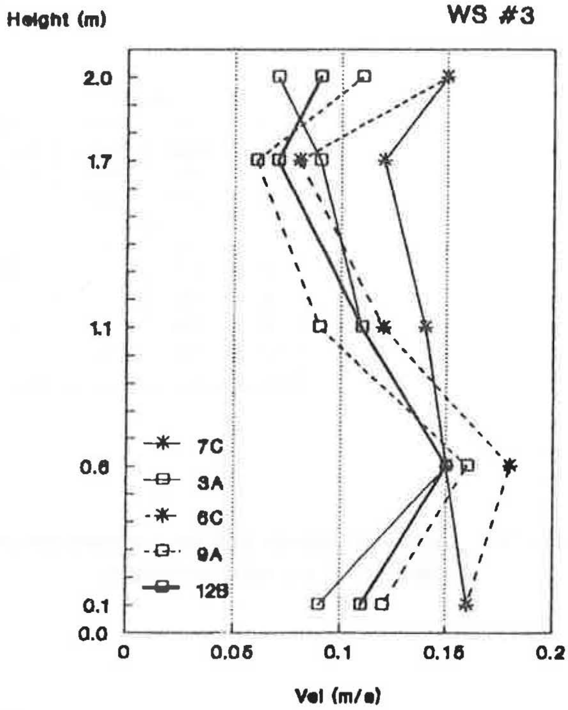
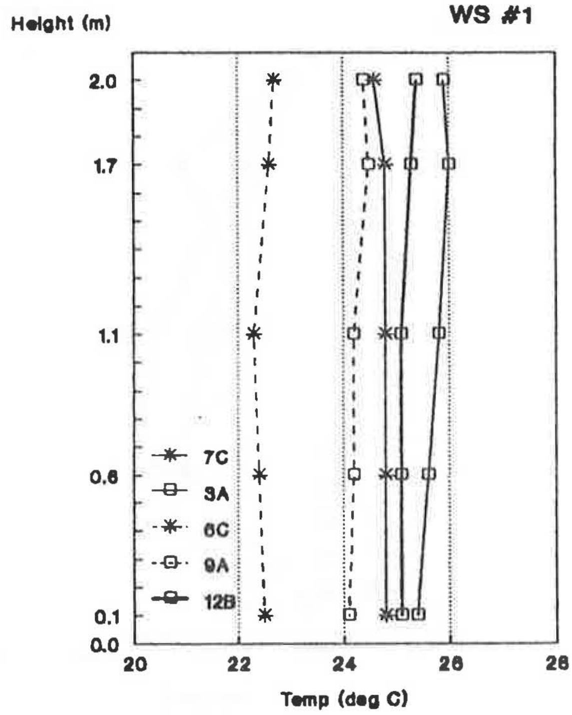
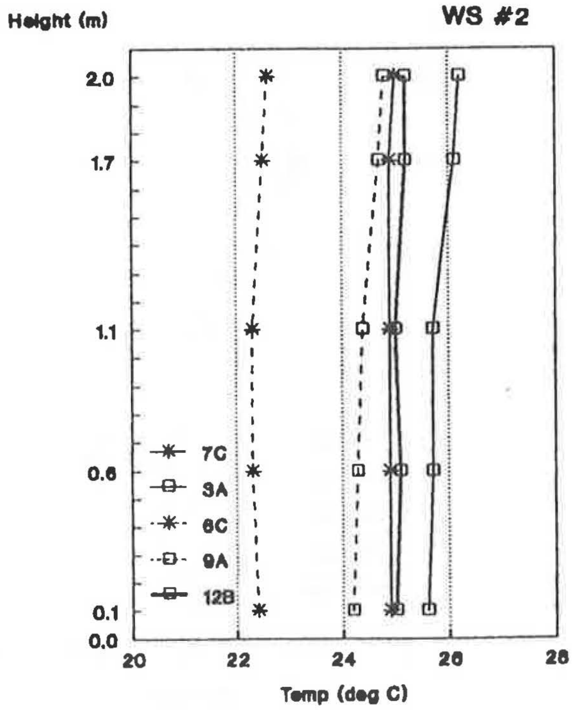


Figure 8a *Supply air volume and temperature: velocity effects for solid partitions.*

1. In WS #1 for test 10, significantly higher velocities (0.3 to 0.35 m/s [60 to 70 fpm]) are obtained at the 1.1-m height. The absence or presence of airflow partitions has no influence on the nature of these results.
2. Although smaller in magnitude, higher velocities are observed at lower heights (0.6 and 1.1 m) in WS #2 during test 12A. In test 12, however, higher velocities at these locations are only observed for airflow partitions (test 12A) and not solid partitions (test 12B).
3. In test 11, the supply diffuser was positioned in the corner of the test room near WS #2. As expected, high air velocities (0.5 m/s [100 fpm]) are obtained near the ceiling in WS #2. No significant differences between solid and airflow partitions are observed in all three workstations. This observation is important because it is contrary to previously obtained results for the base-case test configuration in which the largest partition effects were found in WS #1 due to its proximity to the supply location. The closeness of the return register to the supply location may account





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)
7C	170	13.1	25.4
3A	157	18.0	26.4
6C	296	13.3	22.0
9A	291	18.3	24.3
12B	320	19.5	24.8

*For detailed test conditions, see Table 1

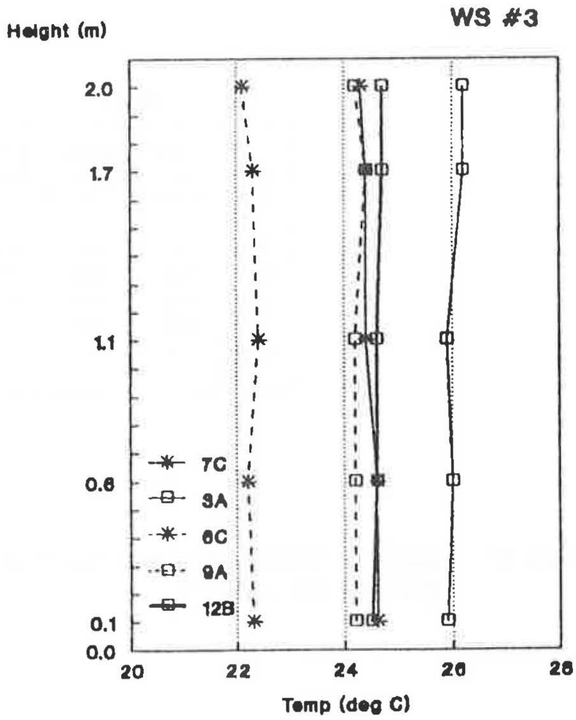
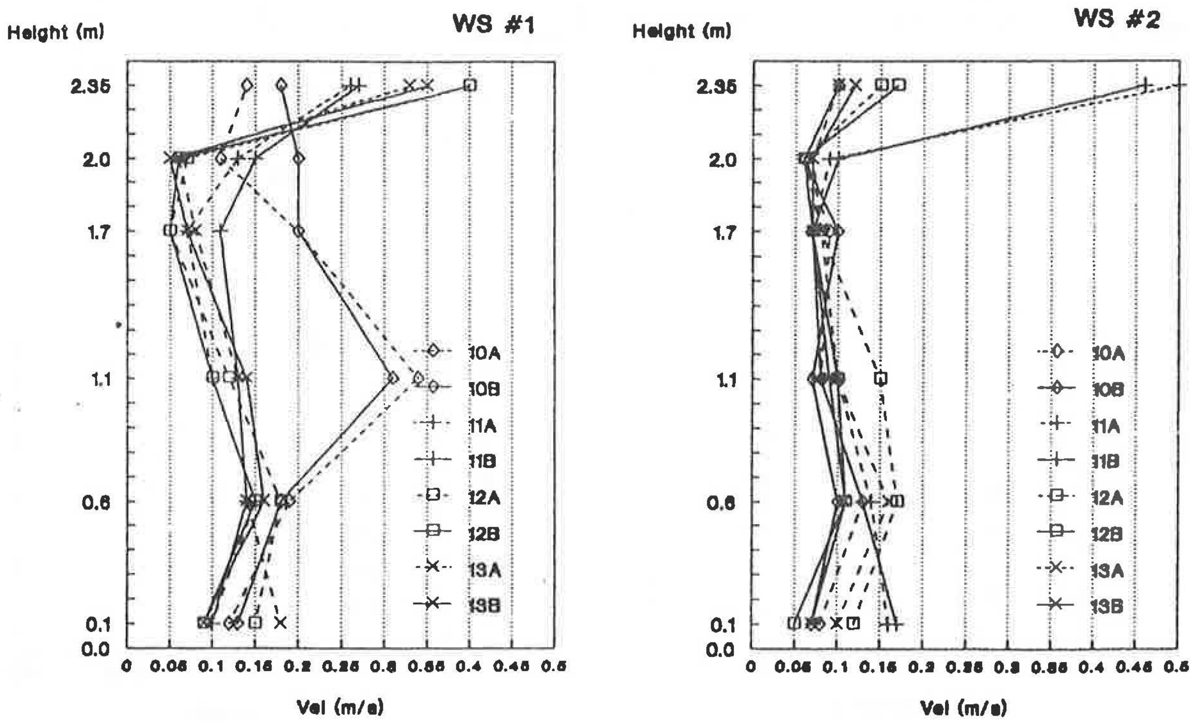


Figure 8b *Supply air volume and temperature: temperature effects for solid partitions.*

for some of the differences found in test 11.

4. In test 13, with two supply diffusers, the highest ceiling-level velocities are obtained in WS #1 and WS #3, the two closest workstations. The airflow partitions have a relatively minor effect in all workstations, except at the 0.1-m level in WS #1, where a 100%, or 0.09 m/s (18 fpm), increase is measured.
5. Overall, WS #2 is the only workstation with a consistent pattern of slightly elevated air velocities in the seated occupant zone (0.1 to 1.1 m) for airflow partitions compared to solid partitions. The one exception to this pattern was obtained when the supply diffuser was positioned in the corner near WS #2.
6. There is no distinct pattern of airflow vs. solid partition effects in WS #1 and WS #3.

Heat Load Density Figures 10a and 10b present average velocity and temperature results for tests 1A, 1B, 4A, and 4B. Test 1 was performed with high heat loads



Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Diff. Loc.†	Part. Air Gap
10A	282	18.1	24.6	(8,5)	Open
10B	282	17.9	24.7	(8,5)	Solid
11A	290	18.1	23.5	(2,8)	Open
11B	291	18.0	23.5	(2,8)	Solid
12A	320	19.3	24.3	(5,2)	Open
12B	320	19.5	24.8	(5,2)	Solid
13A	309	17.9	24.4	(2,2), (8,2)	Open
13B	309	18.0	24.6	(2,2), (8,2)	Solid

*For detailed test conditions, see Table 1
†See Figure 3

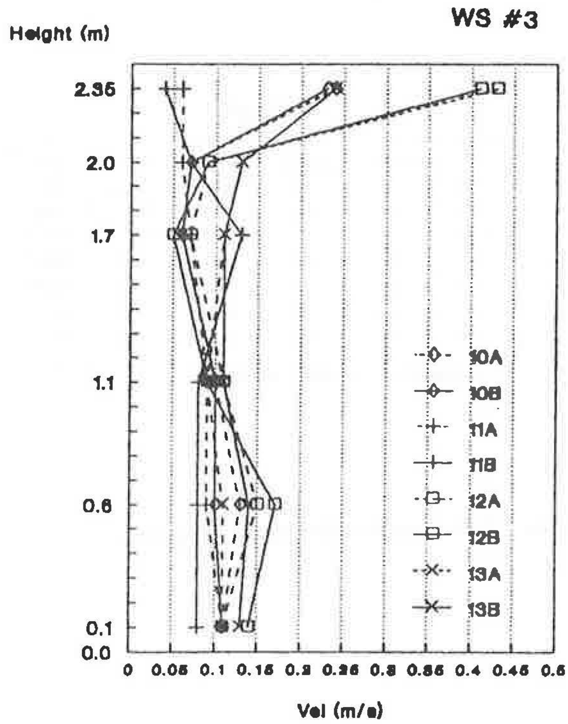
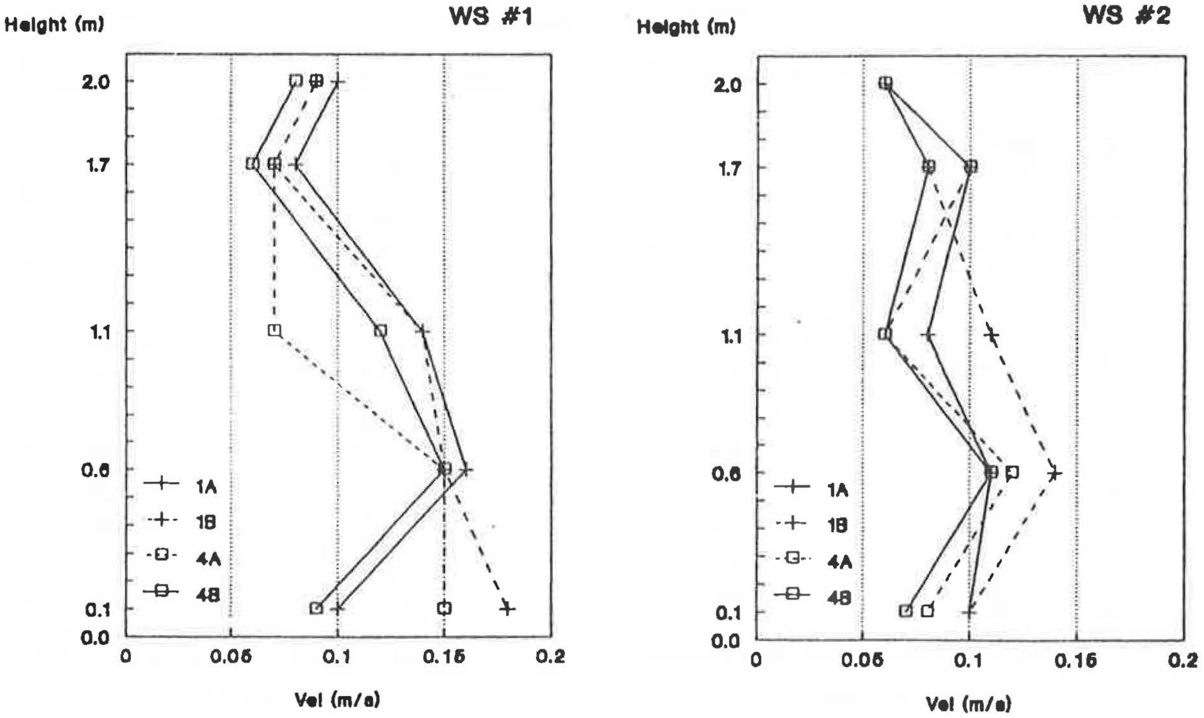


Figure 9   Diffuser location: velocity effects.

(55 W/m^2) and Test 4 with low heat loads (35 W/m^2). Both tests were performed at the lower supply volume (0.5 cfm/ft^2) and lower supply temperature (13°C). As above, results for solid partitions are plotted with a solid line, and results for airflow partitions are plotted with a dashed line. The observations are as follows:

1. In Figure 10a, there are no observable heat load effects, except at the 0.6-m and 1.1-m levels in WS #3, where slight increases in velocity (0.03 to 0.08 m/s [6 to 16 fpm]) are obtained for the higher heat load. These differences are greatest (100% increase) when comparing solid partition test results.
2. The existence of airflow partitions has no noticeable effect, except at the 0.1-m level in WS #1, where small velocity increases occur, similar to previously discussed results.
3. The effects of heat load density on temperature are clearly evident in Figure 10b, where a 2°C (4°F) temperature difference exists between tests 1 and 4.



Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Heat Load (W/m^2)	Part. Air Gap
1A	152	13.1	24.2	55	Solid
1B	152	13.0	24.2	55	Open
4A	155	13.2	21.9	35	Open
4B	154	13.3	22.1	35	Solid

*For detailed test conditions, see Table 1

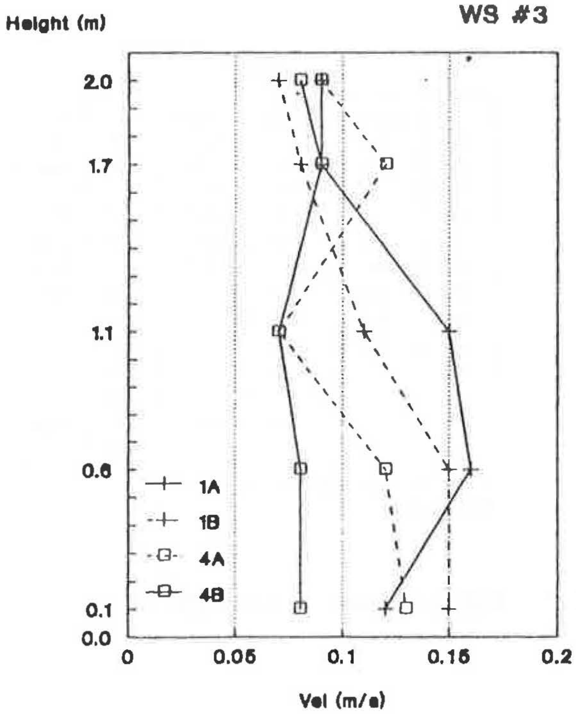
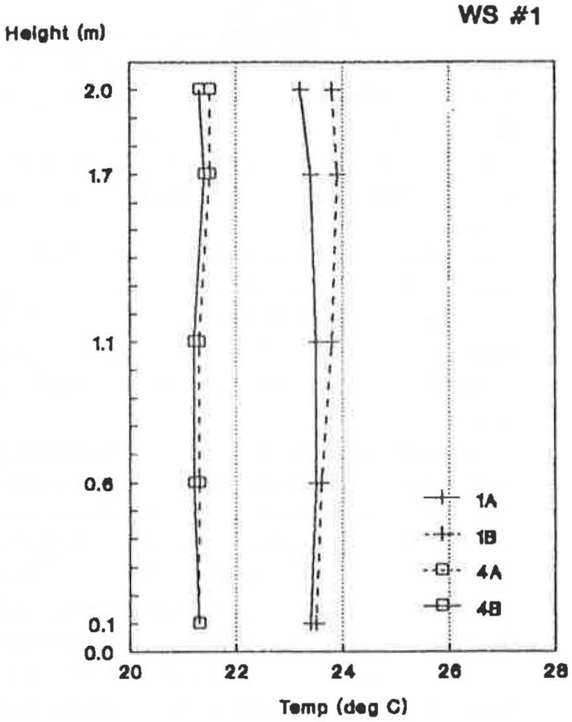


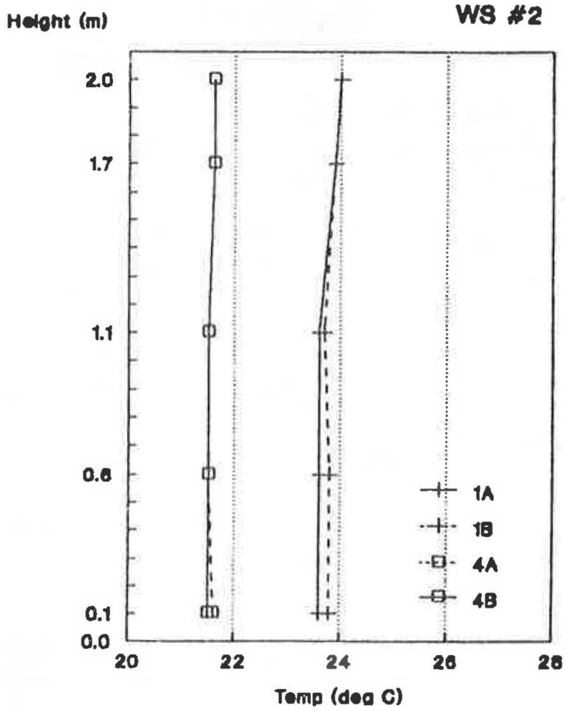
Figure 10a *Heat load density: velocity effects.*

There is no measurable stratification in the room, as previously observed. The existence of airflow partitions has a negligible effect on temperature distributions in all workstations.

Air Diffusion Performance Index ASHRAE (1990) provides a method for evaluating the ability of an air distribution system to produce an acceptable thermal environment, based on air motion and air temperature distribution. The air diffusion performance index (ADPI) is a calculated quantity representing the percentage of measurement locations where velocities and temperatures meet certain criteria in terms of magnitude and uniformity. The air diffusion performance of a system is considered acceptable when an ADPI of 80% or greater is obtained.

Table 3 presents calculated ADPI results for 17 tests selected to cover the full range of test conditions. Eighty points within the three workstations were used for each calculation, consisting of the four heights in the occupied





Key*

Test No.	Supply Vol. (cfm)	Supply Temp. (°C)	Room Temp. (°C)	Heat Load (W/m²)	Part. Air Gap
1A	152	13.1	24.2	55	Solid
1B	152	13.0	24.2	55	Open
4A	155	13.2	21.9	35	Open
4B	154	13.3	22.1	35	Solid

*For detailed test conditions, see Table 1

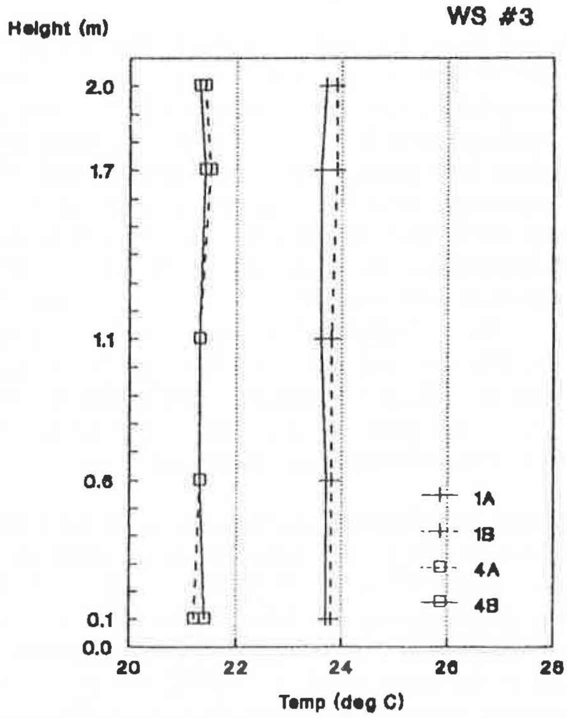


Figure 10b Heat load density: temperature effects.

zone (0.1 to 1.7 m) for location numbers 1-20, as shown in Figure 4a (four locations each in WS #1 and WS #2 and 12 locations in WS #3). The test conditions covered were low (5 to 6 cfm/m^2) and high (10 to 11 cfm/m^2) supply air volume (SAV); low (35 W/m^2) and high (55 W/m^2) heat load density; low (13 to 15°C) and high (18°C) supply air temperature (SAT); no, 42-in., 65-in., and 75-in. partitions (Part); and solid (S) and full open airflow (O) partitions.

The air diffusion performance for all tests is quite acceptable, as calculated ADPI values ranged from 89% to 99%. The largest difference between solid and airflow partitions occurred in test 8, for which the ADPI increased from 90% to 98% for the solid and airflow partitions, respectively. This difference is not considered significant due to the already excellent ADPI for the solid partitions.

Thermal Comfort The Fobelets and Gagge (1988) two-node comfort model was used to predict the characteristic comfort indices for a selected number of tests. The model accounts for the combined effects of air temperature, air velocity, mean radiant temperature, relative

TABLE 3
Air Diffusion Performance Index (ADPI) Results

SAV (cfm/sqm)	Load (w/sqm)	SAT (degC)	Part (in.)	Gap	Test	ADPI
5-6	35	13	65	S	4B	95%
				O	4A	99%
	55	13	42	S	16	94%
			65	S	7C	93%
				O	1B	94%
			75	S	7A	99%
				O	7B	96%
		18	65	S	3A	90%
				O	3B	89%
10-11	55	13-15	65	S	5A	90%
				O	5B	90%
		18	0	-	14	95%
			42	S	15	89%
			65	S	8C	91%
				O	12A	91%
			75	S	8B	90%
				O	8A	98%

humidity, clothing level, and activity level. The measured data used as input to the model consisted of (1) air temperature and velocity data averaged for the 0.1-, 0.6-, and 1.1-m levels directly in front of each desk, representing a whole-body average for a seated worker, and (2) globe temperature measured at the 1.1-m level near the front edge of the desk, allowing the calculation of mean radiant temperature. The other three inputs to the model were assumed constant values, representing typical conditions for sedentary office work: 50% relative humidity, 0.5 clo, and 1.2 met. The comfort model predictions for effective temperature (ET^*), discomfort index (DISC), and predicted mean vote (PMV) are listed in Table 4. The observations are as follows:

1. Thermal conditions in all three workstations for tests 11 and 12 (65-in. partitions) are at or above the upper limit (ET^* = 26°C [79°F]) of the comfort zone, as specified by ASHRAE (1981). These results were obtained despite the maintenance of air temperatures in the range of 23.5 to 25°C (74 to 77°F) within all workstations during these tests. The uncomfortably warm comfort predictions reflect the significant impact of the high heat load levels on radiant temperatures in the workstations.
2. For tests 11 and 12, the most acceptable comfort conditions are obtained in the larger WS #3. Each workstation contains the same magnitude of heat sources, thus generating higher mean radiant temperatures within the smaller WS #1 and WS #2.
3. No comfort improvements are predicted for airflow partitions (tests 11A and 12A) in comparison with solid partitions (tests 11B and 12B). This is an important result because during test 12, the largest overall velocity increases due to airflow partitions were recorded. Measured velocities at the 0.1- to 1.1-m heights in WS #2 were 0.05 to 0.07 m/s (10 to 14 fpm) higher (100% increase) for airflow partitions than for solid partitions (see Figure 9). Despite the improved air motion, no significant comfort benefits were predicted.
4. As expected, comfort predictions are identical in all three workstations when no partitions are present (test 14). The absence of the heat-absorbing partitions also reduces the mean radiant temperatures, producing thermal conditions within the ASHRAE comfort zone.
5. A comparison of results for tests 15 and 16 (42-in. partitions) demonstrates the important influence of supply air temperature. Except in WS #3, conditions are too warm during test 15, which had a high supply air volume and temperature. Test 16, however, produces considerably improved comfort conditions, even though it was performed with a supply air volume about half that used in test 15. The major reason for this improvement is the cooler supply air temperature (13°C [55°F]) in test 16. WS #1 experiences perfectly neutral comfort conditions (ET^* = 24°C, DISC = 0.0, PMV = −0.02) and WS #3 is also predicted to be quite comfortable. WS #2 is

TABLE 4
Comfort Model Results

Test No.	WS No.	ET^* (deg C)	DISC	PMV
11A	1	26.4	0.55	0.56
	2	26.1	0.48	0.50
	3	25.8	0.40	0.41
11B	1	25.9	0.41	0.43
	2	25.9	0.41	0.43
	3	25.7	0.39	0.39
12A	1	26.6	0.60	0.61
	2	27.2	0.77	0.77
	3	25.5	0.33	0.33
12B	1	26.5	0.59	0.60
	2	27.1	0.70	0.75
	3	25.6	0.36	0.36
14	1	25.4	0.32	0.31
	2	25.4	0.31	0.31
	3	25.4	0.31	0.31
15	1	26.3	0.53	0.54
	2	26.7	0.60	0.63
	3	25.3	0.28	0.28
16	1	24.0	0.00	-0.02
	2	25.6	0.33	0.35
	3	24.5	0.08	0.08

noticeably warmer, indicating that the lower supply air volume may have trouble adequately conditioning small workstations located further away from the supply diffuser.

Tracer Gas Measurements

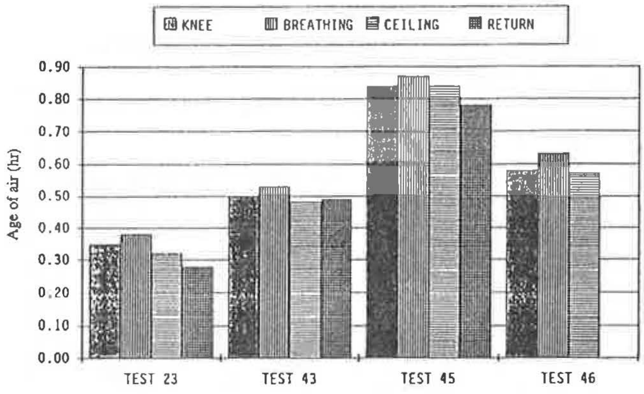
Data Precision The precision of the tracer data is indicated by data from two tests in which the air in the chamber was well mixed and by the repeatability of data from tests run at the same experimental conditions.

During tests 22W and 42W, the chamber air was vigorously mixed with fans, which ideally should produce the same local age of air at every point. Consequently, all age-of-air ratios from these tests should have a value of unity. The coefficients of variation of the measured ages of air (standard deviation divided by the average expressed as a percentage) are 3.1% and 5.3% for tests 22W and 42W, respectively. We assume that errors in measurement of age of air are normally distributed and use twice the largest coefficient of variation, or $\pm 11\%$, as a 95% confidence interval. Consequently, our estimate of the precision of age-of-air measurements is $\pm 11\%$, and smaller differences between two ages are not considered significant.

Tests 21 and 22 are comparable tests (run at the same conditions) and produced local ages at all but three locations that were within $\pm 11\%$ of each other. For reasons that are not apparent, test 21 contains data at two points that are suspect. Therefore, data from test 22 are used in the subsequent discussion for comparison to data of other tests.

We believe that at least three factors cause imprecision in the multiple (multi-point) measurements of age of air. First, there is a small bias between ages determined from numerical integration of real-time data and the bag and syringe samples. We are investigating the cause of this bias. Second, the air in the CEC was probably not perfectly mixed due to the presence of internal partitions. Third, there is undoubtedly some random error in the measured ages due to such factors as instrument imprecision. When we gain more experience and data, a statistical evaluation of measurement precision may become appropriate.

Height Variation in the Age of Air Table 5 lists average age-of-air values for the workstations at the knee level and breathing level and near the ceiling level and the return duct. A consistent increase or decrease in the age of air above the floor would be an indication of a general upward or downward airflow pattern. Only in test 21 (with some suspect data) and test 40 do the data indicate a consistent trend in age with height (excluding the age at the return duct). These trends are not significant, since the average age at the breathing level is nearly identical to the average age at the ceiling level. All other cooling tests show no consistent pattern of age-of-air variation with height.



Average age at the knee level for all workstations, average age at the breathing level for all workstations, average age near the ceiling level above all workstations, and the age in the return duct. Ages are grouped by heating tests.

Figure 11 Variation in age of air with height.

Tests 23, 43, 45, and 46 in the heating mode indicate a consistent pattern of age of air with height; however, the variations in age are small (see Figure 11). The ordering of the age of air is, from the lowest to highest, ceiling level, knee level, and breathing level, with a maximum percent difference between two levels of 19% (test 23, from 0.32 hour to 0.38 hour). Since these ages are averages of measurements at several locations, differences greater than 5% are considered significant from the perspective of measurement precision. (In the well-mixed tests, these averages of several ages of air differed by no more than 5%). We have no explanation for this type of pattern in age of air with height.

We have seen no pattern in the age of air with height that is dependent upon the partition height or the presence of a partition gap at the base of the partitions.

TABLE 5
Averages of Local Ages of Air (in hours) for Different Heights

Test Number	Knee (0.4m)	Breathing (1.1m)	Ceiling (2.1m)	Return
21	0.45	0.40	0.39	0.43
22W	0.46	0.44	0.44	0.45
22	0.40	0.39	0.40	0.44
23	0.35	0.38	0.32	0.28
24	0.43	0.41	0.44	0.43
25	0.45	0.39	0.43	0.44
39	0.27	0.27	0.27	0.30
40	0.25	0.26	0.27	0.29
41	0.79	0.78	0.86	0.88
42W	0.50	0.50	0.52	0.51
43	0.50	0.53	0.48	0.49
45	0.84	0.87	0.84	0.78
46	0.58	0.63	0.57	na

TABLE 6
Age of Air Ratios Related to Short-Circuiting

Test Number	Return / All WS	WS Ceiling / All WS	Aisle Ceiling / Aisle Breathing
21	1.03	0.93	0.98
22W	1.01	1.00	1.01
22	1.13	1.03	0.98
23*	0.78	0.89	0.79
24	1.02	1.05	1.00
25	1.06	1.05	0.96
39	1.08	1.00	0.99
40	1.14	1.04	0.90
41	1.12	1.09	0.93
42W	1.03	1.04	0.95
43*	0.96	0.92	1.03
45*	0.91	0.98	1.00
46*	na	0.95	na

Return: Age in return duct
All WS: Average age of all breathing level (1.1m above floor) and knee level (0.4m above the floor) points in the workstations
WS Ceiling: Average age of all ceiling level (2.1m above floor) points above the workstations
Aisle Ceiling: Location 6 in Figure 6, 2.1m above floor
*Heating test

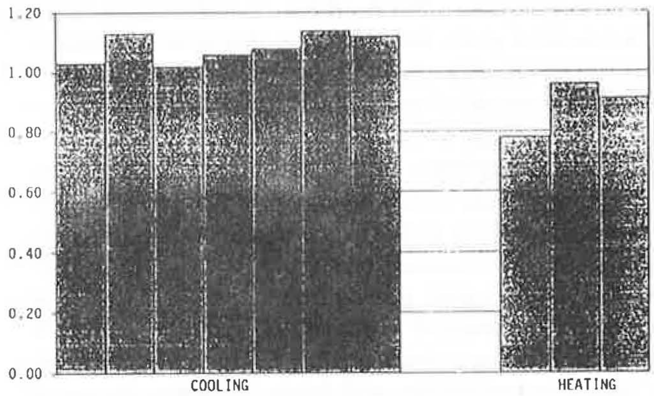
Short-Circuiting Short-circuiting of supply air, for example, air that does not enter the occupied space but travels preferentially to the return, would be evident by ages of air near the ceiling or return duct being lower than ages in the occupied space. Thus, ratios of ages of air near the ceiling or the return to ages of air at the breathing level and knee level should be less than unity for short-circuiting (see Table 6). The values in Table 6 are ratios; therefore, our previous measure of data precision is not applicable. Using our precision for an age-of-air measurement of $\pm 11\%$ for a single point, we calculated the precision of each average value used in the ratios. Using propagation of error analysis (Schenck 1979), we combined the precision values to determine the estimated precisions for each ratio. Thus, the resulting estimated measurement precision for the ratios within the three columns of Table 6 are: ± 0.12, ± 0.06, and ± 0.16, respectively. These estimates of precision should be used to judge whether any value of a ratio is significantly different from unity or whether any two ratio values are significantly different. Ratios from heating tests (43, 45, and 46) indicate slight short-circuiting, and the ratios for test 23 indicate significant short-circuiting (see Figure 12). Test 23 had the supply diffuser closer to the return than tests 43, 45, and 46, thus allowing a shorter path for short-circuiting; however, because of the limited data, we cannot confirm that the diffuser location was a cause of increased short-circuiting.

Short-circuiting is not evident from the age of air in the aisle for tests 43, 45, and 46 in which the supply diffuser was relatively far from the aisle. The reason is not clear. Data for test 21 (a cooling test) yield one ratio less than unity, but this result is not consistent with the data from similar tests. Therefore, short-circuiting is only evident in the heating tests.

Workstation Ventilation Uniformity The preferential ventilation of one workstation over another is of concern to some with regard to partitioned workstations. We have seen no indication of significant preferential ventilation in the configurations tested thus far in the CEC. In Table 7, the maximum difference in ages of air between workstations is less than 20% (test 45, from 0.80 h to 0.94 h, not including test 21, with suspect data, at 24%). With the supply diffuser moved from a central location to a location farther away from WS #2 and the aisle, there is consistently a slightly higher age of air in WS #2 and the aisle. This relatively higher age of air occurs in WS #2 during both heating and cooling tests. The aisle has a slightly higher age only during cooling tests.

CONCLUSIONS

Measurements were made of the thermal and ventilation performance of a conventional overhead ducted supply-and-return air distribution system in an office environment. The experiments were performed in a controlled environment chamber configured to resemble an open-plan modern office building with modular workstation furniture and partitions. Tests were conducted to investigate the effects of partition height, solid vs. airflow partitions, airflow gap size, supply air volume, room/supply temperature difference, supply diffuser location, heat load density, workstation size, and cooling vs. heating mode. The major conclusions are as follows:



Age of air in the return duct divided by the average age of air of all workstations at the breathing level and knee level. Ratios from heating tests and cooling tests are in separate groups. Tests with mixing fans operating are not included.

Figure 12 Short-circuiting.

TABLE 7
Averages of Local Ages of Air (in hours) for Each Workstation and Aisle

Test Number	WS 1*	WS 2*	WS 3*	Aisle**
21	0.38	0.47	0.42	0.45
22W	0.45	0.43	0.45	0.44
22	0.38	0.39	0.39	0.42
23	0.35	0.35	0.37	0.38
24	0.41	0.42	0.43	0.44
25	0.39	0.44	0.41	0.55
39	0.28	0.30	0.26	0.30
40	0.25	0.28	0.25	0.30
41	0.74	0.83	0.79	0.89
42W	0.52	0.52	0.49	0.52
43	0.53	0.55	0.49	0.50
45	0.89	0.94	0.80	0.81
46	na	0.64	0.58	0.58

*Averages for WS are for breathing level (1.1m above floor) and knee level (0.4m above floor)
**Averages for aisle are for the breathing level

1. Variations in solid partition height produced only small differences in overall thermal and ventilation performance, although some nonuniformities existed in comparison to an office without partitions. Partition effects, if any, were strongly dependent on the heat loads within the workstation and the proximity of the supply diffuser.
2. For similar test conditions, only small differences in workstation velocities between solid and airflow partitions were obtained, and in the large majority of cases, the measured differences were experimentally insignificant.
3. When one of the largest overall velocity increases due to airflow partitions was recorded (100% increase in WS #2 during test 12), comfort model predictions indicated no improvement in comfort conditions for airflow vs. solid partitions. Therefore, airflow partitions appear to provide no significant comfort benefits to an office worker for the range of conditions investigated.
4. Except for only a few isolated data points, measured velocities at all locations within the occupied zone (0.1 to 1.7 m) for all tests were within the acceptable summer limits specified by ASHRAE (1981) ($\leq$ 0.25 m/s [50 fpm]). It is not surprising that changes in velocity at this relatively low range have little effect on overall comfort conditions.
5. The air diffusion performance of the overhead supply-and-return system was quite acceptable and essentially identical for 17 tests selected to cover the full range of test conditions. Calculated air diffusion performance index (ADPI) values ranged from 89% to 99%.
6. Heat loads in partitioned workstations had a significant effect on air temperatures, mean radiant temperatures, and overall comfort conditions. As heat load density (W/m^2) increases (or the workstation size decreases for the same heat load level), thermal conditions will become increasingly warm and uncomfortable, unless other means, such as increasing the air motion or reducing the supply air temperature, are used to provide additional cooling.
7. The location and throw characteristics of the supply diffuser had a significant effect on air motion in nearby workstations. Cooler supply air temperatures demonstrated improved movement of air down to the floor level.
8. The effect of airflow partitions was not significantly dependent on supply air volume. Tests at 0.5 and 1.0 cfm/ft^2 demonstrated only small differences in measured velocities.
9. The deviations from uniform ventilation (age of air) noted were slight.
10. Short-circuiting of the supply air to the return occurred during heating tests and was absent in cooling tests.
11. Partition height and gaps at the bottom of the partitions had little or no effect on the variation of age of air with height, short-circuiting, or uniformity of workstation ventilation.

Due to the acknowledged strong potential influence of room air distribution on thermal comfort and satisfaction, indoor air quality, and worker productivity, future work is needed to address the following important issues:

- Comparison of field measurements of thermal and ventilation performance in large partitioned offices with test chamber results. In large office spaces, conditions may occur where the predominantly horizontal movement of the bulk air mass between supply and return locations separated by large distances is influenced by obstructing partitions.
- Additional testing of the effects of heat load density and nonuniform load distribution.
- Investigation of the effects of airflow and workstation design on worker productivity.
- Investigation of the impact of occupant control and task conditioning on comfort and satisfaction.
- Development and implementation of detailed room air distribution numerical modeling techniques for addressing the impacts of a wide range of environmental control and workstation design parameters.

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