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A NUMERICAL STUDY OF INDOOR AIR QUALITY AND THERMAL COMFORT UNDER SIX KINDS OF AIR DIFFUSION

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ABSTRACT

Correct air diffusion, as well as the proper quantity of conditioned air, is essential for good air quality and comfortable conditions in forced-ventilation systems. Research has been carried out to study local indoor air quality and thermal comfort in an office under six kinds of air diffusion for summer cooling conditions. The discomfort due to indoor air quality is calculated by the olf and decipol, the units for perceived air quality, and discomfort due to draft is calculated by a comfort equation accounting for turbulence intensity. The distributions of decipol and draft are computed by an airflow computer program based on a low-Reynolds-number k-ε model of turbulence.

It can be concluded that a fresh air change rate of five is required to maintain less than 20% dissatisfaction due to air quality, even though it is a low-olf office. The percentage of dissatisfied people due to draft is smaller than 15% in the occupied zone in all cases. Outlet location and diffuser characteristics are very important for indoor air quality and thermal comfort.

INTRODUCTION

Because up to 90% of a typical person's time is spent indoors and a large fraction of that time is spent in a residential or commercial environment (Moschandreas and Morse 1979), the quality of indoor air is an important component influencing our overall level of health and comfort. During the past decade, indoor air pollution emerged as an international health issue. Increased awareness of the potential health risks associated with indoor air pollutants (Nero 1988; Morris 1986; Brundage et al. 1988) has stimulated interest in improving our understanding of how ventilation air is distributed and how pollutants are transported in buildings.

Pollutant transportation and distribution depend in general upon the ventilation system, building geometry, pollutant source characteristics, and thermo/fluid boundary conditions such as flow rate, the locations of supply outlets and return inlets, and diffuser characteristics. The task of predicting the pollutant transport by ventilation systems is not a simple one. For a certain kind of building geometry and pollutant source, indoor air quality may be improved by increasing the ventilation rate. However, the increase of ventilation results in higher energy consumption and sometimes increased equipment cost. Although recirculated air and air treatment devices can be used, they may not be economically feasible. Therefore, it is necessary to provide a means for investigating air distribution with different locations of supply outlets and return inlets and diffuser characteristics and, thereby, to assess the nature and severity of indoor air quality problems.

Another objective of air diffusion in an air-conditioning system is to create a comfortable thermal environment with the proper combination of comfort variables (ASHRAE 1989b). The comfort variables are metabolic rate, clothing, air velocity, air temperature, air temperature stratification, radiant temperature, radiant temperature asymmetry, relative humidity, and turbulence intensity in the occupied zone (Fanger et al. 1989; Int-Hout 1990). For the same activity level, clothing type, and room geometry, location, and orientation, thermal comfort is related to air velocity and temperature, temperature stratification, relative humidity, and turbulence intensity. With different diffusers and different locations of supply outlets and return inlets, the distribution of the thermal comfort parameters is different. Hence, it is also necessary to study quantitatively the local thermal comfort under different kinds of air diffusion.

The aim of this paper is to analyze different kinds of air diffusion in an office and to evaluate local indoor air quality and thermal comfort by numerical simulation.

RESEARCH METHOD

Model for the Computation of Air Diffusion

Since indoor air quality and thermal comfort are related to space air diffusion, it is necessary to predict the air movement in a given space. Some articles have been published on the experimental investigation of air diffusion (ASHRAE 1989b; Straub et al. 1956; Straub and Chen 1957), but such investigations are very expensive and detailed measured data are not available. The successful development of numerical techniques to solve the basic equations of fluid mechanics has led to the opportunity to compute space air diffusion, as reviewed by Rhodes (1989) and Nielsen (1989). Mathematical models based on these techniques have been used to predict indoor air quality and thermal comfort, and a number of applications have been reported in the last few years (Awbi 1989; Chen et al. 1990b; Haghighat et al. 1990;

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Murakami et al. 1989). With these techniques, it is possible to study the field distributions of air velocity, temperature, turbulence intensity, relative humidity, and contaminant concentration/air quality for different locations of supply outlets and return inlets and diffusers. The results can be used for an overall evaluation of indoor air quality and thermal environment.

In this study, the airflow program developed by Rosten and Spalding (1987) has been employed to calculate air distribution. The computations involve the solution of three-dimensional equations for the conservation of mass, momentum (u, v, w), energy (H), contaminant concentration (C), turbulence energy (k), and the dissipation rate of turbulence energy (ϵ).

The turbulence model used is a low-Reynolds-number k-ϵ model (Chen et al. 1990a) that has been implemented in the airflow program. This model has been verified as being more suitable for indoor airflow simulations, and a better agreement between computation and experiment has been found with respect to velocity and turbulent energy distributions and heat exchange through solid walls (Chen et al. 1990a). The governing equations of the model can be expressed in a standard form:

$$\mathrm{div}(\rho \vec{V} \phi - \Gamma_\phi \,\mathrm{grad}\, \phi) = S_\phi \qquad (1)$$

where ρ is the air density (lb/ft^3 [kg/m^3]), $\vec{V}$ is the air velocity vector (ft/min [m/s]), Γ_ϕ is the diffusive coefficient (lb·min/ft^2 [N·s/m^2]), S_ϕ is the source term of the general fluid property, and ϕ can be any one of 1, u, v, w, k, ϵ, H, or C. When $\phi = 1$, the equation changes into the continuity equation.

With the field distributions of air velocity, temperature, turbulence intensity, relative humidity, and contaminant concentration, we are now able to study local air quality and thermal comfort. "Local" refers to any point in the room. Because in most cases room air is not well mixed, the air quality and thermal comfort at any given point can be different, and an overall parameter for air quality and thermal comfort (the mean value) is not sufficient. It is necessary to assess air quality and thermal comfort at any point in which one may be interested.

Model for the Analysis of Indoor Air Quality

Several methods have been used to evaluate indoor air quality, such as calculation by local age and local purging flow rate (Sandberg and Sjöberg 1983), by displacement efficiency (Anderson 1988), and by perceived indoor air quality (Fanger 1990). For an evaluation of indoor air quality acceptability on the basis of odor for visitors entering a space, the perceived units for air quality, the olf and decipol, are appropriate. One olf is the emission rate of air pollutants (bioeffluents) from a standard person. Any other pollution source is expressed by the number of standard persons (olfs) required to cause the same dissatisfaction as the actual pollution source. One decipol is the pollution caused by one standard person (one olf) ventilated by 21.2 ft^3/min (10 L/s) of unpolluted air.

The olf and decipol units are applied because human senses are more useful than chemical analysis of the air for determining acceptability. The senses involved are the olfactory, which is sensitive to odorous compounds, and the chemical, which is sensitive to irritating compounds in the air. The application of the olf and decipol units allows one to establish a contaminant balance for the air in a room. If the room air is well mixed, the perceived indoor air quality, C_i, can be determined from (Fanger 1990):

$$C_i = C_o + 21.2\, G/Q \quad \text{(decipol) (I-P unit)} \qquad (2a)$$

$$C_i = C_o + 10\, G/Q \quad \text{(decipol) (SI unit)} \qquad (2b)$$

where C_o is perceived outdoor air quality (decipol), G is total odor sources (olf), and Q is the ventilation rate (ft^3/min [L/s]). The percentage of dissatisfied persons in the room, PD, can be calculated by (Fanger 1990):

$$PD = e^{5.98 - (112/C_i)^{0.25}} \ (\%). \qquad (3)$$

In situations where the "well mixed" assumption does not apply, one must determine air quality locally. With the airflow program, the field distribution of perceived indoor air quality can be computed for different kinds of air diffusion. Consequently, the distribution of percentage dissatisfied persons due to air quality can be determined.

It should be noted that the decipol level expresses how the air is perceived by humans, not the possible health risk. Some contaminants, such as radon, are not perceived but represent a risk of lung cancer. There are other potential effects that will not necessarily be corrected with the perceived evaluation of indoor air quality acceptability. Any such effect should be considered separately, for example, by the concentration of contaminant. The field distribution of the concentration can be computed by the airflow program and the corresponding results will also be presented in the paper.

Fanger's method was adapted for this research and is useful, but other methods may also be used. In fact, more research is needed to determine how indoor air quality should be better quantified.

Model for the Analysis of Thermal Comfort

There are a number of thermal comfort models available. The most common and probably best understood unit is predicted mean vote (PMV) for thermal comfort and the associated percent persons dissatisfied (ISO 1984). However, the influence of turbulence intensity is not included in most previous models. Fanger et al. (1989) pointed out that the turbulence of an airflow has a significant impact on the sensation of draft, and a mathematical model of draft risk including turbulence intensity was developed. The model predicts the percentage of people dissatisfied due to draft as a function of mean air velocity, turbulence intensity, and air temperature. Unfortunately, the influence of relative humidity and temperature stratification on comfort is not considered. For the same control strategy of the HVAC system, the distribution of relative humidity would be very similar for different kinds of air distribution. On the other hand, relative humidity can be treated as one of the contaminant sources in the numerical approach. If the temperature difference between head (3.6 ft [1.1 m]) and ankle (0.33 ft [0.1 m]) is small, the model of Fanger et al. (1989) can be used.

In the model, the percentage of dissatisfied people due to draft, PD, is calculated from

$$PD = (51.8 - 0.5556\,T_a)(0.005\,V - 0.05)^{0.62} \quad (3.14 + 0.00188\,VI) \; (\%) \quad \text{(I-P)} \tag{4a}$$

$$PD = (34 - T_a)(V - 0.05)^{0.62} \quad (3.14 + 0.37\,VI) \quad (\%) \; \text{(SI unit)}. \tag{4b}$$

For $V < 10$ ft/min [0.05 m/s] insert $V = 10$ ft/min [0.05 m/s], and for $PD > 100\%$ use $PD = 100\%$, where T_a is the local air temperature (°F [°C]), V is the mean velocity (ft/min [m/s]), and I is the turbulence intensity (%). The turbulence intensity is defined as the velocity fluctuation over the mean velocity. The relationship between the turbulence intensity, I, and turbulence energy, k, can be written as

$$I = 100\,(2k)^{0.5}/V \quad (\%). \tag{5}$$

T_a, V, and k can be obtained from the airflow computation as mentioned in the previous section, and, therefore, the PD distribution due to draft can be determined.

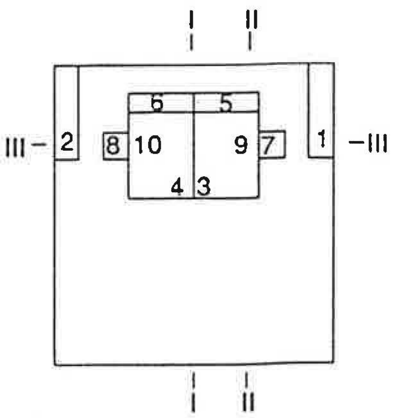
CASE SETUP

Research was conducted for a new office room as shown in Figure 1. The room is 14.76 ft (4.5 m) long, 14.76 ft (4.5 m) wide, and 8.20 ft (2.5 m) high with some furniture, two computers, and two occupants. For simplicity, no aerodynamic blockage is used for the computers. The odor from the floor is assumed to be 0.5 olf; from the walls and ceiling, 0.5 olf; from the books in each bookshelf, 1.0 olf; from the smoking person (occupant a), 5.0 olf; and from the nonsmoking person (occupant b), 1.0 olf. The ventilation system and furniture are supposed to receive appropriate maintenance and were selected such that no odor sources are assumed. This implies that it is a low-olf office with a total odor load (total olf value over the floor area) of 0.014 olf/ft² (0.15 olf/m²) if unoccupied. The inside surface temperature of the enclosure is 72.1°F (22.3°C). To simulate a summer cooling situation, a convective heat gain of 512 Btu/h (150 W) is assumed from the window due to solar radiation. The heat source from each occupant is 273 Btu/h (80 W) and from each computer, 410 Btu/h (120 W). The ventilation rate is 5 ach, or 148 ft³/min (70 L/s). The temperature of the air supplied is 66.2°F (19.0°C). In order to study the air diffusion for different locations of supply outlets and return inlets and for different diffuser characteristics, six cases have been set up as indicated in Figure 2. Cases A through E are also given in ASHRAE (1989b).

Since the effective area of a diffuser is generally smaller than the gross area, the actual air momentum through the supply opening is higher than that through a simple slot. The higher momentum entrains the room air near the supply opening so that a longer projection is expected. In the airflow program, the air momentum of the outlet, $m\,V_{in}$, is set as

$$m\,V_{in} = m(\text{volume inflow rate/effective area}) \tag{6}$$

where m is the mass inflow rate. Equation 6 is a mathematical model for simulating a diffuser. This model is applied to simulate the outlet in Cases A, B, C, and D. In this numerical approach, every cell volume of the outlet is, in fact, characterized by a fraction of the effective area over the gross area of the diffuser. The fraction determines what portion of each cell volume is available for occupancy by the air and what portion of each cell-face area can permit flow, by convection or diffusion, from one cell to its neighbor. If a spreading diffuser should be simulated, it can be done by setting air supply direction for each cell of the outlet. By giving different kinds of



1 - bookshelf a; 2 - bookshelf b; 3 - table a; 4 - table b; 5 - filing cabinet a; 6 - filing cabinet b; 7 - occupant a; 8 - occupant b; 9 - computer a; 10 - computer b; 11 - window.

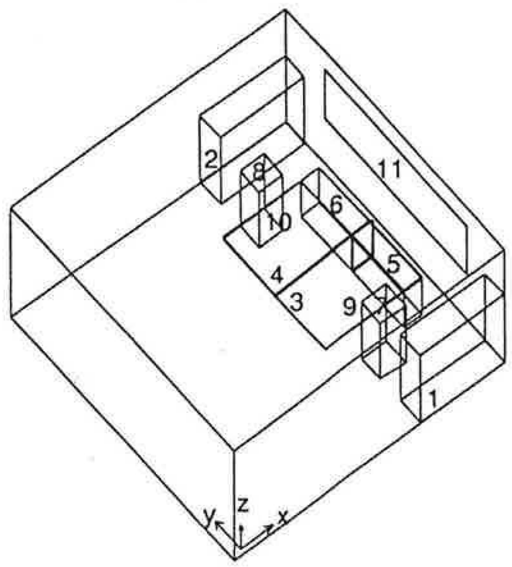
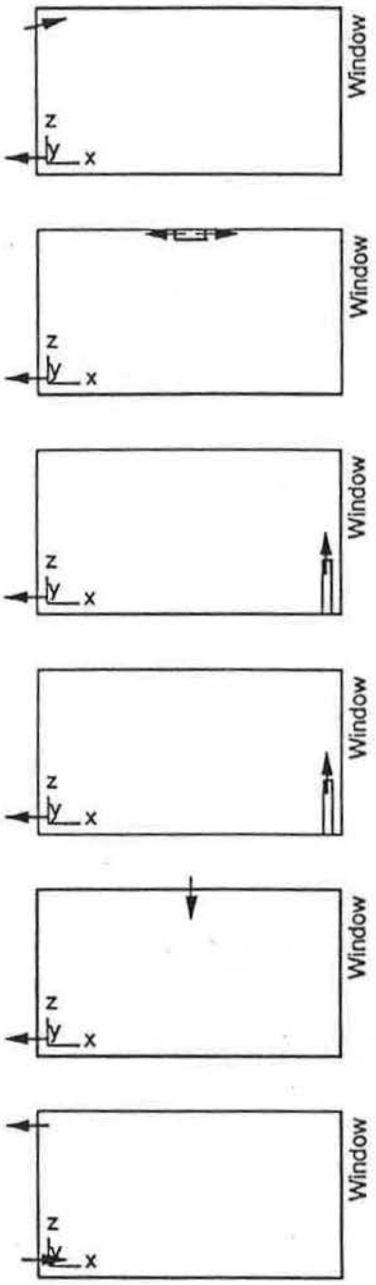


Figure 1 Sketch of the office.



Case A:
Outlet size: w = 2.3 ft (0.7 m), h = 0.5 ft (0.15 m).
Outlet velocity at the effective area: 16.4 ft/s (5.0 m/s).
Non-spreading jet, upward angle: 5.7°
Inlet size: w = 1.64 ft (0.5 m), h = 0.82 ft (0.25 m).

Case B:
Outlet size: w = 1.64 ft (0.5 m), h = 0.16 ft (0.05 m) (Four slots).
Outlet velocity at the effective area: 9.84 ft/s (3.0 m/s).
Non-spreading jet.
Inlet size: w = 1.64 ft (0.5 m), h = 0.82 ft (0.25 m).

Case C:
Outlet size: l = 1.97 ft (0.6 m), w = 0.2 ft (0.06 m).
Outlet velocity at the effective area: 8.2 ft/s (2.5 m/s).
Non-spreading jet.
Inlet size: w = 1.97 ft (0.6 m), h = 0.82 ft (0.25 m).

Case D:
Outlet size: l = 1.97 ft (0.6 m), w = 0.2 ft (0.06 m).
Outlet velocity at the effective area: 5.9 ft/s (1.8 m/s).
Spreading jet, spreading angle: 34°
Inlet size: w = 1.97 ft (0.6 m), h = 0.82 ft (0.25 m).

Case E:
Outlet size: l = 1.64 ft (0.5 m), w = 1.64 ft (0.5 m).
Outlet velocity at the effective area: 0.92 ft/s (0.28 m/s).
Non-spreading jet.
Inlet size: w = 1.64 ft (0.5 m), h = 0.82 ft (0.25 m).

Case F:
Outlet size: w = 1.97 ft (0.6 m), h = 1.97 ft (0.6 m).
Outlet velocity at the effective area: 0.66 ft/s (0.2 m/s).
Non-spreading jet.
Inlet size: w = 1.97 ft (0.6 m), h = 1.08 ft (0.33 m).

Figure 2 Outlet and inlet locations and diffuser characteristics of the six cases.

supply momentum and its initial directions, different diffusers can be simulated. However, this method may not be applied to an outlet with very complex geometry.

The outlet in Case A simulates a nonspreading side-wall grille; in Case B, a nonspreading ceiling diffuser or grille mounted for horizontal forward air projection; in Case C, a nonspreading floor unit; in Case D, a spreading floor diffuser; in Case E, a nonspreading ceiling diffuser or grille designed for vertical downward air projection; and in Case F, a nonspreading diffuser for displacement ventilation.

RESULTS

The computed field distributions of air velocity and temperature, percentage of dissatisfied people due to air quality, perceived air quality, and smoke concentration, and percentage of dissatisfied people due to drafts in different sections are illustrated in Figures 3 through 8 for the office under the six kinds of air diffusion. Subfigures g and h are the distributions of the perceived air quality due to 1.0 olf smoke odor or the distributions of the smoke concentration caused by a 0.01 mL/s smoke

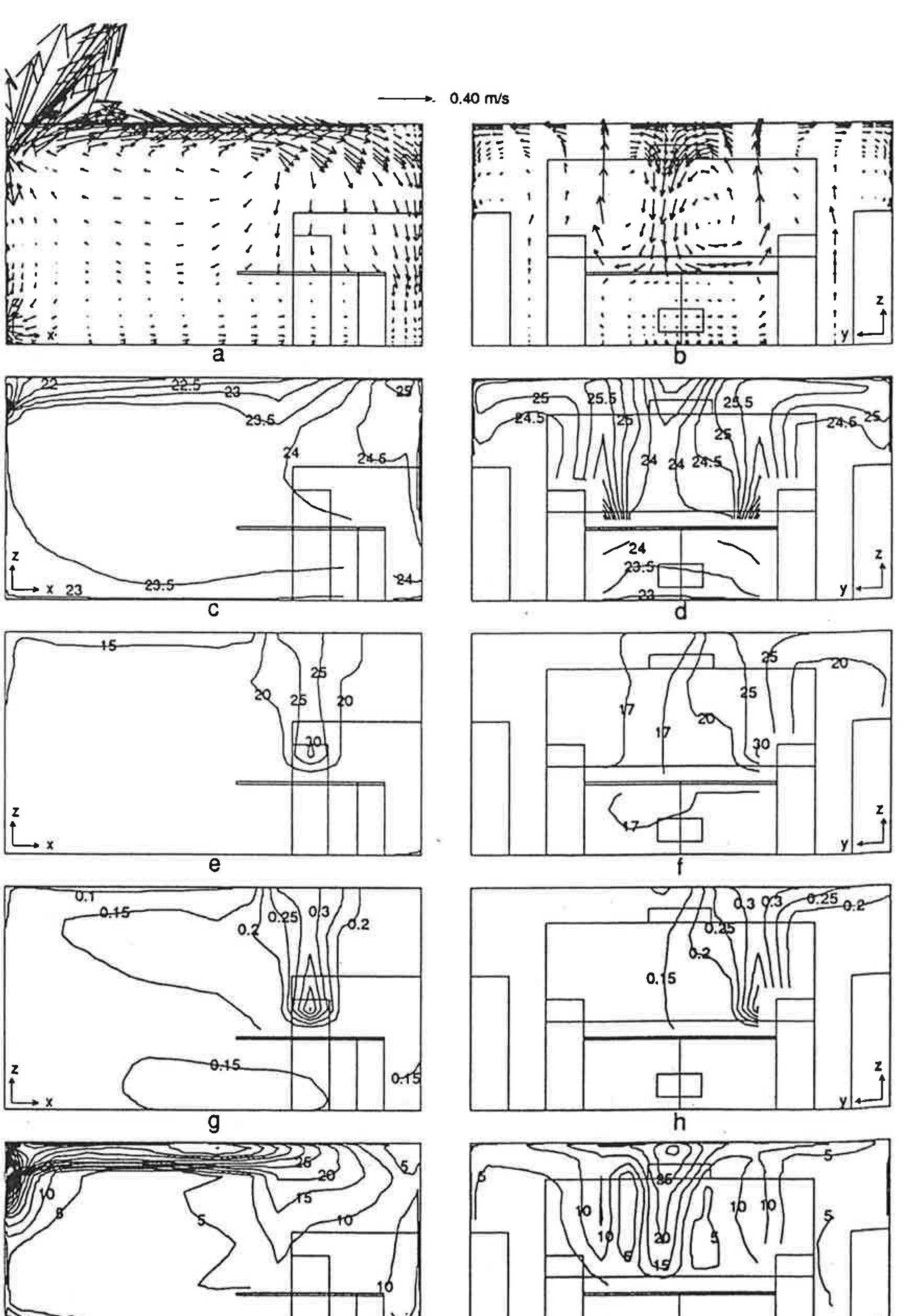


Figure 3 Computed field distributions for Case A: (a) velocity in section I-I; (b) velocity in section III-III; (c) temperature in section I-I [°C]; (d) temperature in section III-III [°C]; (e) percent dissatisfied due to air quality in section II-II [%]; (f) percent dissatisfied due to air quality in section III-III [%]; (g) perceived air quality/smoke concentration in section II-II [decipol or ppm]; (h) perceived air quality/smoke concentration in section III-III [decipol or ppm]; (i) percent dissatisfied due to draft in section I-I [%]; (j) percent dissatisfied due to draft in section III-III [%].

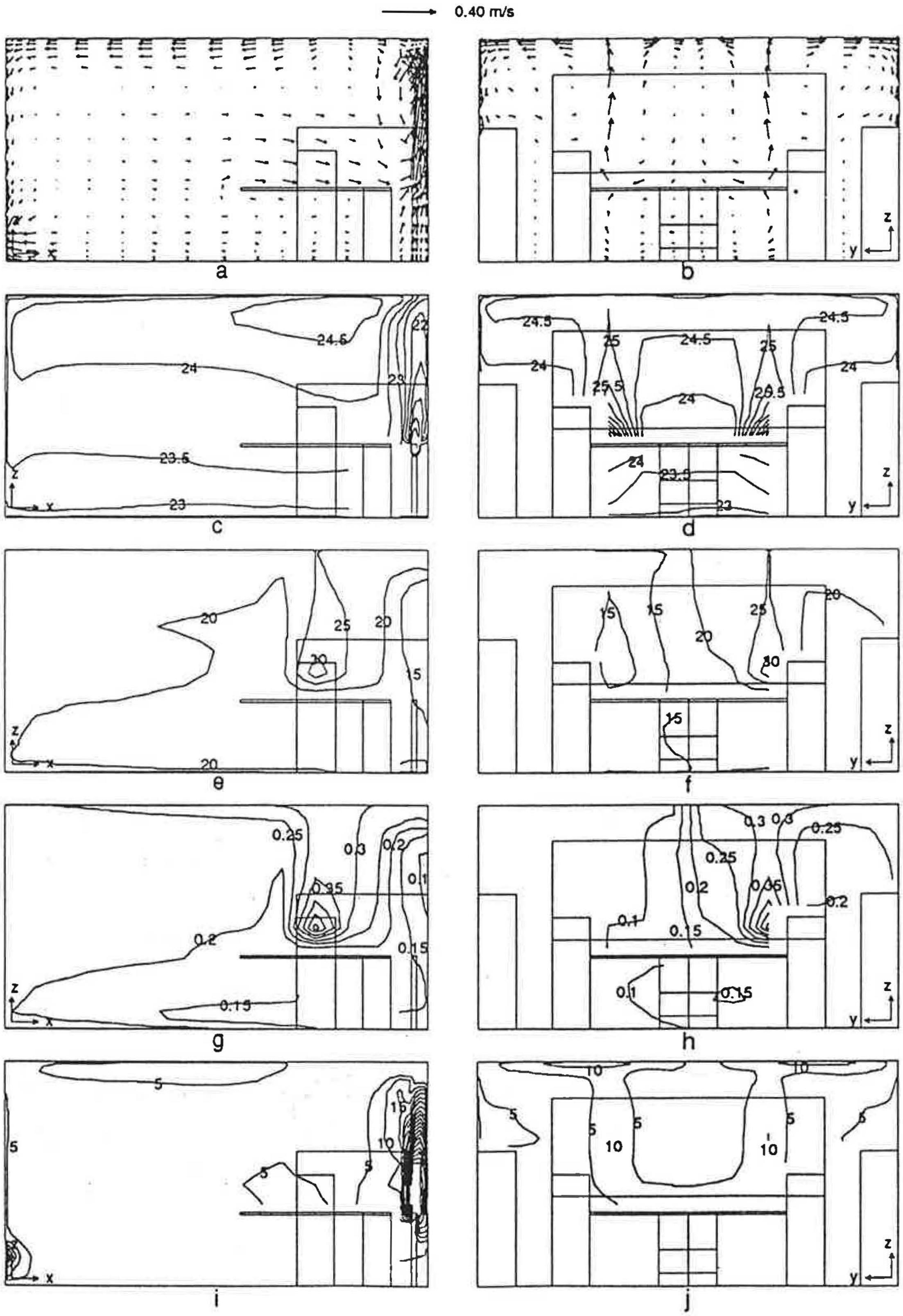


Figure 6 *Computed field distributions for Case D: (a) velocity in section I-I; (b) velocity in section III-III; (c) temperature in section I-I [°C]; (d) temperature in section III-III [°C]; (e) percent dissatisfied due to air quality in section II-II [%]; (f) percent dissatisfied due to air quality in section III-III [%]; (g) perceived air quality/smoke concentration in section II-II [decipol or ppm]; (h) perceived air quality/smoke concentration in section III-III [decipol or ppm]; (i) percent dissatisfied due to draft in section I-I [%]; (j) percent dissatisfied due to draft in section III-III [%].*

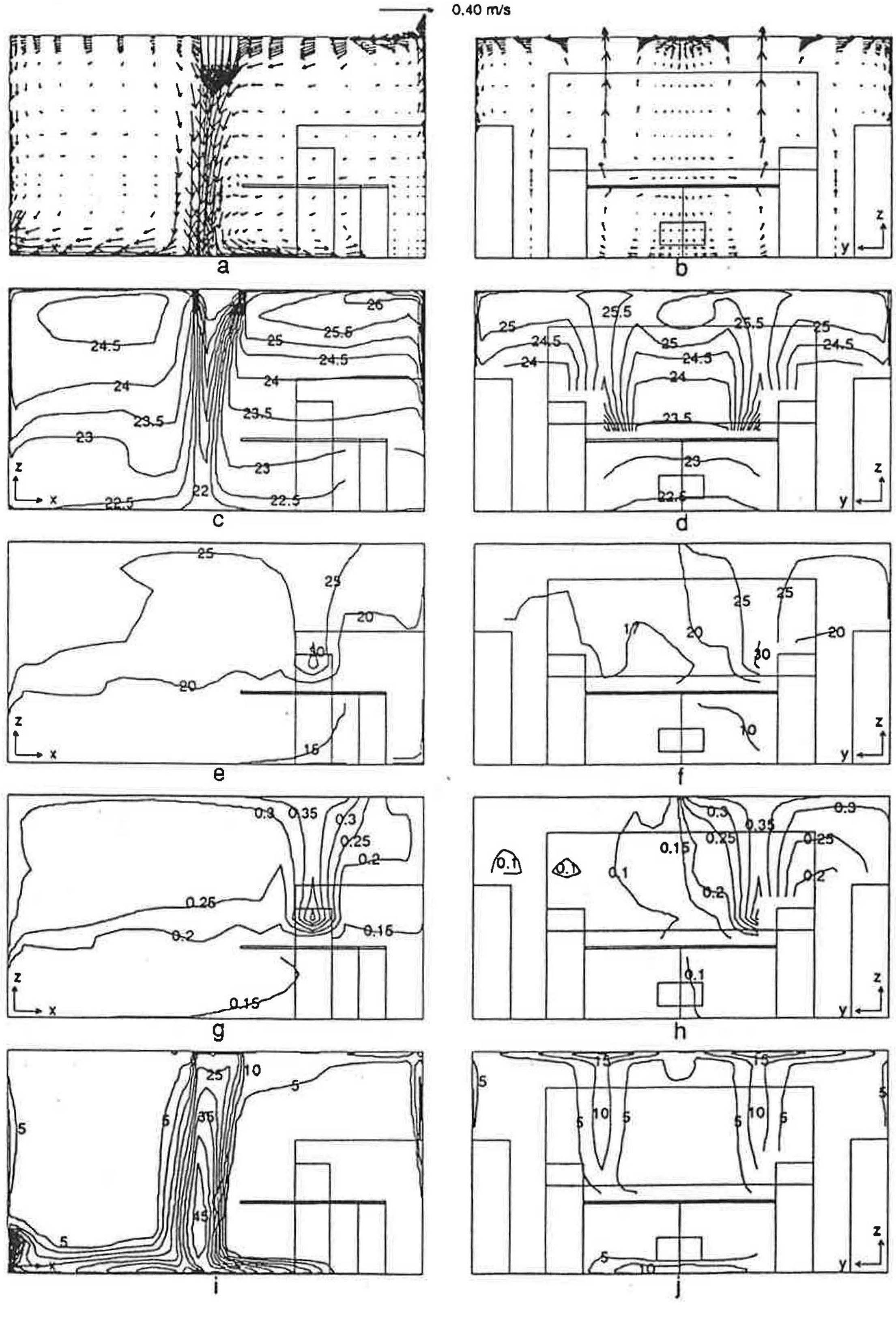


Figure 7 *Computed field distributions for Case E: (a) velocity in section I-I; (b) velocity in section III-III; (c) temperature in section I-I [°C]; (d) temperature in section III-III [°C]; (e) percent dissatisfied due to air quality in section II-II [%]; (f) percent dissatisfied due to air quality in section III-III [%]; (g) perceived air quality/smoke concentration in section II-II [decipol or ppm]; (h) perceived air quality/smoke concentration in section III-III [decipol or ppm]; (i) percent dissatisfied due to draft in section I-I [%]; (j) percent dissatisfied due to draft in section III-III [%].*

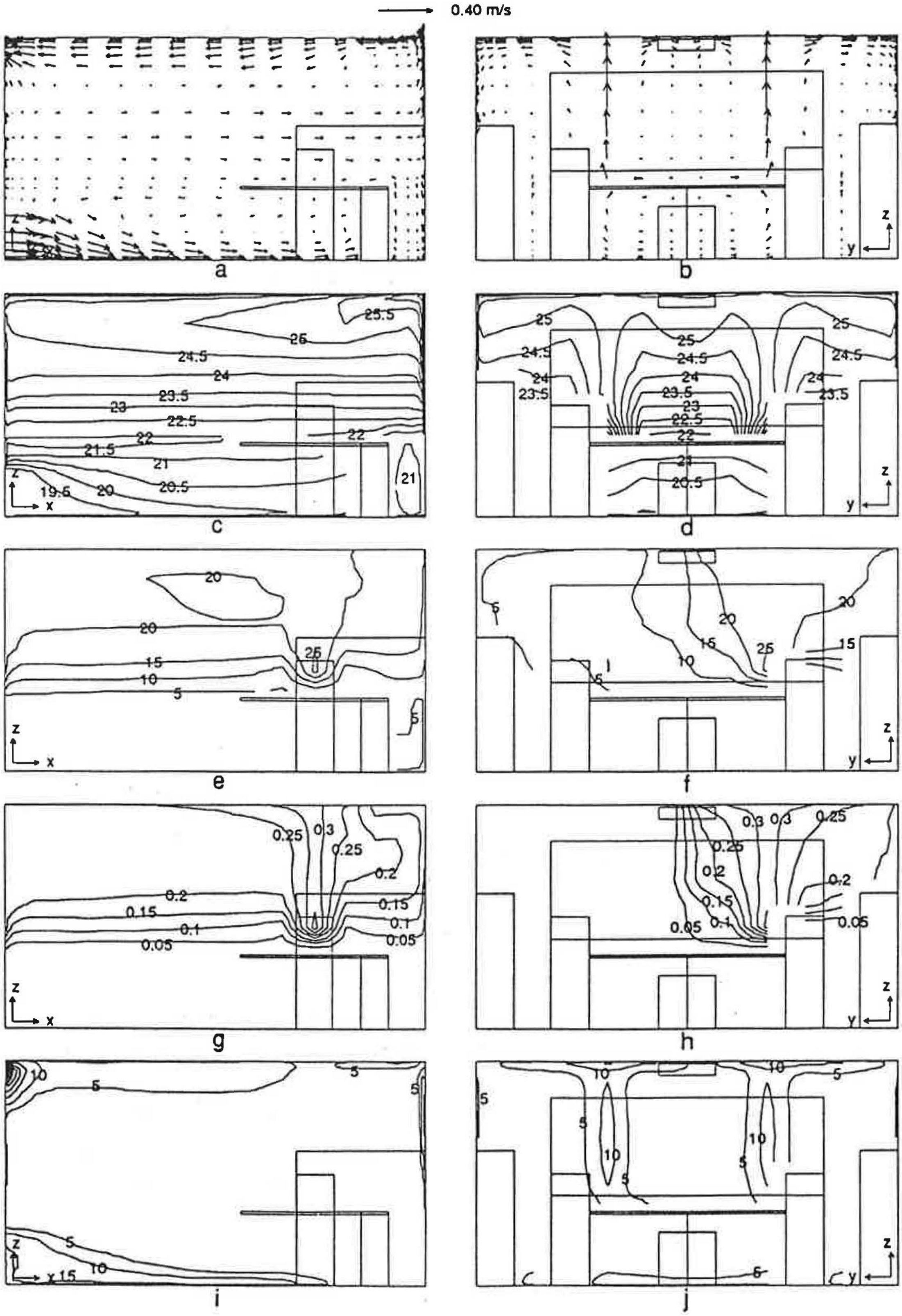


Figure 8 *Computed field distributions for Case F: (a) velocity in section I-I; (b) velocity in section III-III; (c) temperature in section I-I [°C]; (d) temperature in section III-III [°C]; (e) percent dissatisfied due to air quality in section II-II [%]; (f) percent dissatisfied due to air quality in section III-III [%]; (g) perceived air quality/smoke concentration in section II-II [decipol or ppm]; (h) perceived air quality/smoke concentration in section III-III [decipol or ppm]; (i) percent dissatisfied due to draft in section I-I [%]; (j) percent dissatisfied due to draft in section III-III [%].*

source. To simulate a smoking person, the computed concentration caused by the occupant may be scaled according to the ratio of the actual to the nominal source strength of a particular contaminant. The application of dual units of olf/decipol and concentration is necessary because not all pollutants can be perceived. (This was discussed in the previous section.) Figure 9 shows the air velocities in a particular section of each case. A summary of the results for the six cases is given in Table 1.

Case A

Case A is a very commonly used air diffusion. This ventilation method is often preferable from the installation

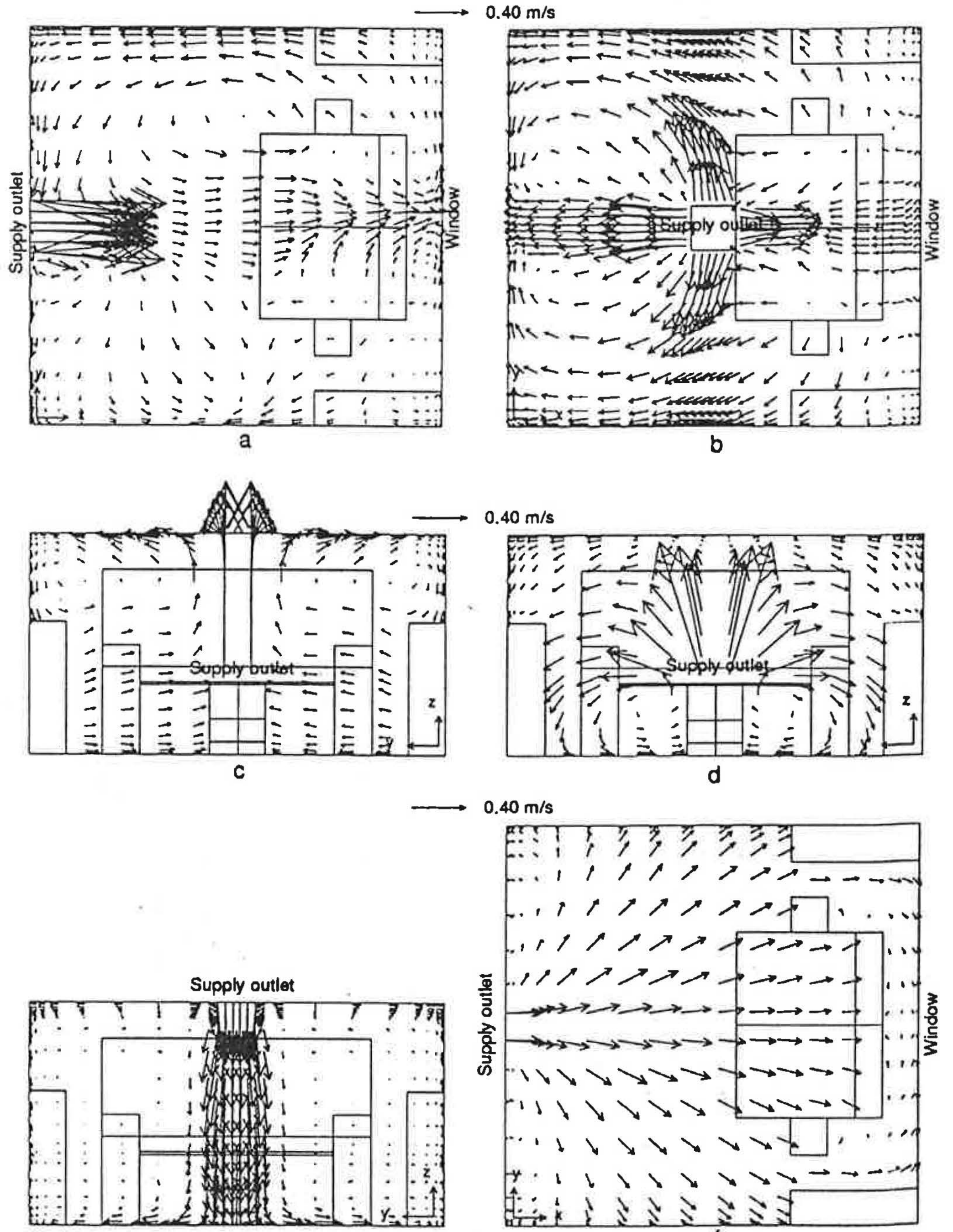


Figure 9 Velocity distributions in a particular section of each case: (a) for Case A in section z = 7.38 ft (2.25 m); (b) for Case B in section z = 7.81 ft (2.38 m); (c) for Case C in section x = 14.27 ft (4.35 m); (d) for Case D in section x = 14.27 ft (4.35 m); (e) for Case E in section x = 7.38 ft (2.25 m); (f) for Case F in section z = 0.16 ft (0.05 m).

TABLE 1
Summary of the Results in the Occupied Zone for the Six Cases

Case	Outlet Type	Jet Type	Vertical Temperature Distribution	PD Due to Thermal Comfort	PD Due to IAQ
A	side-wall grille	nonspreading	uniform	< 15%	< 20%
B	horizontal ceiling diffuser	nonspreading	uniform	< 15%	< 20%
C	floor unit	nonspreading	uniform	< 15%	< 20%
D	floor diffuser	spreading	small gradient	< 5%	< 20%
E	vertical ceiling diffuser	nonspreading	moderate gradient	most < 10%	< 25%
F	displacement ventilation	nonspreading	large gradient	< 10%	< 15%

aspect but may be inadvisable from the thermal viewpoint. This is because high velocities are found above the tables near the window (subfigure 3a). This system is not very suitable for the supply of cooling air at a high rate because, if the supply air velocity is too low, convective currents from the warm surface of the window and the supply airstream meet under the ceiling at some distance from the window and may cause a cold down-draft. However, too-high supply velocity will result in an overthrow condition, causing the total air to drop along the window with too-high velocities.

The velocity distributions shown in subfigures 3b and 9a indicate an asymmetry in the mid-width plane, although the boundary conditions and room geometry are absolutely symmetrical. Theoretically, the differential equation of flow will give a symmetrical flow field, but the upwind scheme and staggered grid arrangement employed in the numerical scheme generate a slight asymmetry. Since the Coanda effect is very strong, as the air supply velocity in this case is very high, an asymmetrical flow field is obtained.

In general, the air temperature distribution in the occupied zone is uniform except in the area where heat sources are introduced. The temperature difference between heights of 0.33 ft (0.1 m) and 5.9 ft (1.8 m) is less than 0.9°F (0.5°C), as illustrated in subfigure 3c.

Adding together all the values of perceived air quality caused by different odor sources, the distribution of the percentage of people dissatisfied due to air quality can be calculated as shown in subfigures 3e and 3f. Although the ventilation rate for the office is considerable (5 ach), the mathematically averaged perceived air quality (C_i) is 1.286, which corresponds to a percentage of dissatisfied people of 18.6%. In this case, the local percentage of dissatisfied people due to air quality is higher at many points than the mathematically averaged percentage because it is hard to mix room air perfectly. However, the percentage of dissatisfied people due to air quality in the side of the room with the nonsmoker is between 15% and 17%, which is slightly less than the average value.

Subfigures 3i and 3j show the distributions of percent dissatisfied people due to draft. The values of percent dissatisfied people due to draft in most parts of the occupied zone are less than 15%. The large air velocities above the tables cause a high dissatisfaction rate there.

Case B

This air distribution is suitable for high ventilation rates because the throws are shorter. Subfigures 4a, 4b, and 9b show that the total air movement to the window is counteracted by the rising natural convection currents from the hot window. The total air drops before reaching the window. It is possible to increase the outlet velocity, but that would result in a too-long throw to the rear wall with large velocities, causing a draft in the occupied zone near the rear wall. It is better to have different supply velocities for different directions. However, this may not be feasible because of the difficulty in manufacturing and of variations of source locations. This type of air supply outlet is better for larger space loads because the cold air is supplied horizontally so that warm air near the ceiling will be mixed with the cold air before it reaches the occupied zone.

The air temperature in this case is very uniform with little temperature stratification, as shown in subfigure 4c. The field distributions of percent dissatisfied people due to air quality and smoke concentration and percent dissatisfied due to draft, illustrated in Figure 4, are very similar to those for Case A. The differences occur mostly in the vicinity of the outlet, inlet, and heat sources. The too-long throw to the rear wall results in an area of high discomfort there, and the too-short throw dropping above the tables also causes draft.

Case C

This air diffusion requires a long throw if good air circulation is to be achieved in the room. The long throw provides a high rate of entrained and recirculated flow, and the method is thus suitable regardless of whether the air temperature differential is large or small. Any tendencies toward cold down-draft from the window are eliminated. With a large air supply velocity, as needed for longer projection, the induction near the outlet is very strong, as illustrated in subfigures 5a and 9c. As a result, room air is quite well mixed. For an air supply velocity that is too low, the total air may drop down to the occupied zone before reaching the rear wall, which would lead to a high discomfort rate.

Since room air is well mixed, there is almost no temperature stratification in the occupied zone, as shown in subfigure 5c. The heat sources of the computers and occupants have a large influence on the local temperature field, as indicated in subfigure 5d. This can also be found in other cases.

The distributions of the percent dissatisfied people due to air quality and smoke concentration (subfigures 5e through 5h) are very similar to those of Cases A and B. This implies that, if air diffusion is of the "well-mixed" type, the temperature distribution, as well as the contaminant distribution, is uniform in the occupied zone except near heat sources.

Subfigures 5i and 5j show that the percent dissatisfied people due to draft is mainly a function of local air velocity and turbulence because the room air temperatures are rather uniform (Equation 4). The thermal environment is accept-able, as the percent dissatisfied people due to draft is less than 15% in the occupied zone. From the results of Cases A, B, and C, we can see that, if a suitable air diffuser is used in a well-mixed system, thermal comfort is not a problem, although the air inflow rate is considerably high.

Case D

Case D is the same as Case C except that a wide-spreading diffuser is used. Subfigures 9c and 9d give a comparison of the airflows near the outlets for the two cases. In Case D, the induction by the outlet air increases the total air temperature and will not cause draft after it falls back into the occupied zone. The velocities in Case D are generally smaller than those in Case C. In addition, a temperature stratification is found in Case C because the total air, which is colder, falls back to the lower part of the room and the warm air from the heat sources reaches to the ceiling without mixing with the cold air.

Subfigures 6e through 6h indicate that the field distributions of percent dissatisfied people due to air quality and smoke concentration are rather uniform. In an earlier study by Chen et al. (1988), they found a very high contaminant concentration at the ceiling and a very low one near the floor. The major difference between the present study and their work is that the air projection in Case D is longer because the diffuser characteristics are different. The longer projection brings polluted air downward to the lower part of the room, and this results in a uniform distribution of the contaminants. In other words, the contaminant distribution is very sensitive to the outlet performance for this type of air diffusion.

As air velocities are smaller, the percent dissatisfied people due to draft is about 5% in most of the occupied zone, as illustrated in subfigures 6i and 6j. The case presents the best thermal environment among the six kinds of air diffusion. Although there is about 1°C temperature difference between head (3.6 ft [1.1 m]) and ankles (0.33 ft [0.1 m]), it will not cause any dissatisfaction (Olesen et al. 1979).

Case E

This type of air diffusion is not very suitable for a low ceiling and high ventilation rates under summer cooling situations. The air diffusion shown in subfigures 7a, 7b, and 9e is with a simple slot. But velocities that are too high are found below the outlet in the occupied zone. Since the mass inflow rate is high, it is difficult to mix the primary air with the room air before reaching the occupied zone.

Due to buoyancy effects, the throw of the cold air is very long and can reach to the floor. Therefore, a temperature stratification is formed together with the warm air generated from the hot window surface, computers, and occupants. Since the supply air velocity is kept at its minimum for better thermal comfort, the induction is not very strong. The cold but clean air is directed to the lower part of the room, and there is also a vertical stratification of the smoke concentration (subfigure 7g). Because smoke is the main odor source, the percent dissatisfied people due to air quality is low in the lower part of the room, as shown in subfigure 7e.

Very high values of percent dissatisfied people due to draft are detected under the outlet, although the supply air velocity is kept at its minimum, as illustrated in subfigure 7i. Nevertheless, thermal comfort is good in the room except for the area within the air throw.

Case F

Case F is a displacement ventilation system. It is usually applied to a room with a lower space load (less than 12.7 Btu/h·ft^2 [40 W/m^2]). From subfigure 8h, we can see that the smoke concentration is low in the left part of the office, although there is a smoker on the right side of the tables. This is because the room geometry and heat sources are symmetrical, and the room air is not well mixed. The percent dissatisfied people due to air quality on the side with the nonsmoking person is less than 10% (subfigure 8f). However, the percent dissatisfied due to air quality, as shown in subfigure 8e, is more or less the same as in Cases A to E on the side with the smoker at standing level (5.6 ft [1.7 m]). This can be explained by the airflow field, as illustrated in subfigure 8b. The heat sources from the computer and human body induce a large stream of air together with the smoke up to the ceiling. This air forms a large circulation that brings contaminated air downward along the side wall. There are three large circulations, as shown in subfigure 8a. It is different from the assumption that there is only one eddy in an office with a displacement ventilation system. The complex flow pattern mixes the air better and a higher contaminant concentration is found in the upper part of the occupied zone of the office. However, the air quality in the lower part and nonsmoker side of the room is certainly much better than that in Cases A through E. Chen et al. (1988) and others also reported that the ventilation efficiency of the displacement ventilation system is higher than that of the other ventilation systems. The displacement ventilation system looks much more suitable for a room with a high ceiling.

The percent dissatisfied people due to draft in most of the occupied zone is less than 10%, as given in subfigures 8i and 8j. Because the velocities near the floor are large, as shown in Figure 9f, at level 0.16 ft (0.05 m) above the floor, special attention should be paid to the region near the floor. In the present study, the air velocity in the outlet is low (39.3 ft/min [0.2 m/s]) and the temperature is rather high (66.2°F [19.0°C]). Those conditions certainly favor comfort. In addition, there is a large temperature stratification in the office, as illustrated in subfigures 8c and 8d. The discomfort due to draft shown in subfigures 8i and 8j does not include the influence of the temperature stratification. Olesen et al. (1979) pointed out that 5.4°F (3°C) of temperature difference between head (3.6 ft [1.1 m]) and ankles (0.32 ft [0.1 m]) will lead to 6% of dissatisfaction. The thermal comfort is still acceptable under the current situation.

DISCUSSION

For an evaluation of indoor air quality acceptability on the basis of odor for visitors entering a space, the olf and decipol should be used. The results indicate that a large amount of fresh air is needed to meet the requirement of comfort. For the office presented here, the air supply is much higher than that given in most existing standards, such as *ASHRAE Standard 62-89* (ASHRAE 1989a), and buildings (Persily 1990). But such a high air-change rate is not recommended because of the cost of energy for ventilation. It is preferable to reduce the odor sources by prohibiting smoking and using low-olf materials.

If the room air is well mixed, the distributions of percent dissatisfied people due to air quality, smoke concentration, and thermal comfort are uniform, except in the areas near contaminant and heat sources. However, a nonuniform air distribution may sometimes lead to a uniform distribution of percent dissatisfied people due to air quality, such as in Case D. The distributions of smoke concentration are normally different from those of percent dissatisfied people because the smoke is not the only odor in the room. The odors from materials, books, and bioeffluents from human bodies also contribute to perceived air quality.

Thermal comfort under all six kinds of air distribution is acceptable, although the ventilation rate is very high (5 ach). In Cases D, E, and F, there are air temperature stratifications. The temperature difference between head and ankles can be as large as 5.4°F (3°C) in Case F. The dissatisfaction due to the temperature difference is less than 6%.

The cases discussed here are with one window and one outlet only. If more windows, exterior walls, and outlets are considered, the results can be different. A study is necessary to determine the sensitivities of indoor air quality to changes of heat or contaminant source location, to furniture location and density, and to the number of outlets.

The present study focused on a summer cooling situation. For heating, the results will be very different with respect to air diffusion, indoor air quality, and thermal comfort. Chen (1990) conducted an investigation for air distribution, thermal comfort, and energy analysis in an office under radiant panel heating, radiator heating, and total air heating.

Under various kinds of air diffusion, the energy consumption of HVAC systems is different because of the presence of air temperature stratifications. Energy cost is not reported in this paper. The reader may refer to Chen et al. (1990) for a general analysis of the influence of air diffusion on energy consumption. A methodology for energy analysis under different kinds of air diffusion has been proposed by Chen and van der Kooi (1990).

CONCLUSIONS

The following paragraphs summarize the most important conclusions of this paper.

- The perceived comfort equation and thermal comfort equation can be applied to evaluate local indoor air quality and draft in an office under different kinds of air diffusion. The distributions of air velocity, temperature, perceived air quality, contaminant concentrations, and thermal comfort in an office under six kinds of air diffusion have been computed by an airflow program with a low-Reynolds-number k-ϵ model of turbulence.
- In this office, the odors from materials, books, and bioeffluents from human bodies contribute to perceived air quality. We therefore used the olf and decipol units for an evaluation of indoor air quality acceptability on the basis of odor for visitors entering a space. For contaminants that cannot be perceived, indoor air quality can be evaluated by the concentration of the contaminants. According to the field distributions of dissatisfaction in the office, 5 ach of fresh air are required to maintain less than 20% dissatisfaction, although it is a low-olf office.
- The comfort due to draft has also been analyzed for this office under six kinds of air distribution. The percent dissatisfied people due to draft is less than 15% in the occupied zone in all cases despite the high air-change rate.
- Cases A, B, and C are "well mixed" cases. The distributions of percent dissatisfied people due to air quality, smoke concentration, and thermal comfort are very uniform. It is difficult to identify which one is the best among the three cases. Cases D, E, and F are with temperature stratifications, but the thermal comfort is acceptable. Case D presents the best environment of thermal comfort among the six cases, although there is a small temperature difference between head and ankle levels. The distribution of air quality or smoke concentration is sensitive to the air jet projection. Case E is not recommended for large ventilation rates, since it causes cold down-draft below the outlet in the occupied zone. Case F, displacement ventilation, seems the best one for obtaining good indoor air quality in cooling situations, but with a high draft risk if the supply air velocity is too high and/or the supply air temperature is too low.
- Outlet location and diffuser characteristics are very important for indoor air quality and thermal comfort.

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