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Comparison of airflow and contaminant distributions in rooms with traditional displacement ventilation and under-floor air distribution systems

Kisup Lee[*] Student Member ASHRAE | Tengfei Zhang, Ph.D. | Zheng Jiang, Ph.D | Qingyan Chen, Ph.D. Fellow ASHRAE

ABSTRACT

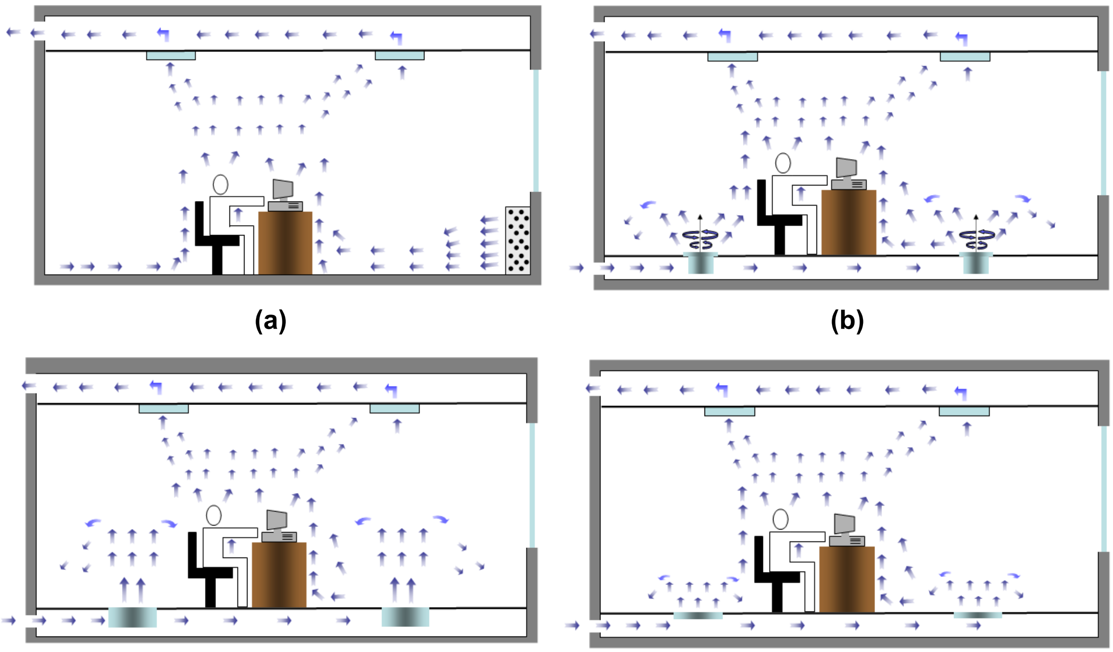
Traditional displacement ventilation (TDV) and under-floor air distribution (UFAD) systems have been used widely because they create better indoor air quality. Many previous studies have compared the TDV or UFAD systems with mixing ventilation systems. This study first compared experimentally the TDV with UFAD systems that use four different diffusers (perforated TDV diffusers, swirl diffusers, linear diffusers, and perforated-floor-panel diffusers) in an environmental chamber that can simulate different indoor spaces of the same size. The two systems had higher ventilation performance than the mixing one under cooling mode as well as under heating mode. Also, the systems with low-height-throw diffusers (all except the linear diffusers) were better. This investigation used a validated CFD program to further study the ventilation performance of the TDV and UFAD systems for an office, a classroom, and a workshop of different sizes. The CFD results further confirmed the findings from the experiment, but with more detailed information and at a lower cost. The air distribution effectiveness with the TDV system and with the low-height -throw UFAD system was in proportion to the ceiling height of the indoor spaces.

INTRODUCTION

Traditional Displacement Ventilation (TDV) and Under-Floor Air Distribution (UFAD) systems have been popular in buildings since the 1970s. Many studies have revealed that TDV and UFAD systems provide better indoor air quality than the traditional mixing ventilation systems (Chen and Glicksman 2003, Bauman and Daly 2003). Some studies (Hu et al. 1999, Im et al. 2005) reported that the two systems can also reduce energy demand because of their high energy efficiency. Typically, a TDV system uses a perforated-sidewall or perforated-corner diffuser that discharges supply air horizontally as shown in Figure 1(a). A UFAD system supplies fresh air from a raised floor panel through swirl diffusers as shown in Figure 1(b), linear diffusers in Figure 1(c), and perforated-floor-panel diffusers in Figure 1(d). Among these diffusers, the swirl diffusers and perforated-floor-panel diffusers generate low-height throws, which means that the air velocity from the diffusers decays quickly and becomes less than 0.3 m/s (60 fpm) at 1.35 m (4.5 ft) above the floor. The linear diffusers generate a high momentum so that the supply air can reach a location at least 1.35 m (4.5 ft) above the floor with an air velocity larger than 0.3 m/s (60 fpm). They also generate high-height throws. The perforated TDV diffuser mounted on the wall or in the corner

[*] Kisup Lee is a Ph.D. student and Qingyan Chen is a professor, School of Mechanical Engineering, Purdue University, West Lafayette, IN. Tengfei Zhang is an assistant professor, Dalian University of Technology, China, Zheng Jiang is a partner of Building Energy and Environment Engineering LLP, Lafayette, IN.

of the room discharges fresh air in a horizontal direction. This diffuser generates low-height throw due to its flow direction and diffuser structure.



(a) (b) (c) (d)

Figure 1 Common TDV and UFAD systems: (a) TDV with a perforated- sidewall or perforated-corner diffuser, (b) UFAD with swirl diffusers, (c) UFAD with linear diffusers, and (d) UFAD with perforated-floor-panel diffusers

On the one hand, the supply air temperature from the TDV or UFAD systems is typically lower than the room air temperature for cooling so that the cool and clean air can stay in the occupied zone. On the other hand, the heat sources in the room generate thermal plumes that bring warm and contaminated air to the upper region to be extracted at the ceiling level. Therefore, the air quality in the occupied zone is normally much better than in the mixing ventilation. The thermal stratification implies a higher extracted air temperature than in the mixing ventilation. Thus, the TDV and UFAD systems can have higher energy efficiency.

Lin et al. (2005) and Kobayashi and Chen (2003) found that diffuser type and ventilation systems could have a significant impact on the ventilation and energy performance of the UFAD systems. Yuan et al. (1998) conducted a similar parametric study on the TDV systems and reached the same conclusions. Thus, unless one can properly select diffusers and ventilation systems, the air distribution created by the TDV or UFAD systems may not always guarantee good performance. For example, Chao and Wan (2004) used linear diffusers with a UFAD system and found that this diffuser may cause discomfort due to high air velocity in the occupied zone. Hu et al. (2003) also found that a UFAD system may cause significant overcooling near the diffusers.

In addition, most studies in the literature assessed the performance of TDV or UFAD systems by comparing each of them with the mixing ventilation. The TDV system is not the same as the UFAD one (McDonel 2003) because of different types of flow driven forces. Unfortunately, little research has been conducted to compare TDV systems with UFAD ones. Furthermore, few comparisons have been made to assess the performance of the various diffusers used in TDV and UFAD systems. It is essential to compare the two ventilation systems and to assess the performance of various diffusers for different types of indoor spaces, such as offices, classrooms, and workshops. This paper addresses these issues in order to assist designers in selecting appropriate ventilation systems and diffusers for their applications.

RESEARCH METHODS

There are two approaches that can be used to compare the performances of different ventilation systems and to assess various diffusers: experimental measurements, and computational simulations by computational fluid dynamics (CFD). In principle, direct measurements give the most realistic information concerning ventilation performance such as the distribution of air velocity, air temperature, relative humidity, and contaminant concentrations. This is why many engineers are still using measurement to evaluate the performance of ventilation systems in fields. However, it is very expensive and time consuming to measure detailed data in an indoor space with a TDV or UFAD system. Unfortunately, the time, the amount of measuring equipment, and the size of the test chamber were very limited in this investigation of the ventilation performances of TDV and UFAD systems in various indoor spaces.

On the other hand, CFD simulations can be used as extensions of the experiment if these simulations are well calibrated by the experimental data. CFD simulations solve a set of conservation equations for flow, energy, and species concentrations in an indoor space and can quickly obtain detailed information concerning ventilation performance at very little cost. The CFD technique is a powerful tool, but it uses approximations to model the flow physics (Versteeg and Malalasekera 1999).

These approximations could bring uncertainties in the numerical results. The uncertainties could come from grid resolution, wall surface and diffuser boundary conditions, and turbulence models. Zhai and Chen (2004) investigated the impact of grid resolution on the CFD results for indoor airflow. They suggested a grid size of 0.005 m (0.2 in) for the natural convection and 0.1m (4 in) for the forced convection, respectively.

The uncertainties due to wall surface boundary conditions come from how the heat transfer is specified: by surface temperature or by heat flux. This investigation used surface temperature, which was not always known. For example, the surface temperature of a lighting fixture was estimated from the power input and the radiation between the lighting and the room, but this estimation could produce some errors.

The uncertainties caused by the approximations used to simulate a diffuser in the CFD could be significant. This investigation used the momentum method (Chen and Moser (1991)) to simulate complex diffusers. Srebric and Chen (2002) compared the momentum method with other methods and recommended it for indoor airflows.

Since indoor airflows are turbulent, the capacity and speed of existing computers cannot calculate the turbulence details without approximation. The most popular approach is to simulate the flow details by a turbulence model. Zhang et al. (2007) demonstrated that a turbulence model may work better for one type of indoor airflow but not well for another. Therefore, it is essential to validate the CFD technique by appropriate experimental data to ensure that all the approximations used are appropriate.

This study used experiments to measure the parameters of ventilation performance of the TDV and UFAD systems and to validate the computational simulation for different indoor spaces such as the distributions of air velocity, air temperature, and contaminant concentrations in several locations. Since the measured data resolution was not sufficiently fine, it was difficult to obtain a complete picture of the ventilation performance in these indoor spaces. Then a CFD program was used to further simulate these cases in great detail. The CFD program was also used to calculate various spaces of different sizes: an office, a classroom, and a workshop.

Experimental approach

The experiment was conducted by using an environmental chamber as shown in Figures 2(a) and (b). The chamber was a well-insulated room with a window. It had a few pieces of furniture and some heated boxes that were used to simulated equipment and occupants in the room. Table 1 gives some information about the chamber size and enclosures. More detailed information about the furniture and heat sources will be presented in the next section. The chamber can simulate a TDV or a UFAD system with various kinds of diffusers. The TDV system employed a perforated-sidewall or perforated-corner diffuser as shown in Figure 2(c) to supply fresh air.

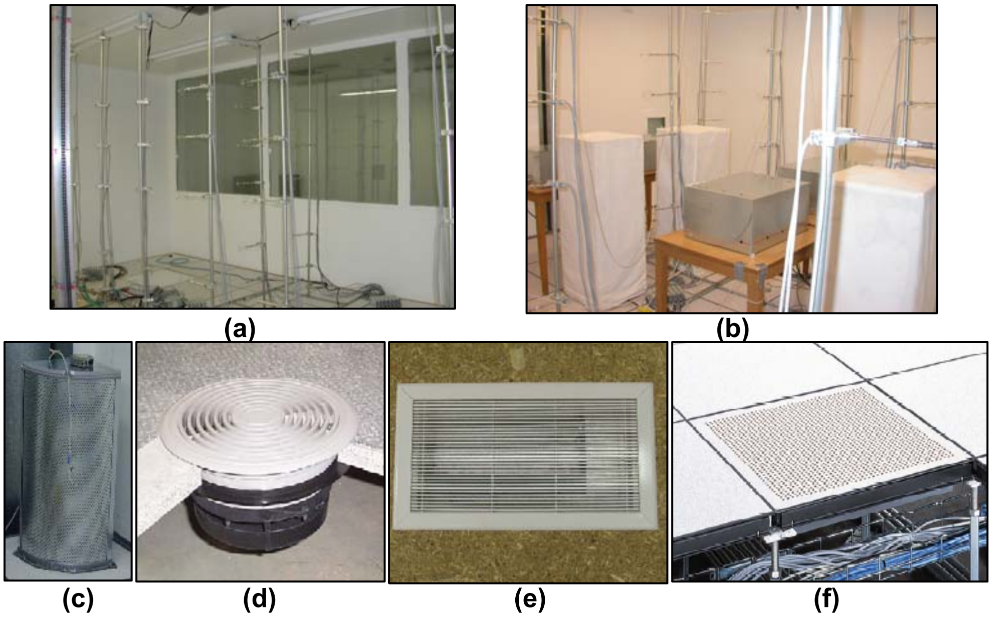


Figure 2 Environmental chamber and diffusers used for the TDV and UFAD systems: (a) and (b) environmental chamber, (c) perforated-corner diffuser, (d) swirl diffuser, (e) linear diffuser, and (f) perforated-floor-panel diffuser

Table 1 Dimensions and enclosure properties of the environmental chamber

	Dimension
Chamber	4.2 m (18 ft) wide × 4.8 m (15.7 ft) long× 2.73 m (8.95 ft) high (including 0.3m (0.98 ft) high floor plenum height)
Window	4.65 m (15.25 ft) wide × 1.55 m (5.1 ft) high
	Thermal resistance
Walls, ceiling, door	5.45 Km^2/W (30 $F \cdot ft^2 \cdot h/Btu$)
Floor	5.45 Km^2/W (30 $F \cdot ft^2 \cdot h/Btu$)
Window	0.25 Km^2/W (1.42 $F \cdot ft^2 \cdot h/Btu$) (double glazing)

The UFAD system utilized a swirl, linear, or perforated-floor-panel diffuser as depicted in Figures 2 (d), (e), and (f), respectively. The specifications of the diffusers provided by the manufacuturers are listed in Table 2. It should be noted that the information provided by a manufacturer may not always be useful for our research. For example, the variation of the throw and the surface distribution of air velocity from a diffuser depends on the supply air temperature and room temperature, which are typically not provided by manufacturers. Thus, this investigation obtained the information by additional experimental measurements.

Table 2 Specifications of various diffusers provided by manufacturers

Diffusers	**Specifications**
Perforated-corner diffuser	- Adequate flow rate : 250 ~ 430 m^3/h (148 ~ 256 cfm) - Total pressure drop : 1.5 mm. w. (0.06 in. w.) at 270 m^3/h (160 cfm)
Swirl diffuser	- Throw heights: 0.73 m (2.41 ft) at 0.3 m/s (50 fpm) with 100 m^3/h (60 cfm) - Horizontal discharging range : 0.3 m (1.0 ft) 0.3 m/s (50 fpm) with 100 m^3/h (60 cfm) - Pressure drop: 0.74 mm. w. (0.029 in. w.)

Linear diffuser	- Adequate flow rate : 85 ~ 550 m^3/h (50 ~ 325 cfm) - Throw height : 2.43 m (8.0 ft) at 0.3 m/s (50 fpm) with 128 m^3/h (75 CFM) - Pressure drop: 0.127 mm. w. (0.005 in. w.)
Perforated-floor-panel diffuser	- Dimension : 600 × 600 mm (2 × 2 ft)

The chamber has an HVAC system to maintain desirable thermal and flow conditions. Table 3 shows the capacities of the HVAC system, and Figure 3 describes the schematic of the system.

Table 3 Specifications of the HVAC system

	Environmental chamber
Supply Fan	849.5 m^3/h (500 cfm)
Return Fan	849.5 m^3/h (500 cfm)
Preheter	8 kW (27,297 Btu/h)
Reheter	8 kW (27,297 Btu/h)
Humidifier	11.34 kg/h (25 lb/h)
Chiller	17.59 kW (6 tons)

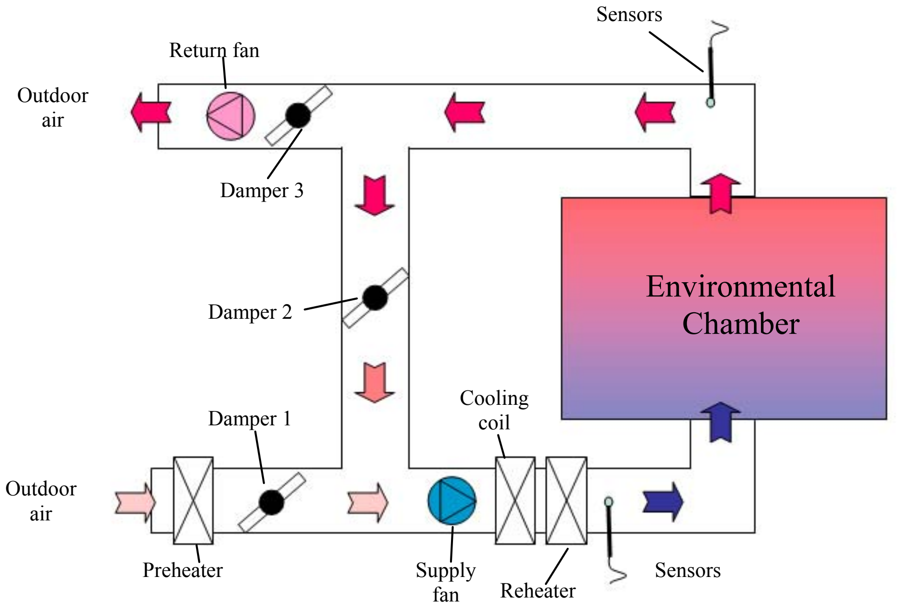


Figure 3 HVAC system for onmental chamber

This investigation measured the distributions of air velocity, air temperature, and tracer-gas concentrations. Fifty-four omni-directional hot-sphere anemometers were used to measure air velocity and temperature in the chamber. Table 4 provides detailed information about the sensor probes. The instantaneous air velocity can be used to determine turbulence intensity. A multi-gas monitor and analyzer system, based on the photo-acoustic infrared detection method, was used to measure tracer-gas concentrations in the chamber. The specifications of the tracer-gas analyzer are listed in Table 5. The air velocity, air temperature, and tracer-gas concentrations were measured at six to eight different vertical positions in nine vertical poles. The experiment used a tracer-gas (SF_6 or CO_2) to simulate a gaseous contaminant generated in the chamber.

Table 4 Specifications of the anemometers

	Hot-spherical anemometer
Velocity sensor	Spherical, diameter=3mm (0.12 in)
Velocity range	0.05 m/s ~ 5 m/s (10 ~ 1000 fpm)
Repeatability	0.01 m/s (2 fpm)
Temperature range	0 ~ 60 ^oC (32 to 140 ^oF)
Temperature accuracy	0.3 ^oC (0.5 ^oF)

Table 5 Specifications of the multi-gas sampler, monitor, and analyzer system

	Photoacoustic infrared spectroscopy
Measurement range	Minimum 0.001 ppm
Measurement unit	ppm or mg/m^3
Response time	about 30 s to 100 s
Repeatability	1% of measured value
Operating temperature	5 ~ 40 ^{o}C (40 to 105 ^{o}F)

The measurements were conducted under steady-state conditions with constant supply airflow rate, supply air temperature, enclosed surface temperatures, and heat sources in the chamber. The experiment had a good repeatability for the air velocity and air temperature. The only uncertainty came from the measurements of contaminant concentrations. Unlike the air velocity and air temperature, the multi-gas analyzer can measure concentrations just a few points at a time. Thus, the distributions of the measured tracer-gas concentrations were not obtained at the same time for all the measuring locations. However, the system was run under steady state, so the tracer-gas concentrations measured over time should be valid.

Numerical approach

The CFD program used in this study solved a set of partial differential equations that can be written in a general form:

$$\frac{\partial(\rho\phi)}{\partial t} + (\rho\vec{u}\cdot\vec{\nabla}\phi) = \Gamma_\phi \vec{\nabla}^2\phi + S_\phi \tag{1}$$

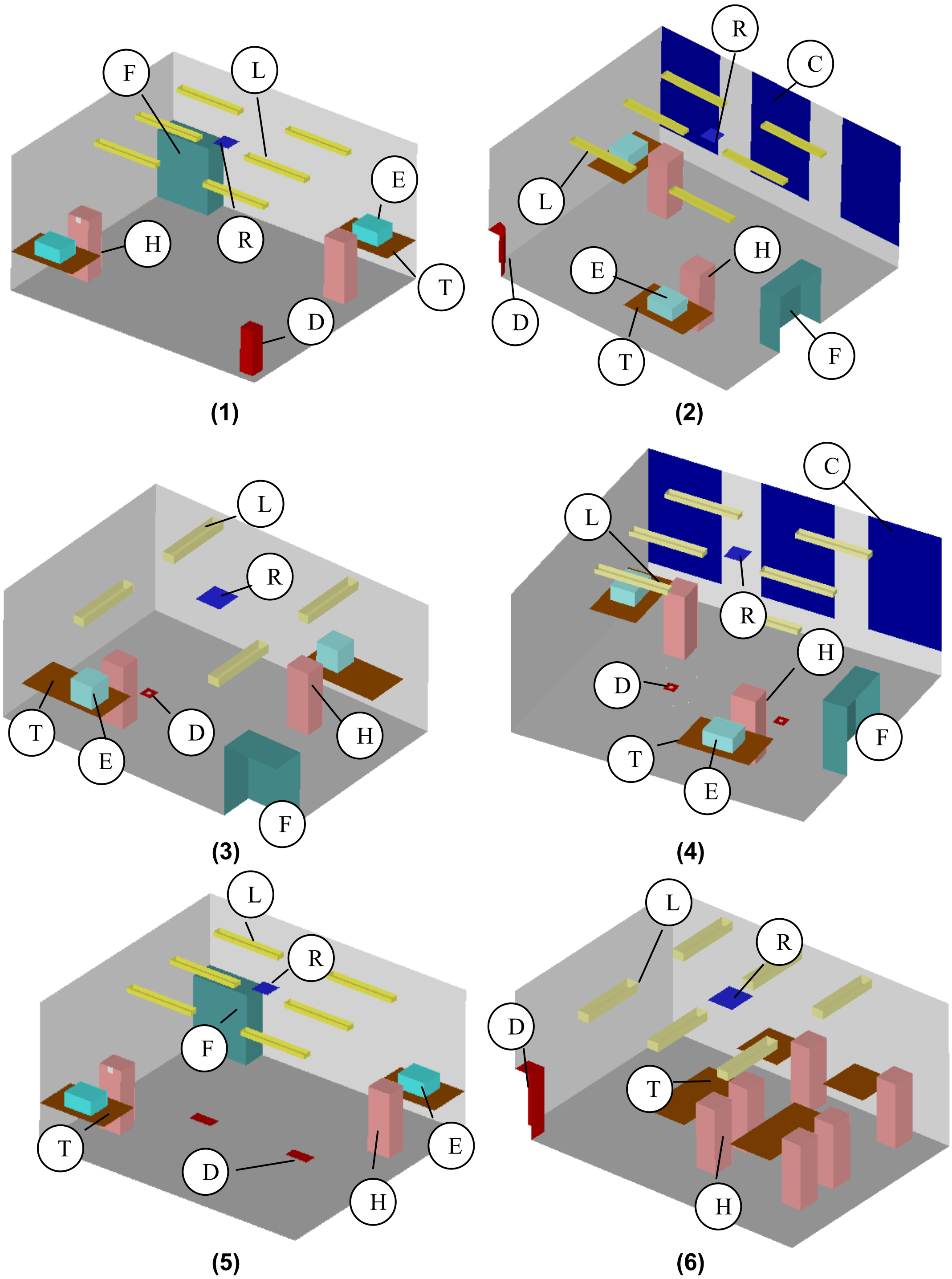
where, ρ is density, ϕ is a dummy scalar variable, t is time, $\vec{u}$ is the velocity vector, Γ_ϕ is the effective diffusion coefficient, and S_ϕ is the source term. The dummy scalar variable can represent mass continuity (ϕ = 1), momentum (ϕ = U, V, and W), energy (ϕ = T), species such as tracer-gas (ϕ = C), turbulent kinetic energy (ϕ = k), and dissipation rate of turbulent kinetic energy (ϕ = ε). The k and ε are the two turbulence parameters used to approximate the turbulence nature of the flow by the Re-Normalization Group (RNG) k-ε model (Yakhot et al. 1992). The model is widely used in indoor environments as suggested by Zhang et al. (2007). The differential equations were solved by the CFD program with the finite volume method that divided the flow domain into many spatial cells. The SIMPLE algorithm was employed to couple pressure and air velocity to obtain the airflow distribution and subsequently the distributions of other scalar variables (Versteeg and Malalasekera 1999).

EXPERIMENTAL MEASUREMENTS

Description of experimental cases

Both the experimental and numerical approaches were used to study the ventilation performance of the TDV and UFAD systems with various air supply diffusers. The environmental chamber was set up to simulate seven cases, which represented a small office, a quarter of a classroom, and a part of a workshop, because it was not possible to vary the size. The experiment tested a perforated-corner TDV diffuser, swirl diffusers, linear diffusers, and perforated-floor-panel diffusers under both cooling and heating modes. Figure 4 shows a sketch of the seven cases. Typically, a space simulated had air supply diffusers, a return grille, simulated occupants, tables, equipment, furniture, and windows with a supply air temperature different from the room air temperature.

The office cases compared the diffuser performance for both cooling and heating modes. The classroom and workshop were added to test the ventilation performance for different indoor spaces. Table 6 lists the internal heat loads and ventilation rates for the seven cases. Cases 1, 2, 4, 5, and 7 were new in this investigation, while the other two cases were selected from the literature: case 3 from Kobayashi (2001) and case 6 from Yuan et al. (1999). The experimental results for these cases can be compared with each other and can also be used to validate the CFD simulations.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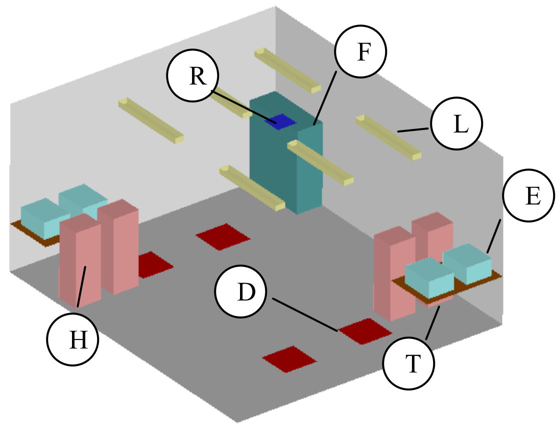


(7)

Legend: C – cooling panel, D – diffuser, E – equipment, F – furniture, H – human, L – lighting, R – return outlet, T – table, W – window

Figure 4 The space layouts used in the experiment: (1) office with a perforated-corner TDV diffuser for cooling mode, (2) office with a perforated-corner TDV diffuser for heating mode, (3) office with two swirl diffusers for cooling mode, (4) office with two swirl diffuser for heating mode, (5) office with two linear diffusers for cooling mode, (6) quarter of a classroom with a perforated-corner TDV diffuser for heating mode, and (7) part of a workshop with four perforated-floor-panel diffusers for cooling mode

Table 6 Thermo-fluid conditions for the seven experimental cases

Cases	Occupants		Total heat generated from equipment [W] ([Btu/h])	Total heat generated from lighting [W/m²] ([Btu/h·ft²])	Ventilation rate [ACH]	Thermal conditions
	Number [-]	Total heat generated [W/person] ([Btu/h·person])				
1	2	75 (256)	200 (682)	28.84 (9.14)	6	Cooling
2	2	75 (256)	200 (682)	28.84 (9.14)	6	Heating
3	2	75 (256)	200 (682)	13.5 (4.27)	4	Cooling
4	2	75 (256)	200 (682)	28.84 (9.14)	6	Heating
5	2	75 (256)	200 (682)	28.84 (9.14)	6	Cooling
6	6	75 (256)	0 (0)	10.8 (3.42)	4	Heating
7	4	200 (682)	400 (1,365)	28.84 (9.14)	10.7	Cooling

Experimental results

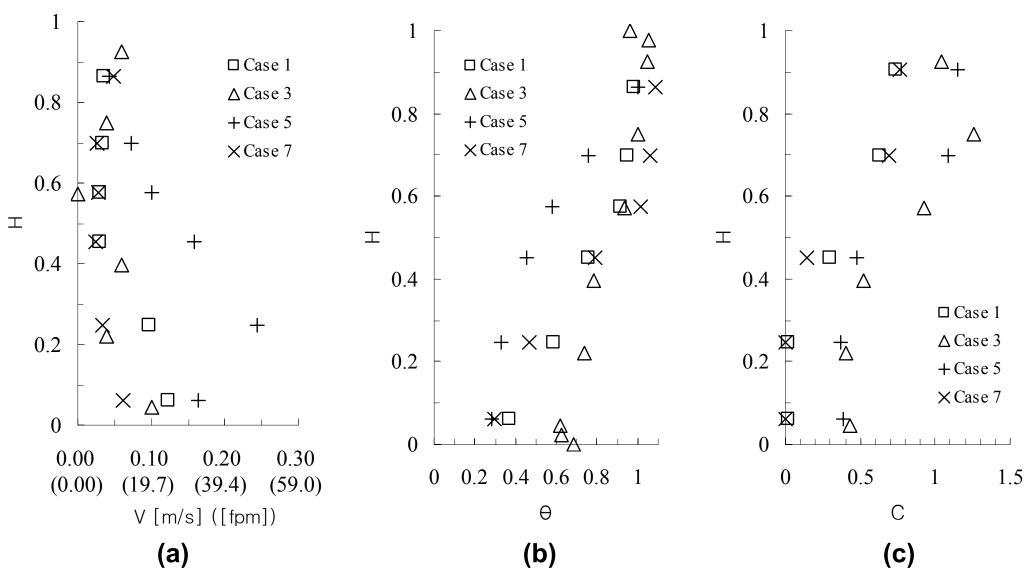
Table 7 lists the face air velocity, air temperature, and tracer-gas concentration at the supply inlets and exhaust for the experiment. The air velocity from the perforated TDV diffusers and perforated-floor-panel diffusers was uniform and low at 0.1 ~ 0.3 m/s (20 ~ 60 fpm). The air velocity from the swirl diffusers was uniform in the peripheral direction at a high value of 1.5 m/s (300 fpm), while the air velocity from the linear diffusers was not uniform and was also high, ranging from 0.2 to 1.5 m/s (40 to 300 fpm). Although the experiment was supposed to use 100% outside air, there was a shortcut between the exhaust outlet and supply intake. Thus, Table 7 shows non-zero tracer-gas concentration at the inlet for SF_6, even though the background concentration of SF_6 should be zero.

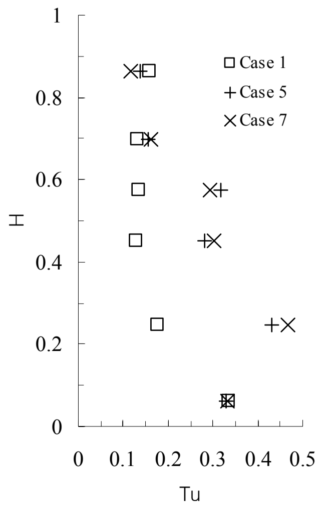
Table 7 Measured parameters from experiments

Case	V_s [m/s] ([fpm])	T_s [^oC] ([^oF])	T_e [^oC] ([^oF])	Tracer Gas	C_s [ppm]	C_e [ppm]
1	0.3 (59.1)	19.0 (66.2)	26.2 (79.2)	SF_6	0.18	0.95
2	0.3 (59.1)	29.4 (84.9)	25.8 (78.4)	SF_6	0.25	1.05
3	1.5 (295.3)	18.8 (65.8)	28.0 (82.4)	SF_6	0.023	0.461
4	1.67 (328.7)	29.9 (85.8)	25.4 (77.7)	SF_6	0.21	1.04
5	0.2~1.5 (39.4~295.3)	20.1 (68.2)	25.3 (77.5)	SF_6	0.18	0.82
6	0.1 (19.7)	19.0 (66.2)	25.8 (78.4)	CO_2	432	924
7	0.132 (25.9)	19.0 (66.2)	26.7 (80.1)	SF_6	0.2	0.88

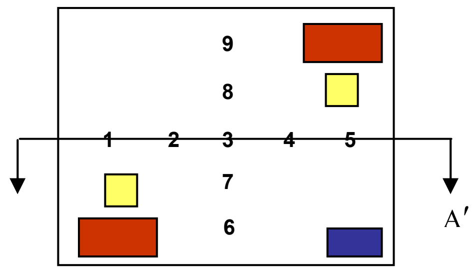
V_s=supply air velocity, T_s = supply air temperature, T_e = extracted air temperature, C_s= supply tracer-gas concentration, and C_e= extracted tracer-gas concentration.

Figure 5 illustrates the averaged air velocity, air temperature, tracer-gas concentration, and turbulent intensity measured in the mid-section (at A-A′ section) of the room. Note that the results were for the cooling mode, and the data presented in the figure were normalized: H= z / h, where h = room height that was 2.27 m (7'-6") for case 3 and 2.43 m (8') for the rest of the cases; θ= (T-T_s) /(T_e-T_s) for cooling mode and θ= (T_e-T_s) /(T-T_s) for heating mode, where T_s is the supply temperature and T_e is the extracted air temperature; and C= (C-C_s)/(C_e-C_s), where C_s is the supply tracer-gas concentration and C_e is the extracted tracer-gas concentration.





(d)



(e)

Figure 5 Comparison of average data profiles at A-A′ section for cooling mode: (a) air velocity, (b) air temperature, (c) tracer-gas concentrations, (d) turbulence intensity, and (e) measurement locations

The velocity profile shown in Figure 5(a) illustrates that the air velocity was generally low in the space except in case 5 with the linear diffusers. The linear diffusers did not diffuse their momentum after discharging it. In addition, the linear diffusers had a very high air velocity at one side and low at the other due to the asymmetrical air coming from the floor plenum. This made the throw from the diffuser much longer and led to a much higher velocity in the room. The averaged air velocity in the occupied zone with the linear diffusers was higher than 0.2 m/s (40 fpm), which could cause thermal discomfort or draft. The air velocity in the space with the other diffusers was small to provide a thermally comfortable environment.

Figure 5(b) shows the vertical air temperature profiles with the four diffusers. The results indicate that case 3 had the smallest temperature gradient, while case 7 had the largest. Since case 3 was with swirl diffusers, the diffusers created a well-mixing condition near them, which is why the gradient was small. Case 7 was with perforated-floor-panel diffusers. The reason for such a large gradient was because of the high heat sources in the space. The temperature difference between the head and foot levels can exceed 3 K (5^{o}F) in this case, which could cause thermal discomfort. Case 5 was with the linear diffusers, which did not mix supply air with the surrounding air in the lower part of the space. Thus, the figure depicts a small temperature gradient in the lower part of the room. However, the temperature gradient was the largest in the middle height, which is not good for thermal comfort. Please note that the results were for the mid-section of the room where the diffusers were located. In the rest of the room, the air temperature gradient was the smallest with the linear diffusers due to the strong mixing caused by the high momentum from the diffusers.

Figure 5(c) depicts the vertical tracer-gas concentration profiles at the A-A′ section of the space. Cases 1 (perforated-corner TDV diffusers) and 7 (perforated-floor-panel diffusers) generated a low-height throw that did not mix supply air with room air in the lower part of the space. As a result, the tracer-gas concentration was almost zero there. Cases 3 (swirl diffusers) and 5 (linear diffusers) had a somewhat high concentration of tracer-gas at the lower part of the space because the diffusers mixed the supply air with that of the surroundings. Again, the results were for the mid-section of the room. By looking at the results in the other part of the space, the concentration gradient with the perforated-corner TDV diffuser was much larger. Thus, the ventilation performance for the entire occupied zone with the perforated-corner TDV diffusers was almost the same as with the swirl diffusers. The linear diffusers created a very uniform concentration profile in the other part of the occupied zone. The ventilation performance for the entire occupied zone with the linear diffuser was the lowest that is close to perfect mixing conditions.

The vertical turbulence intensity profiles as depicted in Figure 5(d) show that case 1 had the lowest intensity, while the other two cases had the highest. Unfortunately, the data for case 3 were not available from the literature. It is clear that the perforated-corner TDV diffusers generated a smooth, low turbulence jet near the floor. The room air was not disturbed, so the turbulence intensity was the lowest. Although the perforated-floor-panel diffusers generated a turbulence intensity as high as that for the linear diffusers, the actual velocity fluctuation was much lower. This is because the perforated-floor-panel diffusers generated a much lower air velocity in the occupied zone than did the linear diffuser.

Figure 6 further compares the differences between the cooling and heating modes. Cases 1 and 2 were with the perforated-corner TDV diffusers and cases 3 and 4 with the swirl diffusers. Cases 1 and 3 were for cooling mode, while cases 2 and 4 were for heating mode. In heating mode, the case with the perforated-corner TDV diffuser (case 2) had a jet that went upwards from the diffuser due to thermal buoyancy. As a result, the air velocity in the occupied zone was lower than in the cooling mode, as shown in Figure 6(a). The swirl diffusers generated a longer throw in the heating mode because of the thermal buoyancy from the supply air. Thus, the air velocity in the upper part of the room was higher.

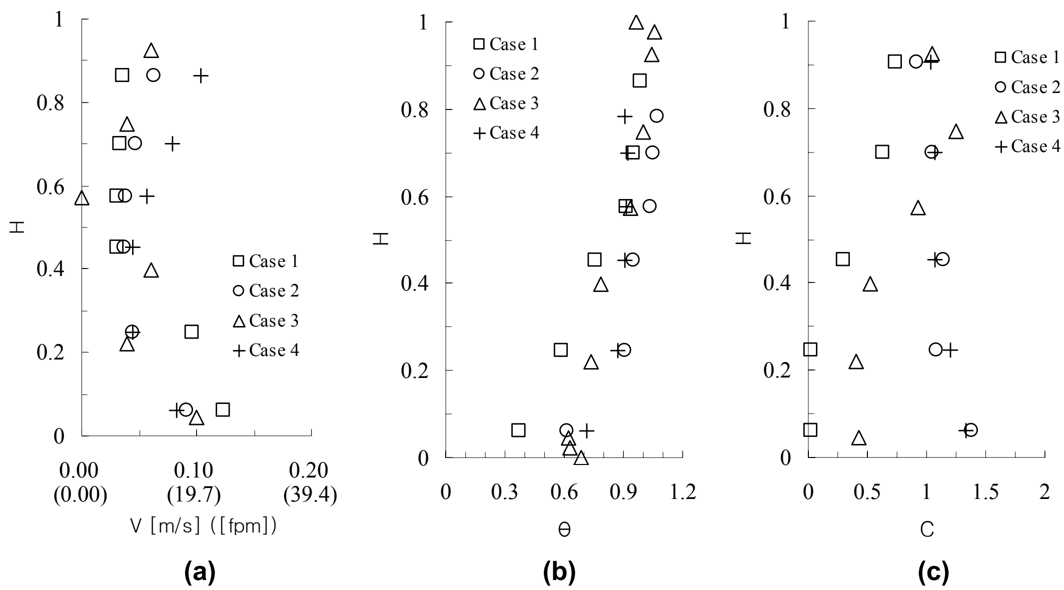


Figure 6 Comparison of the average data profiles at A-A′ section for cooling and heating modes: (a) air velocity, (b) air temperature, and (c) tracer-gas concentration

Figure 6(b) indicates a distinct difference of vertical temperature profiles between cooling and heating modes with the perforated-corner diffuser. The gradient was much smaller for heating due to the higher air supply temperature. For the cases with the swirl diffuser, the temperature profiles look the same in this section. In fact, the temperature gradient in the entire lower part of the room for the heat mode was larger due to the down draft of cold air from the room enclosure.

Figure 6(c) shows a reversed tracer-gas concentration profile in heating mode because fresh air coming from the diffusers went up to the upper part of the room due to the buoyancy force. Then the air was cooled and mixed with the contaminant (tracer gas) and could go downwards to the lower part of the room. Therefore, the distribution of tracer-gas concentration for the heating mode was close to that for the mixing ventilation.

The experiments in this study provided realistic data to compare the thermal comfort and ventilation performance of TDV and UFAD systems. The results show that the diffuser types played an important role. The perforated-corner TDV diffuser, swirl diffuser, and perforated-floor-panel diffuser created a low air velocity in the occupied zone. There were temperature stratifications in the space with those diffusers. The case with the perforated-floor-panel diffuser could have too high a temperature difference between the head and foot levels. The linear diffuser generated the least temperature gradient but the lowest ventilation performance and the highest velocity in the occupied zone. The other three diffusers could have a much higher ventilation performance than that for the mixing ventilation for cooling. For heating mode, the perforated-corner TDV diffuser and the swirl diffuser in this study created mixing conditions. It is anticipated that the other two diffusers will do the same.

NUMERICAL SIMULATIONS

Although the experiment gave the most realistic data, its resolution was low. It is difficult to have a complete picture of the air distributions in such a complex space. The experiment had limited flexibility in representing different indoor spaces. For example, the chamber used in this study can only simulate a quarter of a classroom. It is uncertain if the experiment would actually represent the case of a whole classroom. On the other hand, CFD can give very detailed information of the air distribution and can easily simulate indoor spaces of different sizes. However, the CFD has some uncertainties because of the approximations used in it. Thus, it is essential to validate the CFD so that this validated CFD can be used with confidence to analyze air distribution for different indoor spaces.

Validation of the CFD model

Our validation has been performed for all the seven cases above. In order to properly model the supply diffusers, this study used the momentum method, which was extensively validated by another ASHRAE research project (RP-1009) (Srebric and Chen 2002) for various diffusers. This method could generate an acceptable flow distribution in the downstream but it may not accurately represent the flow in the proximity of a diffuser. Due to the limited space available for this paper, only the validation of the CFD model using the experimental data of case 1 is presented here. The validation of this case is a typical one.

Figure 7 compares the CFD simulation results with the experimental data at three different locations. The experiment measured the data in nine locations as shown in Figure 5(e). For clarity, only the representative data from three locations were used in Figure 7. The calculated air velocity agreed with the measured data as shown in Figure 7(a). The velocity was lower than 0.15 m/s (30 fpm). With such a low velocity, the experimental measurements had a great uncertainty. Again, the calculated air temperature (Figure 7(b)) was in pretty good agreement with the experimental data. However, the agreement between the calculated and measured tracer-gas concentration as shown in Figure 7(c) was not perfect. A very significant difference was found in the upper part of pole 6. The reason for this discrepancy could be explained as follows. There were large-scale vortices at the upper part of the room which could cause unstable mixing at the upper parts. The unstable structure was of low frequency. As a result, the measured tracer-gas concentration was unstable and had a very significant uncertainty. The CFD simulation was only performed under steady-state conditions. Thus, it is hard to achieve accurate agreement in all the measuring positions.

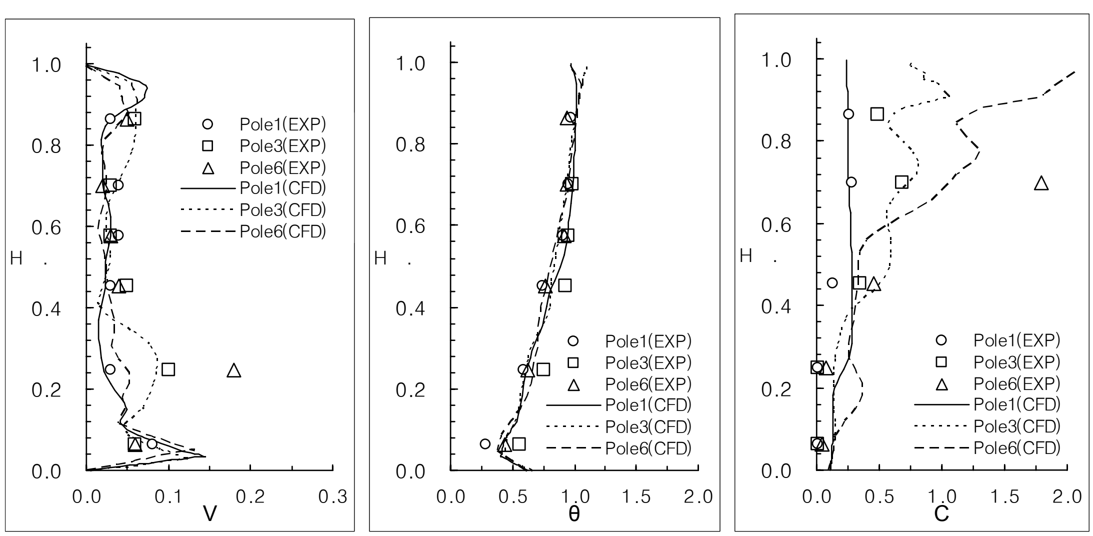
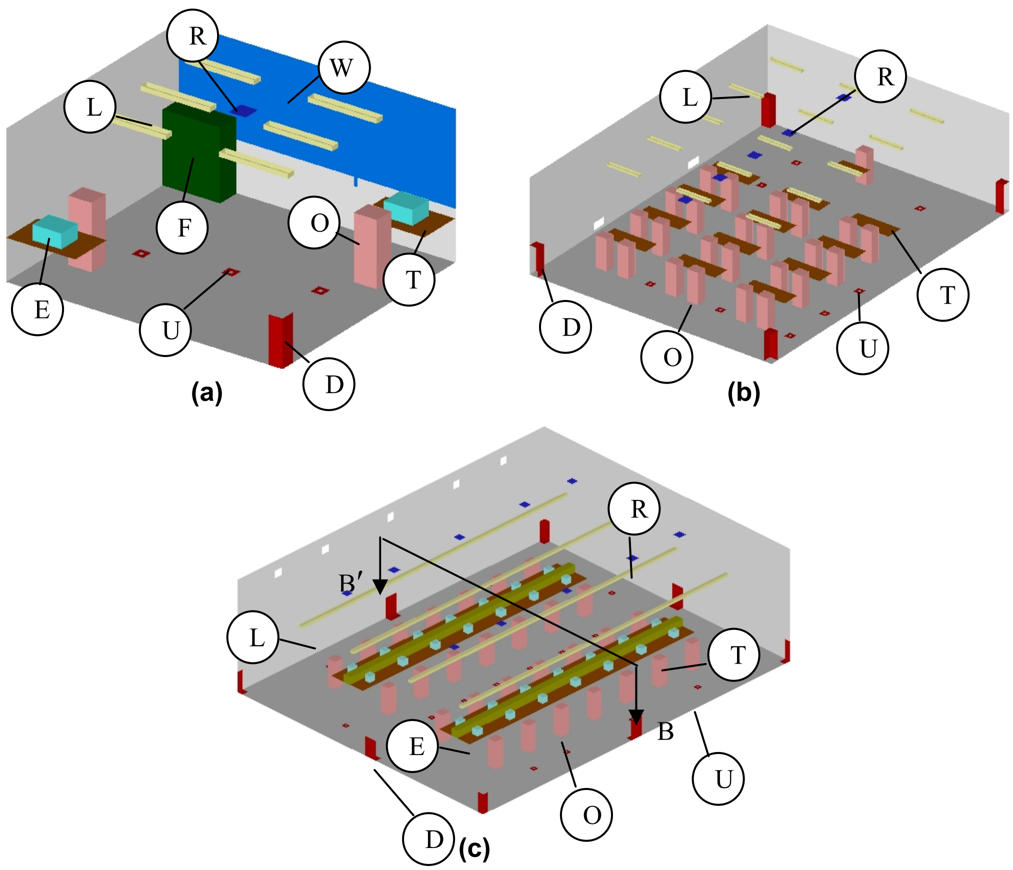


Figure 7 Validation of the CFD results by the experimental data of case 1 (TDV system with a perforated corner diffuser): (a) air velocity, (b) air temperature, and (c) tracer-gas concentration (Pole number was shown in Figure 5(e))

Nevertheless, the agreement between the CFD simulations and the experimental data is acceptable for analyzing ventilation performance. The discrepancies could be as large as those in different experimental measurements as reported in the International Energy Agency Annex 20 conducted in the early 1990s (Lemarie 1993). Hence, the CFD model would be used as a tool to analyze the airflow and contaminant transport in different spaces with the traditional displacement ventilation and under-floor air distribution systems.

Description of different indoor spaces

With the validated CFD model, one can simulate various indoor spaces under realistic thermo-fluid conditions. This approach overcomes the shortcomings in the experimental approach in which the changes in indoor space size were difficult and the measurement resolution was low. The numerical simulations reported in this section were for a two-person office, a classroom, and a workshop, as shown in Figure 8. Offices and workshops are typical working spaces. Classrooms are important for children. They are all important and interesting indoor spaces. These tested indoor spaces had supply diffusers, return grilles, occupants, equipment, furniture, windows, and lighting. As shown in Figure 8, supply diffusers were distributed accordingly to achieve uniform distributions of air temperature and air quality in horizontal directions.



Legend: D- TDV diffuser, E- equipment, F -furniture, O- occupants, L- lighting, R- return grille, T- table, U- UFAD diffuser, W- window.

Figure 8 Layouts used in the numerical simulations: (a) office, (b) classroom, and (c) workshop

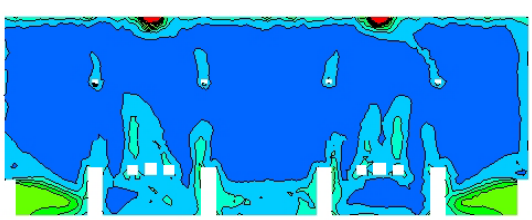
Table 8 lists further detailed information on the boundary conditions used in the numerical simulations. Each type of space used three different air distribution systems, which made a total of nine cases. Hereafter, this paper uses TDV to stand for the traditional displacement ventilation system using the perforated diffusers, L-UFAD for the under-floor air distribution system with low-height-throw diffusers (swirl diffuser), and H-UFAD for the under-floor air distribution system with high-height-throw diffusers (linear diffuser). In the CFD simulations, the perforated-floor-panel diffusers were excluded because the experiment indicated that the diffuser performed in a similar way to the perforated TDV diffusers.

Table 8 Thermo-fluid conditions for the three indoor spaces for cooling

Indoor space type	Diffuser type	Total heat generated [W] ([Btu/h])			Ventilation rate/diffuser number [ACH]/[-]	Supply temperature [^{o}C] ([^{o}F])
		Occupants	Equipment	Lights		
Office	TDV	150 (512)	200 (682)	204 (696)	11.5/1	19.5 (67.1)
	L-UFAD	150 (512)	200 (682)	204 (696)	8/3	16.5 (61.7)
	H-UFAD	150 (512)	200 (682)	204 (696)	8/3	16.5 (61.7)
Classroom	TDV	1,875 (6,400)	0	510 (1,740)	7.5/4	19.0 (66.2)
	L-UFAD	1,875 (6,400)	0	510 (1,740)	6/16	17.2 (62.9)
	H-UFAD	1,875 (6,400)	0	510 (1,740)	6/16	17.5 (63.5)
Workshop	TDV	5,600 (19,096)	2,240 (7,673)	2,200 (7,506)	6/8	19.5 (67.1)
	L-UFAD	5,600 (19,096)	2,240 (7,673)	2,200 (7,506)	4.5/22	17.0 (62.6)
	H-UFAD	5,600 (19,096)	2,240 (7,673)	2,200 (7,506)	4.5/22	17.0 (62.6)

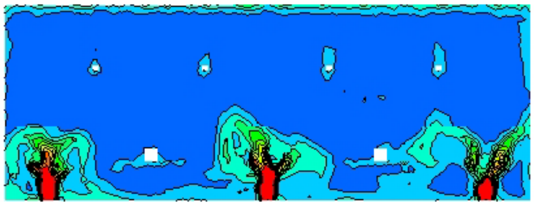
Numerical simulation results

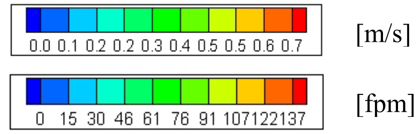
This paper used the workshop case as an example to illustrate the CFD results. The numerical simulations provided detailed flow patterns. For instance, Figures 9, 10, and 11 show the distributions of air velocity, air temperature, and air distribution effectiveness created by the TDV, L-UFAD, and H-UFAD systems under the cooling mode. The figures are for the mid-section (B-B′ section in Figure 8(c)) of the workshop. Figure 9 shows that the workshop maintained a very low air velocity, less than 0.1 m/s (19.5 fpm) in the upper part of the room regardless of air distribution system used. Also, the air velocity distribution created by the TDV system in the lower part of the room was very low, ranging from 0.0 to 0.2 m/s (0 to 40 fpm), except for the regions near the diffusers. However, the H-UFAD system generated long throws and created high air velocity in the proximity of the diffusers, which could lead to thermal discomfort on the part of the occupants. The air velocity in the lower part of the workshop with the L-UFAD system was only high at the diffusers. Thus, the risk of thermal discomfort was low.



(a)

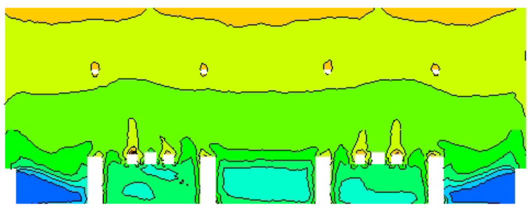
(b)

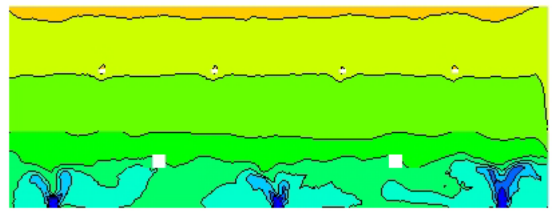




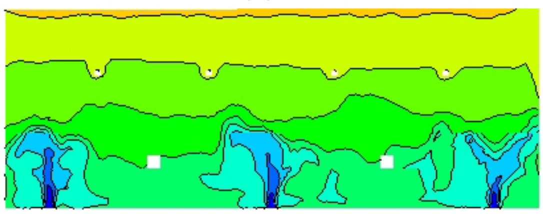
(c)

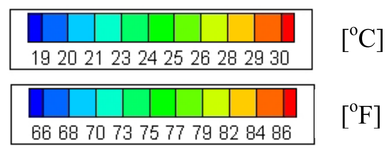
Figure 9 Air velocity distributions under cooling mode in the mid-section of the workshop with (a) TDV diffusers, (b) L-UFAD diffusers, and (c) H-UFAD diffusers





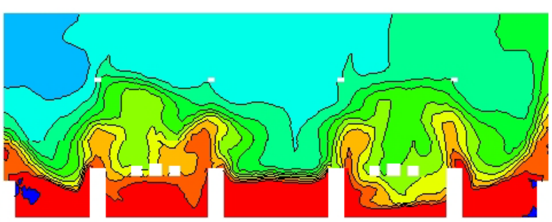
(a) (b)

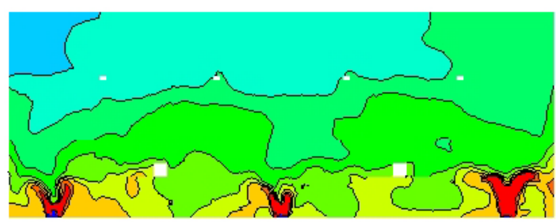




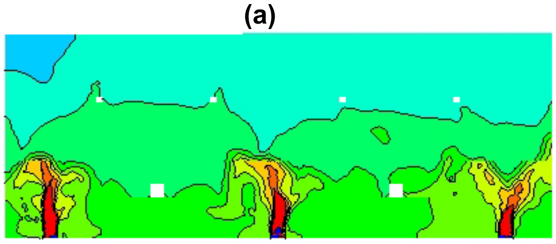
(c)

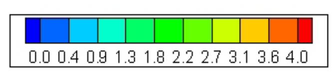
Figure 10 Air temperature distributions under cooling mode in the mid-section of the workshop with: (a) TDV diffusers, (b) L-UFAD diffusers, and (c) H-UFAD diffusers





(a) (b)





(c)

Figure 11 Distributions of air distribution effectiveness under cooling mode in the mid-section of the workshop with (a) TDV diffusers, (b) L-UFAD diffusers, and (c) H-UFAD diffusers

Figure 10 compares the air temperature distributions formed by the three types of air distribution systems in the workshop under the cooling mode. The air temperature stratifications in the upper part of the room were similar for all three cases. However, there were significant differences in the lower part of the room depending on diffuser type. The figure indicates that the temperature stratification was large with the TDV and L-UFAD systems in the lower part of the room in this section. The diffusers in the L-UFAD system mixed the supply air with the surrounding air, so the air temperature in the vicinity of the diffusers was quite uniform. The H-UFAD system created the largest cold regions, which could cause a draft.

Figure 11 illustrates the air distribution effectiveness (*E*) under the cooling mode. To evaluate the ventilation performance, air distribution effectiveness was employed. Air distribution effectiveness is defined as $(C_e\text{-}C_s)/(C\text{-}C_s)$. The *E* in the upper part of the room was rather uniform, with a value close to 1.0, which was the same as that of the mixing ventilation system. On the other hand, in the lower part of the room where the occupants were, the air distribution effectiveness of the TDV and L-UFAD systems was much higher than that of the mixing ventilation system. The TDV system generated a very high *E* near the floor because the supply air was not mixed with the surrounding air as in the L-UFAD and H-UFAD systems. Even for the H-UFAD system, it can create high air distribution effectiveness in the lower part of the room. This is because the workshop ceiling was very high (4.5 m or 14.8 ft). The high throws from the diffusers were relatively low compared with the ceiling height.

Figures 12(a), (b), and (c) show the averaged vertical air temperature and air distribution effectiveness profiles with three air-distribution systems under the cooling mode. The TDV systems had the largest temperature gradients in the three indoor spaces studied. Since the air from the supply diffusers was not mixed with the surrounding air, the temperature gradients in the occupied zones were particularly large. The H-UFAD systems had the smallest gradients because the diffusers could mix well the supply air with the surrounding air.

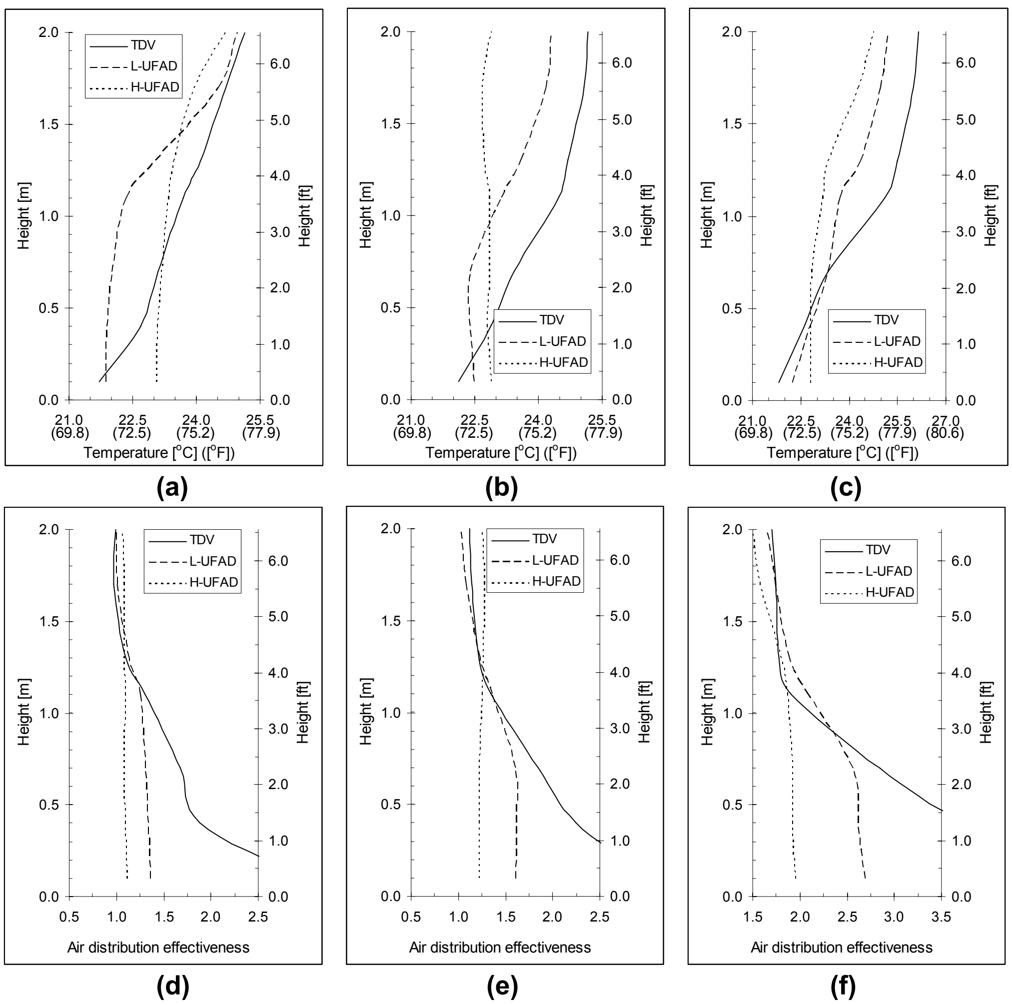


Figure 12 Averaged air temperature and air distribution effectiveness profiles at three different heights under cooling mode: (a) air temperature, the office, (b) air temperature, the classroom, (c) air temperature, the workshop, (d) air distribution effectiveness, the office, (e) air distribution effectiveness, the classroom, and (f) air distribution effectiveness, the workshop

Table 9 shows the vertical air temperature difference between the head and ankle level for a sitting person and that for a standing person. According to the ASHRAE Handbook Fundamentals on thermal comfort, the temperature difference should not exceed 3 ^{o}C (5 ^{o}F). The temperature differences were thus acceptable for most of the cases except were quite high for the TDV system in the workshop. This problem could be remedied by increasing supply air temperature and flow rate.

Table 9 Air temperature difference between head and ankle level and air distribution effectiveness in the three indoor spaces with different air distribution systems

Indoor space type	Air distribution systems	Air temperature differences				Air distribution effectiveness		
		$\Delta T_{sitting}$		$\Delta T_{standing}$		$E_{sitting}$	$E_{standing}$	E_{ave}
		[^{o}C]	[^{o}F]	[^{o}C]	[^{o}F]			
Office	TDV	1.99	3.58	2.98	5.36	1.30	0.96	1.10
	L-UFAD	0.47	0.85	2.63	4.73	1.26	1.02	1.12
	H-UFAD	0.27	0.49	0.87	1.57	1.10	1.08	1.09
Classroom	TDV	2.35	4.23	2.93	5.27	1.34	1.14	1.33
	L-UFAD	0.65	1.17	1.70	3.06	1.37	1.09	1.32
	H-UFAD	-0.05	-0.09	-0.20	-0.36	1.26	1.29	1.26
Workshop	TDV	3.05	5.49	3.81	6.85	1.91	1.75	1.82
	L-UFAD	1.46	2.63	2.72	4.90	2.10	1.74	1.90
	H-UFAD	0.40	0.72	1.49	2.68	1.88	1.59	1.76

$\Delta T_{sitting}$: Temperature difference between 0.1 m (0.3 ft) and 1.1 m (3.6 ft) from the floor
$\Delta T_{standing}$: Temperature difference between 0.1 m (0.3 ft) and 1.7 m (5.6 ft) from the floor
$E_{sitting}$: *E* at the breathing level of sitting occupants (1.1 m (3.6 ft) from the floor)
$E_{standing}$: *E* at the breathing level of standing occupants (1.7 m (5.6 ft) from the floor)
E_{ave} : Averaged *E* in the breathing zone

Figures 12(d), (e), and (f) illustrate the averaged air distribution effectiveness for the three indoor spaces with the three air distribution systems. In the lower part of the room, the air distribution effectiveness was very different, with the TDV system creating the highest and the H-UFAD system, the lowest. Very similar to our explanation for the temperature stratifications, the differences were mainly caused by the diffusers and their locations. For a sitting occupant, the TDV system was the best and the H-UFAD the worst in terms of air distribution effectiveness. However, at 1.7 m (5.6 ft) from the floor, the *E* was close to each other. Brohus and Nielsen (1996) showed that a standing person actually breathed air below the breathing zone in a space with displacement ventilation. This is due to the thermal plume generated from the thermal buoyancy of the body. Thus, air distribution effectiveness below 1.7 m (5.6 ft) is more important in evaluating the ventilation performance of TDV and UFAD systems.

Table 9 also compares the air distribution effectiveness at sitting and standing levels quantitatively. The table gives the averaged *E* value between 1.0 to 1.7 m for the office and workshop and between 0.7 and 1.7 m for the classroom. The averaged *E* takes into consideration the reality in these spaces where some occupants may be sitting and others may be standing. The height is lower for the classrooms to include the presence of small children. Clearly, all three air-distribution systems generated a better *E* for sitting occupants than for standing ones. The performance of the TDV system was almost the same as that of the L-UFAD system. The H-UFAD system was the worst among the three systems, but it was still better than that of the perfectly mixing ventilation systems. The averaged air distribution effectiveness for the TDV and L-UFAD systems ranged from 1.1 for the office to 1.8 for the workshop. Please note that the ceiling heights for the office, classroom, and workshop were 2.43, 3.3, and 4.5 m (8, 10.8, and 14.8 ft), respectively. The *E* value is in proportion to the ceiling height.

The numerical simulations in this study provided more detailed data than the experimental measurements to compare the thermal comfort and the ventilation performance of various air distribution systems with different types of diffusers. The results from the numerical simulations agreed well with those from the experiments. The TDV system created the largest vertical temperature gradient and lowest air velocity, while the H-UFAD system created the smallest temperature gradient and the highest velocity. However, the temperature gradients were generally acceptable for thermal comfort. Thus, the H-UFAD system has the highest draft risk. The TDV and the L-UFAD systems gave high air distribution effectiveness in the breathing zones. Even the H-UFAD system that was poorer than the other two systems was better than complete mixing systems.

CONCLUSIONS

This study compared the airflow, air temperature, and contaminant or air distribution effectiveness distributions in indoor spaces between the TDV and UFAD systems using both experimental and numerical approaches. The experimental approach gave the most realistic data to describe the ventilation performances of the systems but had low data resolution. The numerical simulations, well calibrated by the experiment, can provide detailed information on the ventilation performance and can easily show different sizes of indoor spaces, such as an office, a classroom, and a workshop. The results obtained with the CFD program are in good agreement with the experimental data.

The experimental results show that the perforated-corner TDV diffuser, swirl diffuser, and perforated-floor-panel diffuser created low air velocity in the occupied zone. However, the perforated-corner TDV and perforated-floor-panel diffusers could generate a high temperature difference between the head and ankle level of an occupant. The linear diffuser created the highest velocity in the occupied zone, making the potential draft risk high. The TDV and UFAD systems had better ventilation performance than the mixing ventilation system in cooling mode. For heating mode, the TDV and UFAD system created mixing conditions except in the vicinity of the floor.

The results from the numerical simulations for an office, a classroom, and a workshop confirm the findings from the experiment. The three systems created higher air distribution effectiveness than did the mixing ventilation system. The effectiveness for TDV and L-UFAD systems was similar but higher than that for the H-UFAD system. The air distribution effectiveness seems in proportion to the ceiling height. In addition, the H-UFAD systems using linear diffusers had a high draft risk.

REFERENCES

ASHRAE. 2005. *ASHRAE Handbook-Fundamentals*. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Bauman, F., and A. Daly. 2003. *Underfloor air distribution (UFAD) design guide*. Atlanta, GA: ASHRAE, 2003.

Brohus, H., and P. V. Nielsen, 1996, Personal exposure in displacement ventilated rooms. *Indoor Air* 6 (3):157-167

Chao, C.Y.H., and M.P. Wan, 2004, Airflow and air temperature distribution in the occupied region of an underfloor ventilation system, *Building and Environment* 39(7): 749-762

Chen Q., and A. Moser. 1991. Simulation of a multiple nozzle diffuser. *Proceedings of the Twelfth AIVC Conference* 2: 1-14.

Chen, Q., and L. Glicksman. 2003. System performance evaluation and design guidelines for displacement ventilation. *Atlanta, GA: ASHRAE 2003.*

Hu, S., Q. Chen, and L. Glicksman. 1999. Comparison of energy consumption between displacement and mixing ventilation systems for different U.S. buildings and climates. *ASHRAE Transactions* 105(2): 453-464.

Hu, J.S., M.P. Wan, C.Y. Chao, and A. Law. 2003. A Application of underfloor ventilation system: Some design considerations. *Proceedings of the 2003 4th International Symposium on Heating, Ventilating and Air Conditioning 2003*: 354-361.

Im, P., A. Cho, and J. Haberl. 2005. Literature review on underfloor air distribution (UFAD) system. Technical Report of the Energy Systems Laboratory, Texas A&M University System, USA.

Kobayashi, N. 2001. Floor-Supply Displacement Ventilation System. Master's thesis, Department of Architecture. M.I.T., Cambridge, MA.

Kobayashi, N., and Q. Chen. 2003. Floor-Supply Displacement Ventilation in a Small Office. *Indoor and Built Environment* 12(4): 281-291.

Lemarie, A.D. 1993. Room air and contaminant flow, evaluation of computational methods. subtask-1 summary report, International Energy Agency Annex 20: Air Flow Patterns within Buildings.

Lin, Z., T.T. Chow, C.F. Tsang, K.F. Fong, and L.S. Chan. 2005. CFD study on effect of the air supply location on the performance of the displacement ventilation system. *Building and Environment* 40(8): 1051-1067.

McDonell, G. 2003. Underfloor & displacement: Why they're not the same. *ASHRAE Journal 45(7)*: 18-24.

Srebric J and Q. Chen. 2002. Simplified numerical models for complex air supply diffusers. *HVAC&R Research* 8(3): 277-294.

Versteeg, H.K., and W. Malalasekera. 1999. *An introduction of computational fluid dynamics.* England: Longman Scientific & Technical.

Yakhot, V., S. Orzag, S. Thangam, and T. Gatski. 1992. Development of turbulence models for shear flows by a double expansion technique. *Phys. Fluids* 4(7): 1510-1520.

Yuan, X., Q. Chen, and L. Glicksman. 1998. Critical review of displacement ventilation. *ASHRAE Transactions* 104(1A): 78-89.

Yuan, X., Q. Chen, and L. Glicksman. 1999. Performance Evaluation and Development of Design Guidelines for Displacement Ventilation. Report of ASHRAE Research Project - RP-949. M.I.T., USA

Zhai, Z. and Q. Chen. 2004. Numerical determination and treatment of convective heat transfer coefficient in the coupled building energy and CFD simulation. *Building and Environment* 39(8): 1001-1009

Zhang, Z., W. Zhang, Z. Zhai, and Q. Chen. 2007. Evaluation of various turbulence models in predicting airflow and turbulence in enclosed environments by CFD: Part 2 - Comparison with experimental data from literature. *HVAC and R Research* 13(6): 871-886.