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Thermal environment in indoor spaces with under-floor air distribution systems: 1. Impact of design parameters (1522-RP)

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The Under-Floor Air Distribution (UFAD) system creates a cleaner indoor environment and provides enough energy saving potential for it to have become increasingly popular in North America. The system also generates thermal stratification in an occupied zone since it supplies cold air directly to the occupied zone of an indoor space. This study was to identify the impact of different design parameters on thermal stratification. Our investigation used the orthogonal method to identify the most important parameter and to establish a set of test cases for the parametric study. The parametric study used both experimental measurements in an environmental chamber and numerical simulations by computational fluid dynamics (CFD). The environmental chamber was used to simulate an office, a classroom, and a conference room with swirl, square, and linear diffusers. The experimental data was used to validate the numerical results by CFD. These results indicate that the swirl diffusers created the largest thermal stratification while the linear diffusers provided almost uniform distribution of the air temperature in the occupied zone. The swirl diffuser can create a very good mixing in the occupied zone so the supply air temperature can be lower. The thermal stratification was smaller when the cooling load was higher in indoor spaces such as conference rooms.

Introduction

Stratified air-distribution systems, such as traditional displacement ventilation (TDV) systems (Chen and Glicksman 2003) and under-floor air distribution (UFAD) systems (Lau and Chen 2007), have recently become popular for buildings in North America. The TDV and UFAD systems supply cool and fresh air into the occupied zone of a room and create thermal stratification in that space. Many previous studies have reported that these systems have the potential to reduce energy demand as well as to provide better indoor air quality (Chen and Glicksman 2003, Bauman and Daly 2003, Lau and Chen 2006, Lee et al. 2009), compared with mixing ventilation systems. This is because the conditioned air from the diffusers of the systems stays in the lower part of the room due to thermal buoyancy. The existence of the temperature stratification means a higher exhaust air temperature that can reduce the supply airflow rate if the supply air temperature remains the same. Finding out how to determine the supply airflow rate is essential for designing stratified systems with acceptable thermal comfort.

Chen and Glicksman (2003) developed a method on how to determine the supply airflow rate for TDV systems under cooling mode. But the determination of the airflow rate for UFAD systems is more complicated due to the air momentum generated from the diffusers. Compared with TDV systems, UFAD systems have many more types of air supply diffusers that generate very different air discharging angles, mixing strength, and throws. Several methods for estimating the supply airflow rate from the UFAD systems have been developed by (Lee et al. 2009, Bauman et al. 2007, Loudermilk 1999, Sodec and Craig 1991, York 1999). All of these methods need to estimate heat gains above the occupied zone of a space and the return air temperature and do not normally calculate the same supply airflow rate. Therefore, it is necessary to develop a method based on solid research for estimating the supply airflow rate in UFAD systems.

In order to calculate the supply airflow rate, the thermal stratification in a room with a UFAD system should be correctly estimated under design conditions since the stratification affects the exhaust air temperature and thermal comfort. Several researchers (Lee et al. 2009, Kobayashi and Chen 2003, Lin et al. 2005) have found that diffuser type has a significant impact on the temperature distribution of UFAD systems. Other parameters, such as occupancy patterns (Rock 1995), space types (Akimoto 1999), supply air temperature and thermal load (Di Tommaso 1999, Xu 2001, Lin et al. 2005a), and air supply conditions (Xing 2001), have also been found to be important in determining the thermal stratification. Lee et al. (2009) studied these parameters for TDV and UFAD systems and created a database of air distribution effectiveness under various design parameters for six different indoor spaces with swirl diffusers. Based on the effort in the previous studies, the objective of this investigation was to systematically study the effect of important design parameters on the supply airflow rate, such as diffuser type, diffuser number, space type, and supply air temperature. The research results obtained from this investigation will be reported in two papers. This paper is the first of the two papers and it focuses on the impact of different design parameters on thermal stratification in indoor spaces with a UFAD system. The companion paper (Xue et al. 2012) will report on how to determine the supply airflow rate.

Research method

Orthogonal method. To systematically study the design parameters for indoor spaces with UFAD systems, the total number of cases can be several hundred. But this would require a multiple-person effort over many years, which may not be economically feasible. Thus, it is essential to distinguish the most important design parameters that need to be studied in depth.

This study employed the orthogonal method (Taguchi and Rajesh 2000) to evaluate several design parameters in order to establish a test matrix for a parametric study of UFAD systems. Many previous studies in the literature have listed a variety of design parameters that may affect the thermal environment in indoor spaces with UFAD systems. However, very little research has examined which parameters are the most important. The orthogonal method is a statistical method often used for parameter selection. By using this method, one can identify important parameters with fewer tests.

For example, to test four parameters with three levels of values (high, medium, and low), the total number of cases would normally be 3 × 3 × 3 × 3 = 81 by using a conventional listing method. But if the orthogonal method is used, one can identify the most important parameters with nine cases, as shown in Table 1. Table 1 is called $L_9(3^4)$, where 4 means that the table is applicable for a maximum of 4 parameters; 9 stands for the total number of cases which need to be performed in order to identify the most important parameters; and 3 stands for the 3 levels of values for each parameter.

The importance of each parameter can be calculated by using Eqs. (1) and (2). S_z is an index used for evaluating the importance of a parameter. The larger the S_z for a parameter, the more important the parameter.

$$S_z = \sum_{j=1 \text{ to } 3} \left(\frac{(\sum Y_{zj})^2}{\text{Number of cases for each level}} \right) - CF \tag{1}$$

$$CF = \left(\sum_{i=1 \text{ to } 9} Y_i \right)^2 / \text{ Total number of test} \tag{2}$$

where Y are the most important results obtained from the four parameters, *i* is the row number, *j* is the column number, and *z* is the parameters. This study used the air temperature difference between the head and ankle level of a standing person, $T_{head} - T_{ankle}$, and the average air temperature in the occupied zone, $T_{occupied\ zone}$, as the Y since both of them are crucial for the thermal comfort in an indoor space with a UFAD system.

Parametric study. As soon as the orthogonal method identifies the most important parameters, they need to be studied in detail in the parametric study, which can use two main approaches for the study of the effect of these design parameters on the thermal environment of indoor spaces with UFAD systems. The two approaches are experimental measurements in an environmental chamber and numerical simulations by CFD.

It is well known that experimental measurements of an indoor thermal environment give the most realistic information. But collecting the air velocity, air temperature, contaminant concentrations, and relative humidity at key locations of the indoor space using this method can be very time consuming and costly. This is because UFAD systems still involve many parameters even after selection by the orthogonal method. Furthermore, the number of sensors installed in an indoor space is always limited, so experimental measurements may not provide very detailed thermal environment information. It is also not easy to change the size and layout of the room in the measurements, but they are parameters influencing UFAD system performance.

The numerical approach by CFD simulations solves a set of conservation equations for flow, energy, and species concentrations in an indoor space. The simulations can generate quickly detailed thermal environment information in a space with a UFAD system at very little cost. Thus, the CFD simulations are ideal for testing the cases identified by the orthogonal method. Although the CFD sounds powerful, it has uncertainties in the results due to the use of a turbulence model and other approximations to solve the flow (Zhang et al. 2007). Since no universal turbulence model is available for the flow in an indoor space with a UFAD system, it is important to validate the numerical results with the corresponding experimental data. This study used the Re-Normalization Group (RNG) k-ε model (Yakhot et al. 1992), which is recommended especially for indoor airflows (Zhang et al. 2007). With the validated CFD program, one can perform a systematic study of the design parameters on the thermal environment conditions in indoor spaces with UFAD systems. Therefore, this investigation conducted a limited number of experimental measurements to generate the experimental data to validate the CFD program and use this program to conduct the parametric study.

Orthogonal tests

Many studies in the literature have listed some design parameters which may have an influence on the ventilation performance of UFAD systems. The diffuser type, such as the swirl, square, and linear diffusers often used in North America, has an influence on UFAD system performance. Since the supply air temperature and airflow rate from a diffuser are inter-related, they should be considered together as another parameter. The space types with different cooling loads are important since the cooling load and corresponding total airflow rate are critical for determining the thermal environment of UFAD systems. This investigation selected three indoor spaces: an office, a conference room, and a classroom, which also represent different levels of cooling load. The conference room had the highest cooling load as it had more occupants, equipment, and laptops compared to the other spaces, while the office and classroom each had a similar cooling load. However, the locations and heights of the heated objects in the office were different from those in the classroom. In addition, this investigation added zone type (interior and exterior zones) as another parameter as they have a very different cooling load.

Therefore, the design parameters tested in this investigation were diffuser type, diffuser number, air temperature difference between supply and return grille ($T_r - T_s$), indoor space type, and zone type. Except for zone type, which only has interior and exterior ones, other parameters should be tested with multiple levels. If three levels (variations) are tested for diffuser type, diffuser number, $T_r - T_s$, and space type for interior and exterior zones, a total of 3×3×3×3×2=162 cases need to be studied. In fact, some of the parameters may not be very important, so it may not be necessary to test all 162 cases. Therefore, this investigation used the orthogonal method (Taguchi and Rajesh 2000) to distinguish the most important parameters that have a major impact on the ventilation performance of UFAD systems.

Selections of the range of each parameter are critical for the orthogonal method. Each parameter should have a wide range, but an undesirable variation in the parameter would generate uncertainties. The selection of the range often needs experience and may involve a few side tests, so this is not always straightforward. Table 2 lists the final selection of the cases used in the orthogonal tests.

Table 3 shows the S_z calculated from the orthogonal tests. Among the parameters tested, $T_r - T_s$ had the highest S_z. The tests meant that $T_r - T_s$ was the most important parameter and so should be tested with more variations (high, medium, and low) than the other parameters. Since the swirl, square, and linear diffusers are often used for UFAD systems, all of them should also be further tested. The diffuser number needs only to be further tested in two variations (high and low) as its S_z was not very high. This investigation still plans to test three space types (offices, classrooms, and conference rooms) that represent different cooling loads. As a result, the orthogonal tests reduced the total number of cases from 162 to 54 (3 diffuser types × 2 variations of diffuser number × 3 variations of $T_r - T_s$ × 3 space types). This was a very remarkable reduction of the total number of cases just by only conducting a handful of tests using the orthogonal method.

From the orthogonal tests, this study was able to design a test matrix for the parametric study that could study the impact of the design parameters on the thermal environment in indoor spaces with UFAD systems. Table 4 illustrates the design of the test matrix that had 18 for each space type. The total number of cases for the three indspace types was then 54. We agreed with our sponsor to test both interior and exterior zones, which were again for different cooling loads due to the solar heat gains through the windows and walls. The total number of cases finally studied was 108, but 54 cases would have been sufficient.

Impact of design parameters on a thermal environment with UFAD systems

This investigation used both experimental measurements and computer simulations to study the impact of design parameters on a thermal environment in rooms with UFAD systems. The experimental measurements were conducted in an environmental chamber that simulated a small office, a small conference room, and a quarter of a classroom, as shown in Figures 1(a), (b), and (c), respectively. The room dimensions were 4.8 m long × 4.2 m wide × 2.43 m high.

This study used three different types of diffusers as shown in Figure 2. The swirl diffusers generated low-height throws so that the air velocity from the diffusers decayed and mixed quickly with the surrounding air. The linear diffusers generated a high momentum so that the supply air could 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). The square diffusers had four small grilles with inclined vanes so that the diffusers could create various flow patterns depending on how each grille was placed. Thus, the square diffusers could generate either low-height-throws or high-height-throws. This investigation set up the diffusers to spread cold air at a maximum discharging angle.

Table 5 summarizes the range of parameters, which varied for each type of indoor space. For each standard case, the parameters were set so as to keep a comfortable temperature range, 22 ~ 24^{o}C (72 ~ 75^{o}F), in the occupied zone. In addition, the temperature difference between head and ankle level was controlled so as to make it less than 3 K (5^{o}F) for most cases in order to prevent thermal discomfort.

As shown in Table 4, this investigation conducted three cases of the experiment for each type of indoor space. The measurements were for interior and exterior zones. Therefore, the total number of experimental cases was 18. The data measured in the experiment were air velocity, air temperature, and contaminant concentration. A tracer gas was used in the experiment to simulate a gaseous contaminant released from the breathing of an occupant. The measurements in the environmental chamber were conducted at nine locations, as illustrated in Figure 3(d). In these locations, the air velocity, temperature, and tracer-gas concentration were measured at multiple points from floor to ceiling. Due to the limited space available in this paper, Figures 3(a), (b), and (c), respectively, compare the CFD simulation results along the room height with the corresponding experimental data for the office with the swirl diffusers (Refer to Case 1 in Table 4). The results were typical among the 18 cases measured, and the figures show the representative comparison. The agreement between the CFD results and the experimental data was rather good. Thus, the CFD tool was regarded as validated and can be used to study the thermal environment in buildings with UFAD systems. The results shown below were from the CFD simulations. Although not shown in this paper, our experimental data led to the same conclusions.

Different diffusers generated various throw heights and mixing effects in the lower part of the indoor space with the UFAD system. Figure 4(a) shows the impact of the diffuser type on the temperature gradient of the office, with two diffusers at an airflow rate of 200 m^3/h (117 CFM) and $T_r - T_s$ of 5 K (9^{o}F). The linear diffusers created the smallest thermal gradient, while the swirl diffusers had the largest thermal gradient. Figure 4(b) presents similar results for the quarter of the classroom. The high throw generated by the linear diffusers kept a relatively high air temperature in the lower part of the room. Since the swirl diffusers had a low throw but a large spreading angle, the cold air from the diffusers mixed rapidly with the room air and stayed in the lower part of the room. The air temperature at ankle level was the lowest due to the large thermal gradient. The square diffuser can generate various throw heights and spreading angles. But this investigation used only one configuration with the maximum spreading angle. Its performance was between the other two diffusers. The temperature gradients created by the three diffusers were nearly the same for the conference room, as shown in Figure 4(c). This is because the conference room had a much larger cooling load compared with that of the office and classroom, so the supply airflow rate became higher.

Our investigation studied interior and exterior zones separately. Figure 5 depicts how the zone type could affect the thermal gradient in rooms with UFAD systems. The figure compares the results only for an office with an airflow rate of 200 m^3/h (117 CFM) in an interior zone and 280 m^3/h (165 CFM) in an exterior zone. Since the exterior zone had a high cooling load, the airflow rate per diffuser was higher than that for the interior zone. Thus, the diffusers used in the exterior zone had a strong momentum in the lower part of the room, and the thermal gradient in the lower part of the room was almost uniform among all the diffuser types. However, the air temperature difference between the head and ankle level in the exterior zone was similar to that of the interior zone since the supply air temperature for both zones was the same. This also confirmed that the supply air temperature was the most important design parameter as found by the orthogonal method. Figure 5(b) shows the air temperature in the upper part of the room. Above the occupied zone, the air was more stratified than that of the interior zone. The exterior zone had a high cooling load that generated ascending airflow. Therefore, the heated air from the occupants, equipment, and enclosures could move up to the ceiling level and stay in the upper part of the room. A part of the heat may be extracted directly through a properly positioned exhaust without increasing the supply airflow rate.

Figure 6 depicts the effect of the diffuser number or discharge velocity on the temperature difference between the head and ankle level of a standing person, $T_{head} – T_{ankle}$, in the interior zone of the rooms. The figure compares the results only for the office and classroom but with the three diffuser types and an airflow rate of 200 m^3/h (117 CFM). When the diffuser number was reduced from four to two, the supply air velocity increased significantly. As a result, all types of diffusers generated higher throws, and the $T_{head} – T_{ankle}$ reduced significantly. However, the figure also shows that the swirl diffusers were less affected by the diffuser number. This is because the swirl diffusers had a small opening area and the air momentum was sufficiently strong when the flow rate was low so it could mix with the surrounding air. An increase in the supply air momentum by the reduction of the diffuser number would not further increase the mixing.

Figure 7 presents the impact of the $T_r – T_s$ on the temperature difference between the head and ankle level of a standing person, $T_{head} – T_{ankle}$. Generally, a larger supply airflow rate implies a lower $T_r – T_s$ and a larger $T_{head} – T_{ankle}$, which can clearly be seen in the figure. However, the slope of the $T_r – T_s$ and $T_{head} – T_{ankle}$ relation was different for different diffusers. The swirl diffuser had the flattest slope so the diffuser could create a good mixing when the supply air temperature was low. The swirl diffuser has been proven to be the most effective in practice. The performance of the linear diffusers was similar to that of the square diffusers as their slopes were almost identical. If square diffusers or linear diffusers are used for UFAD systems, designers should be very careful to maintain $T_{head} – T_{ankle}$ at less than 3 K (5^{o}F).

As mentioned previously, this investigation considered separately the impact of different space types on an indoor thermal environment. Figure 8 shows the impact of three different indoor spaces on the $T_{head} – T_{ankle}$. The cases shown in the figure used the swirl diffusers. The office had the lowest cooling load and the conference room had the highest. In order to maintain the same thermal condition, the conference room needed a high supply airflow rate if the supply air temperature was the same. As a result, the conference room had the lowest $T_{head} – T_{ankle}$ since the supply air velocity from the diffusers was the highest, which can enhance the mixing of the supply air with the surrounding air in an occupied zone. The impact of space type on the $T_{head} – T_{ankle}$ was reflected by the cooling load.

Conclusions

This investigation studied the impact of design parameters on the thermal environment in indoor spaces with UFAD systems. The study led to the following major conclusions:

- The tests of the following design parameters on an indoor thermal environment would normally need 162 cases: diffuser type, diffuser number, air temperature difference between supply and return air, indoor space type, and zone type. By using the orthogonal method, this investigation was able to achieve the same results by conducting only 54 tests.
- Both experimental measurements in an environmental chamber and numerical simulations by CFD can be used to study the impact of the design parameters on the thermal environment. The two results look similar but the experimental measurements was more expensive and time consuming. The validated CFD tool can be used to study the impact of design parameters on a thermal environment for indoor spaces with UFAD systems.
- Among the three different diffusers tested, the linear diffusers created the lowest thermal stratification in the occupied zone, while the swirl diffusers created the highest stratification even though the air temperature gradient was acceptable for thermal comfort. The swirl diffuser was more desirable in mixing the supply air with the surrounding air to generate a comfortable condition when the supply air temperature was low. The swirl diffuser was also least sensitive to change in the supply airflow rate or diffuser number. Furthermore, the study found that, in indoor spaces with a high cooling load, UFAD systems would create a thermal environment with low temperature stratification.

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Table 1.Orthogonal test matrix ($L_9(3^4)$) for 4 parameters with 3 levels

Case No.	Parameter 1	Parameter 2	Parameter 3	Parameter4
1	Level 1	Level 1	Level 1	Level 1
2	Level 1	Level 2	Level 2	Level 2
3	Level 1	Level 3	Level 3	Level 3
4	Level 2	Level 1	Level 2	Level 3
5	Level 2	Level 2	Level 3	Level 1
6	Level 2	Level 3	Level 1	Level 2
7	Level 3	Level 1	Level 3	Level 2
8	Level 3	Level 2	Level 1	Level 3
9	Level 3	Level 3	Level 2	Level 1

Table 2. Cases for the orthogonal test matrix ($L_9(3^4)$)

Cases	Conditions						Results			
	Diffuser type	Diffuser number (#)	Cooling load		$T_r - T_s$		$T_{head} - T_{ankle}$ of a standing person		$T_{occupied\ zone}$	
			(W/m^2)	(W/ft^2)	(^{o}F)	(^{o}C)	(^{o}F)	(^{o}C)	(^{o}F)	(^{o}C)
1	Square	4	50	4.65	9.0	5.0	2.16	1.2	76.33	24.63
2	Square	3	40	3.71	11.7	6.5	5.32	3.0	74.24	23.47
3	Square	2	30	2.78	14.4	8.0	9.93	5.5	73.52	23.07
4	Linear	4	40	3.71	14.4	8.0	9.19	5.1	73.14	22.86
5	Linear	3	30	2.78	9.0	5.0	3.76	2.1	76.76	24.87
6	Linear	2	50	4.65	11.7	6.5	2.50	1.4	76.49	24.72
7	Swirl	4	30	2.78	11.7	6.5	5.58	3.1	74.24	23.47
8	Swirl	3	50	4.65	14.4	8.0	8.42	4.7	72.57	22.54
9	Swirl	2	40	3.71	9.0	5.0	1.35	0.8	77.45	25.25

Table 3. S_z calculated from the orthogonal tests for the parameters listed in Table 2

	Diffuser type	Diffuser #	Cooling load	$T_r - T_s$
$T_{occupied\ zone}$	0.34	1.00	0.04	6.57
$T_{head} - T_{ankle}$ of a sitting person	0.40	0.69	2.70	9.01
$T_{head} - T_{ankle}$ of a standing person	0.28	0.83	1.97	22.25

Table 4 Parametric matrix studied for each indoor space					
	Diffuser type	Diffuser number	T_r - T_s	Experiment	CFD
1	Square	2	Low	×	×
2	Square	2	Med	-	×
3	Square	2	High	-	×
4	Square	4	Low	-	×
5	Square	4	Med	-	×
6	Square	4	High	-	×
7	Swirl	2	Low	×	×
8	Swirl	2	Med	-	×
9	Swirl	2	High	-	×
10	Swirl	4	Low	-	×
11	Swirl	4	Med	-	×
12	Swirl	4	High	-	×
13	Linear	2	Low	-	×
14	Linear	2	Med	×	×
15	Linear	2	High	-	×
16	Linear	4	Low	-	×
17	Linear	4	Med	-	×
18	Linear	4	High	-	×

Table 5. Key thermal and flow conditions used for the indoor spaces				
Space type	Cooling load [w]	Supply flow rate (ACH)	Supply air temperature, [^{o}C] ([^{o}F])	Diffuser # [-]
Office	810	4.4 ~ 7.3	16.1 ~ 19.5 (61.1 ~ 67.1)	2 ~ 4
Conference room	955	5.8 ~ 9.4	16.2 ~ 19.4 (61.1 ~ 67.0)	2 ~ 4
Classroom	875	4.7 ~ 7.8	16.5 ~ 20.1 (61.7 ~ 68.2)	2 ~ 4

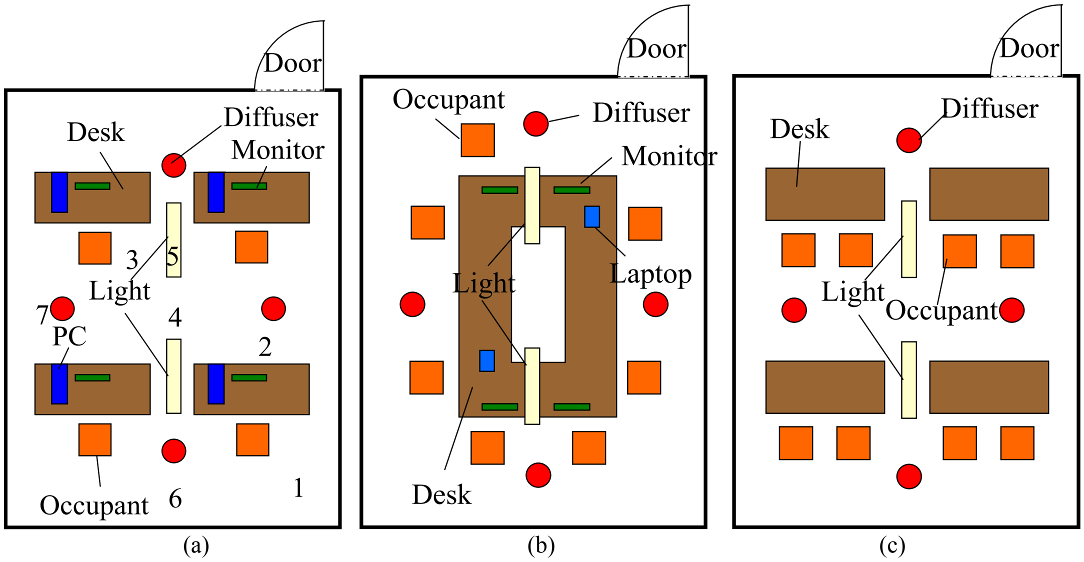

Figure 1. The space layouts simulated in an environmental chamber: (a) a small office, (b) a small conference room, and (c) a quarter of a classroom

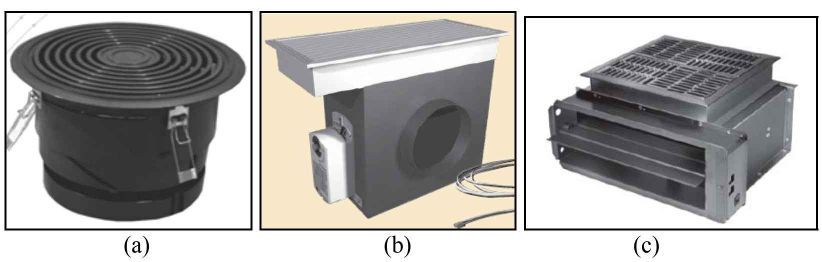

Figure 2 Diffusers used for the UFAD systems: (a) swirl, (b) linear, and (c) square

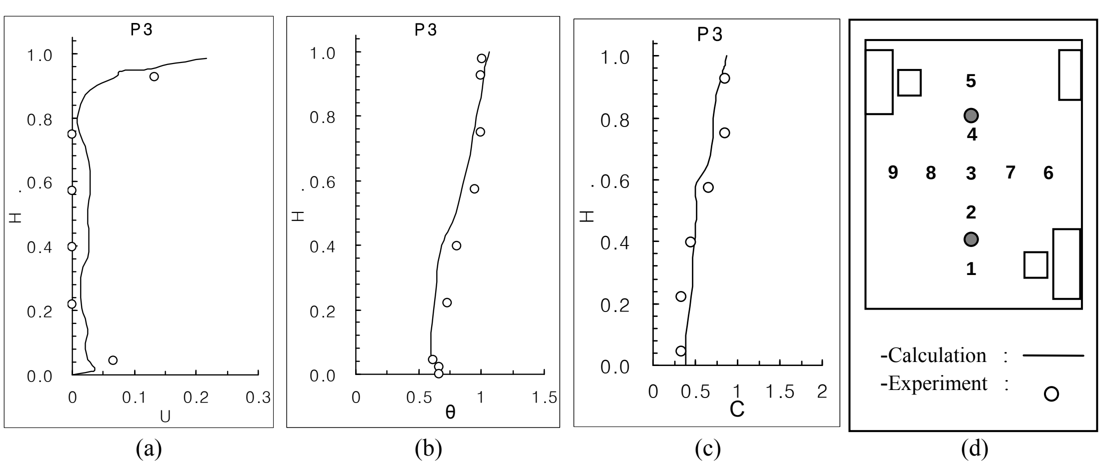

Figure 3. Validation of the CFD results by the experimental data in the center of the room (location 3), H=z/h, h=2.27m (7.44 ft): (a) air velocity profile, $u=U/U_s$, u_s = 1.5 m/s (295 fpm), (b) air temperature profile, $\theta=(T-T_s)/(T_r-T_s)$, and (c) contaminant concentration profile, $C=(c-c_s)/(c_r-c_s)$, (d) measuring locations in plane view.

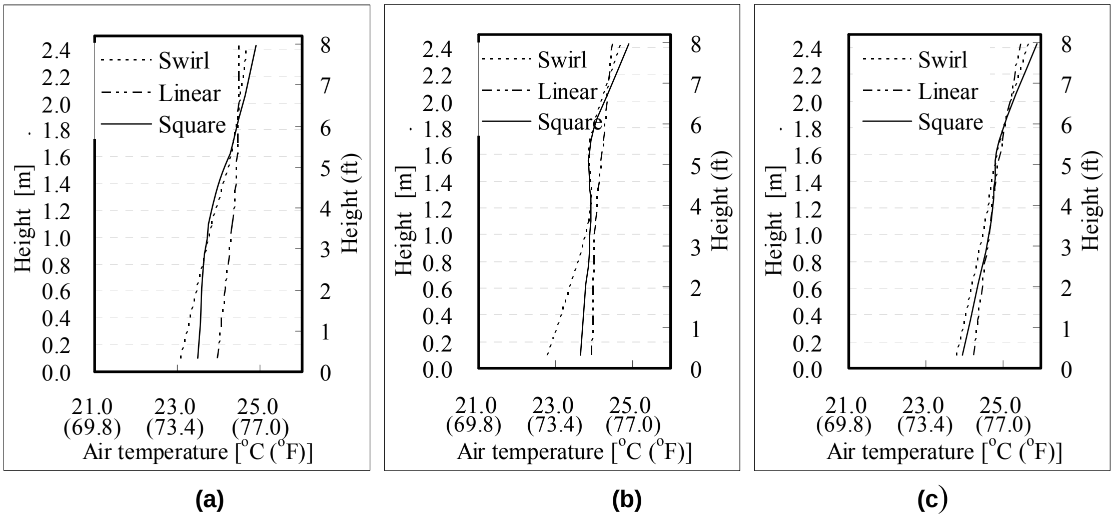

Figure 4. Impact of diffuser type on thermal stratification in the interior zone of: (a) office, (b) classroom, and (c) conference room.

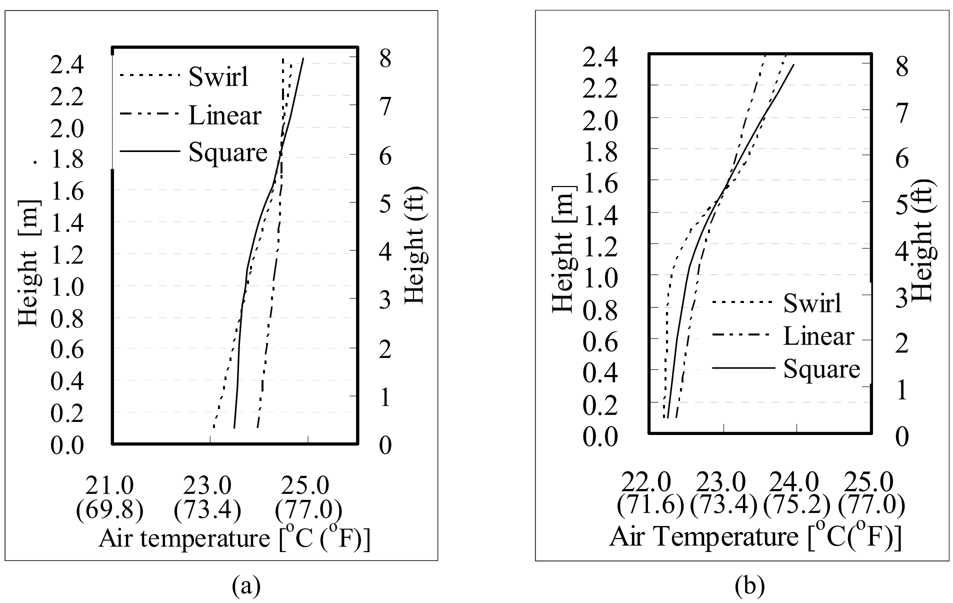

Figure 5. The impact of the diffuser type on the thermal stratification in an office: (a) interior zone and (b) exterior zone.

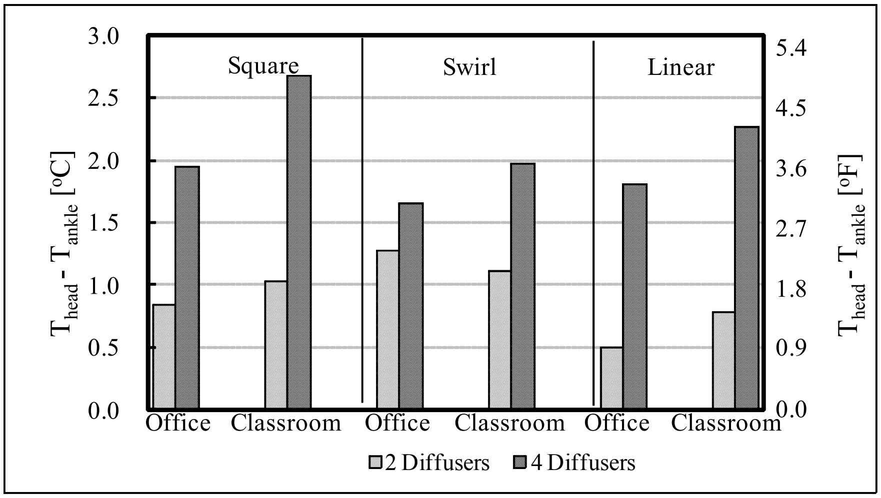

Figure 6. The impact of the discharge velocity on the temperature difference between the head and ankle level of a standing person in the interior zone of the rooms.

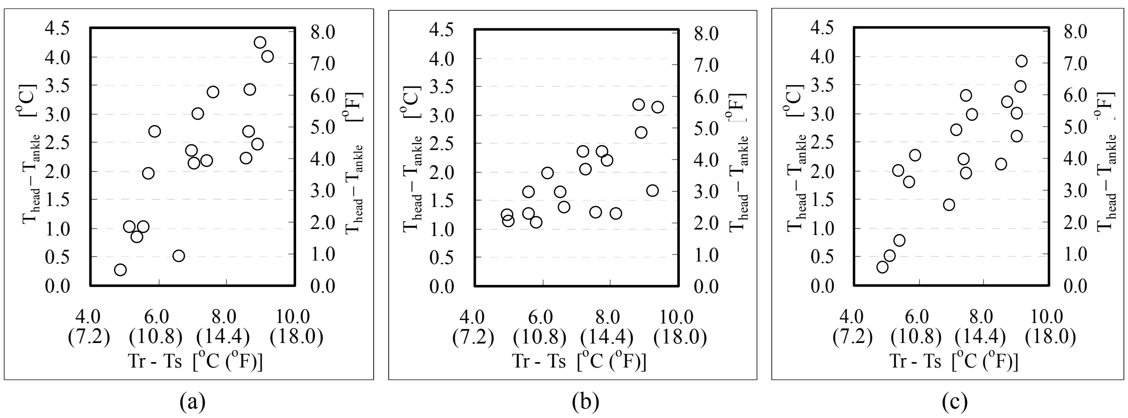

Figure 7. The impact of T_r – T_s on the temperature difference between the head and ankle level of a standing person in the interior zone of the rooms: (a) square diffuser, (b) swirl diffuser, and (c) linear diffuser.

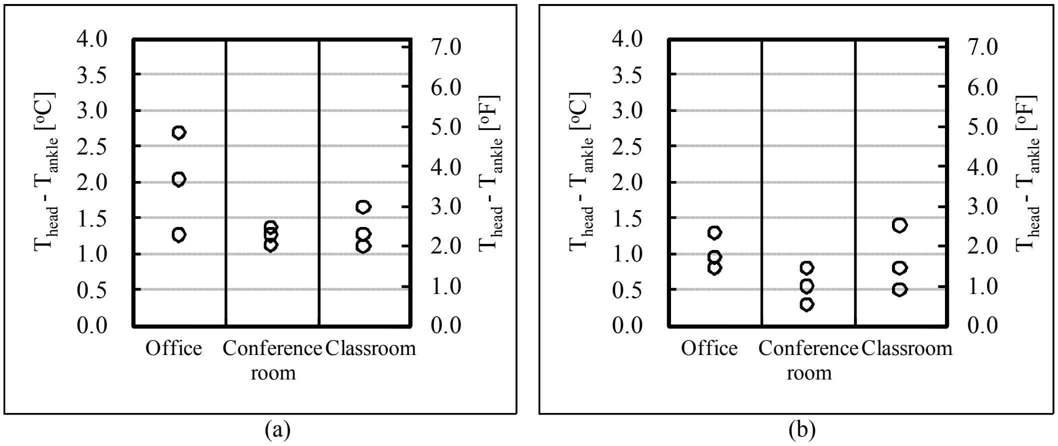

Figure 8. The impact of the space type on the temperature difference between the head and ankle level of a standing person: (a) interior zone and (b) exterior zone