Air Distribution Effectiveness with Stratified Air Distribution Systems

Kisup Lee
Student member ASHRAE

Zheng Jiang, PhD

Qingyan Chen, PhD
Fellow ASHRAE

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ABSTRACT

Stratified air distribution systems such as Traditional Displacement Ventilation (TDV) and Under-Floor Air Distribution (UFAD) systems have been known to provide better indoor air quality. This study examined the influence of several key design parameters on air distribution effectiveness by using a validated CFD program. The parameters studied were space type, diffuser number, supply air temperature, cooling load, return location, total airflow rate, and secondary heating system. Six indoor spaces were investigated to develop a database: classrooms, office spaces, workshops, restaurants, retail spaces, and auditoriums. The air distribution effectiveness at breathing zone was at 1.1 ~ 1.6 for offices, classrooms, restaurants and retail shops, and 1.6 ~ 2.0 for workshops and auditoriums. The spaces with a high ceiling such as workshops and auditoriums had higher air distribution effectiveness than those with a low ceiling. Thus, the stratified air distribution systems are better for spaces with a high ceiling. The air distribution effectiveness for the TDV and UFAD with low throw height was similar and was higher than that of UFAD with high throw height and mixing ventilation. A database was established containing 102 cases of the parametric study results. With this database, the investigation identified the six most important parameters to follow in developing a set of correlation equations for calculating air distribution effectiveness through statistical analysis. The air distribution effectiveness calculated by the equations was mostly within 10% of that for the corresponding case in the database.

INTRODUCTION

Stratified air distribution systems such as Traditional Displacement Ventilation (TDV) and Under-Floor Air Distribution (UFAD) systems are becoming popular because they can create better indoor air quality (Chen and Glicksman 2003, Bauman and Daly 2003). This is because they supply fresh air directly to the occupied zone at a temperature slightly lower than that of the air in the room. Due to the thermal buoyancy, the cold but fresh air can stay in the lower part of the room. In many cases, contaminant sources in the room are associated with heat sources, such as occupants, equipment, etc. The thermal plumes generated by the heat sources bring the contaminants to the upper part of the room since the exhausts are typically located at or near the ceiling level. Thus, the contaminants can be extracted directly through the exhausts without mixing with the fresh air. In addition, the thermal plume from an occupant induces the fresh air from the lower part of the room to the breathing level of the occupant. The air breathed by the occupant is rather clean. This has been further confirmed by our recent investigation reported in a companion paper (Lee et al. 2009).

The ventilation performance of the stratified air distribution systems has been taken into consideration by the ASHRAE standards through the air distribution effectiveness. For example, Table 6-1 of ANSI/ASHRAE Standard 62.1-2004 (ASHRAE 2004) defines the minimum required amount of outdoor air, V_{bz}, delivered to the space (or zone) for controlling contaminant concentration. Table 6-2 of the standard defines zone air distribution effectiveness, E_z, for different air distribution configurations. The outdoor airflow required at

Kisup Lee is a PhD candidate and **Qingyan Chen** is a professor in the Department of Mechanical Engineering, Purdue University, West Lafayette, Indiana. **Zheng Jiang** is a partner of Building Energy and Environment Engineering LLP, Lafayette, Indiana.

the zone (usually through the supply diffusers) is determined as V_{bz} (from Table 6.2) divided by E_z. Thus, the zone air distribution effectiveness plays an important role in determining the minimum required amount of outside air for a space.

The stratified air distribution systems, such as TDV systems and low-height-throw under-floor air distribution (L-UFAD) systems, are assigned with $E_z = 1.2$ in cooling mode. The low height throw is defined as a situation in which the air velocity from a supply jet decays to less than 0.3 m/s (60 fpm) at a height of 1.35 m (4.5 ft) above the floor. When the systems are used for heating, the E_z drops to 0.7 with ceiling return or 1.0 with floor return. High-height-throw under-floor air distribution systems (H-UFAD) are assigned with $E_z = 1.0$, where the air velocity from the supply jet is still higher than 0.3 m/s (60 fpm) at a height of 1.35 m (4.5 ft) above the floor. It appears that these coefficients are not affected by space layout, load distribution in the space, supply airflow and temperature, number or type of diffusers, etc.

Early research found that many parameters play an important role in the performance of the TDV and UFAD systems. The occupancy patterns (Rock et al. 1995), system types (Akimoto et al. 1999), supply air temperature and thermal load (Di Tomaso et al. 2001, Xu et al. 2001, and Lin et al. 2005), and airflow rates and inlet locations (Xing et al. 2001 and Lin et al. 2005) were found to be such parameters. These conclusions are consistent with those found by Yuan et al. (1999) for TDV systems. Kobayashi and Chen (2003) revealed that diffuser types are crucial for ventilation performance. Sherman and Walker (2008) indicated that the location of sources of contaminants could lead to different contaminant distributions. However, although these studies are useful, they are fragmented. The systems studied by one researcher could be different from those studied by another. It is hard to make a direct comparison. Some of the studies were limited to the same system or the same space layout. Therefore, it is difficult to extend these results to the general design of TDV and UFAD systems for various kinds of spaces.

It is important to systematically study the impact of these parameters on the ventilation performance of the stratified air distribution systems. This is because more new offices, classrooms, restaurants, retail shops, workshops, and auditoriums at present are using these systems in the United States. Previous studies (Akimoto et al. 1999; Chen and Glicksman 2003) have implied that the required minimum amount of outdoor air for displacement ventilation in these buildings can be smaller than that for mixing ventilation due to the high effectiveness in ventilation. Bauman and Daly (2003) indicated the same for the UFAD ventilation systems. This has been acknowledged in ASHRAE Standard 62.1-2004 but with a fixed E_z value. Obtaining an accurate E_z value for stratified air distribution systems may, in many cases, justify reduction of the minimum outside air required to be supplied to a space under peak cooling load conditions without compromising indoor air quality. This, in turn, will reduce capacity and energy consumption of cooling systems, and further reduce greenhouse gas emissions. Therefore, it is desirable to develop an equation for calculating E_z value in design practice.

The equation to be developed should take into consideration the impact of space layouts that are likely to use the stratified air systems as well as the parameters that have been shown to be important according to the literature. The space layout should include offices, classrooms, restaurants, workshops, retail shops, and auditoriums. According to the literature review above, the parameters should be diffuser type, diffuser number, supply airflow rate, supply air temperature, heat source strengths, return outlet locations, and heating or cooling operating modes. This investigation reports our effort to create a database of air distribution effectiveness values for the TDV and UFAD systems and to develop a set of equations of air distribution effectiveness from the database through statistical analyses.

RESEARCH METHODS

Evaluation of ventilation system performance can use different parameters, such as ventilation effectiveness and air distribution effectiveness. Ventilation effectiveness, E_v, is a description of an air distribution system's ability to remove internally generated pollutants from a building, zone or space. In Chapter 27 of the ASHRAE Fundamental Handbook (ASHRAE 2005), ventilation effectiveness is defined as

$$E_v = \frac{C_e - C_s}{C_b - C_s} \tag{1}$$

where

E_v = the ventilation effectiveness
C_e = the contaminant concentration at the exhaust
C_s = the contaminant concentration at the supply
C_b = the contaminant concentration at the breathing zone

However, HVAC design engineers do not have knowledge of or control of actual pollutant sources within buildings, so the ventilation effectiveness may change dramatically if the pollutant source is moved slightly from one location to another. Therefore, this study used air distribution effectiveness. The definition of air distribution effectiveness is the same as the definition for ventilation effectiveness, but the contaminant source is assumed to be uniformly distributed in the entire indoor space.

$$E = \frac{C_e - C_s}{C - C_s} \tag{2}$$

where E is the air distribution effectiveness and C is the contaminant concentration at the location where the air distribution effectiveness is determined

By using the averaged C value in the breathing zone, the E becomes E_z, the averaged air distribution effectiveness in the breathing zone.

To develop a database of air distribution effectiveness for various space layouts and under different thermal and flow

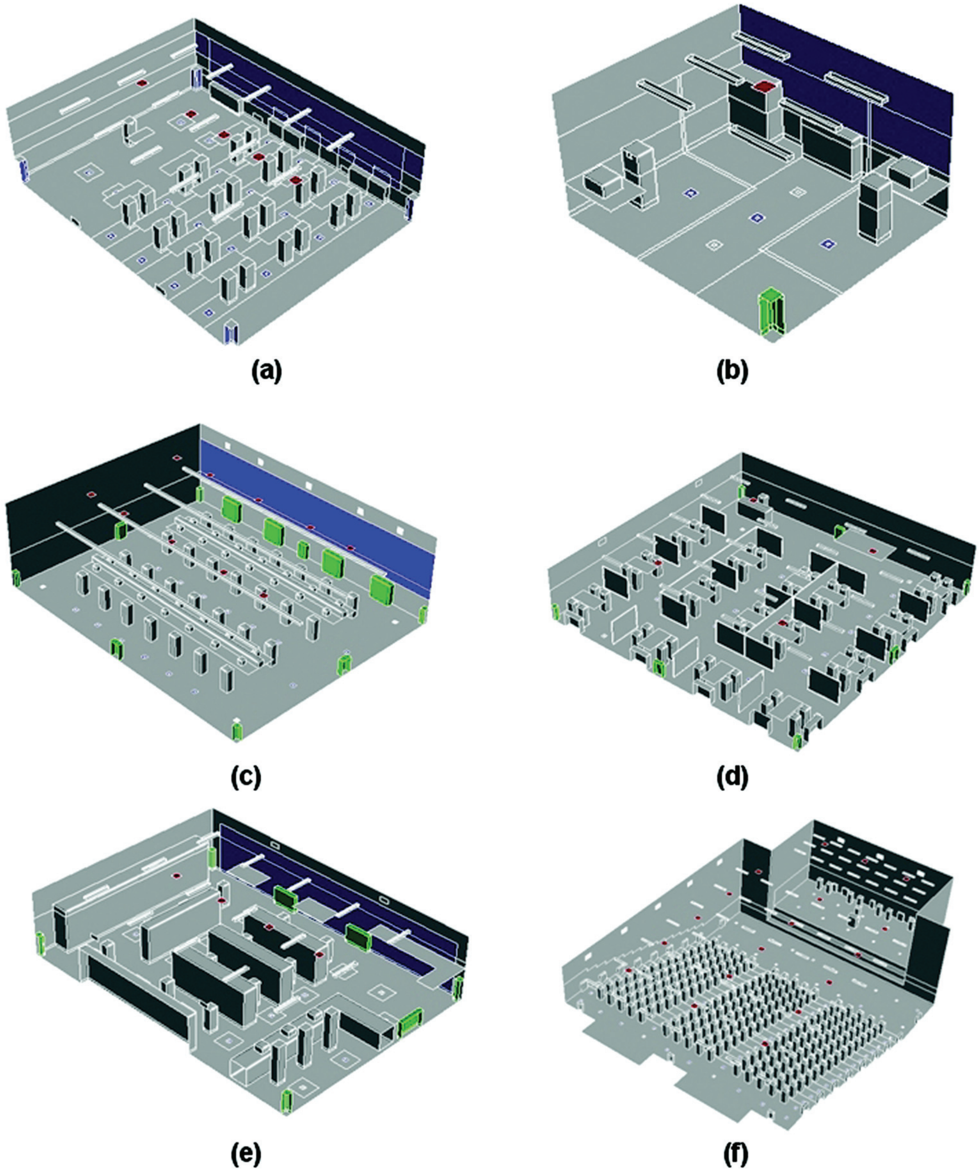

Figure 1 *Indoor space types used to create the air distribution effectiveness database: (a) classroom, (b) office, (c) workshop, (d) restaurant, (e) retail shop, and (f) auditorium.*

conditions, two approaches are possible: experimental measurements and computer simulations by CFD. Our companion paper (Lee et al. 2009) has discussed the pros and cons of the two approaches. Obviously, CFD is becoming more reliable, more user-friendly, less expensive, and faster compared with traditional mock-up tests. Thus, our investigation used the validated CFD program discussed in our companion paper to create the database.

The development of the database used six different types of spaces: classrooms (Fig. 1(a)), office spaces (Fig. 1(b)), workshops (Fig. 1(c)), restaurants (Fig. 1(d)), retail spaces (Fig. 1(e)), and auditoriums (Fig. 1(f)). For each type of space, there is a reference case, which is with the L-UFAD system under typical summer cooling conditions. Table 1 gives the size and the key thermal and flow boundary conditions for the reference cases.

For each type of indoor space, this investigation varied the thermal and flow parameters so that a total of 17 cases was studied as shown in Table 2. The standard conditions were those shown in Table 1. Variations were made on the parameters, which may have a major impact on the air distribution effectiveness according to the literature review. When changing a param-

Table 1. Size and Key Thermal and Flow Conditions Used in the Reference Case for Each Type of Space

Space Type	Dimension [m] ([ft])	Total Cooling Load		Supply Air Flow Rate [ACH]	Supply Air Temperature [°C] ([°F])	Diffuser Number [–]
		[w] ([Btu/h])	[w/m^2] ([Btu/h·ft^2])			
Classroom	11.7 × 9.0 × 3.3 (38.3 × 25.9 × 10.8)	5,840 (19,930)	51.5 (16.25)	6.0	17.2 (62.9)	16
Office	4.2 × 4.8 × 2.43 (13.7 × 15.7 × 7.9)	1,528 (5,215)	70.4 (22.20)	8.0	16.5 (61.7)	3
Workshop	15.0 × 12.0 × 4.5 (42.9 × 33.5 × 14.7)	13,705 (46,765)	70.7 (22.30)	4.5	17.0 (62.6)	22
Restaurant	15.0 × 15.0 × 3.0 (49.2 × 49.2 × 9.8)	15,484 (52,830)	64.0 (20.18)	6.0	17.0 (62.6)	22
Retail shop	10.0 × 14.0 × 3.0 (32.8 × 45.9 × 9.8)	8,553 (29,180)	56.9 (17.95)	5.5	17.0 (62.6)	16
Auditorium	30.0 × 20.0 × 8.0 (98.4 × 65.6 × 26.2)	36,204 (123,530)	54.7 (17.25)	3.0	17.5 (63.5)	74

Table 2. Parametric Study Matrix for Each Type of Indoor Space

Case Number	Diffuser Type	Diffuser Number	Supply Airflow or Air Temperature	Heat Sources	Return Location	Operating Mode
REF	L-UFAD	Standard	Standard	Standard	Ceiling	Cooling
Case 2	**TDV**	Standard	Standard	Standard	Ceiling	Cooling
Case 3	**H-UFAD**	Standard	Standard	Standard	Ceiling	Cooling
Case 4	L-UFAD	**Variation 1**	Standard	Standard	Ceiling	Cooling
Case 5	L-UFAD	**Variation 2**	Standard	Standard	Ceiling	Cooling
Case 6	L-UFAD	Standard	**Variation 1**	Standard	Ceiling	Cooling
Case 7	L-UFAD	Standard	**Variation 2**	Standard	Ceiling	Cooling
Case 8	L-UFAD	Standard	Standard	**Variation 1**	Ceiling	Cooling
Case 9	L-UFAD	Standard	Standard	**Variation 2**	Ceiling	Cooling
Case 10	**L-UFAD**	Standard	Standard	Standard	Ceiling	**Heating**
Case 11	**L-UFAD**	Standard	Standard	Standard	**Side wall**	**Heating**
Case 12	**TDV**	Standard	Standard	Standard	Ceiling	**Heating**
Case 13	**TDV**	Standard	Standard	Standard	**Side wall**	**Heating**
Case 14	**H-UFAD**	Standard	Standard	Standard	Ceiling	**Heating**
Case 15	**L-UFAD**	Standard	Standard	Standard	Ceiling	**2nd Heating**
Case 16	**TDV**	Standard	Standard	Standard	Ceiling	**2nd Heating**
Case 17	**H-UFAD**	Standard	Standard	Standard	Ceiling	**2nd Heating**

eter, it is usual to reduce by 15-40% or increase by 15-40%, depending on the space type and possible variation that one would find in reality. All the variations of the parameters are in bold letters in Table 2. The diffuser number for the reference case was the one recommended by diffuser manufacturers, which must have a flow rate within the range specified in the product catalog. Variation 1 means using fewer diffusers and variation 2, more diffusers than for the reference case. The supply airflow rate and supply air temperature must be varied at the same time since the cooling/heating load is the same. This implies that variation 1 used a lower airflow rate and a lower temperature and variation 2, a larger flow rate and higher temperature than those for the reference case for cooling. Obviously, the supply airflow rate and air temperature are interrelated, so they were counted as one parameter. When the heat source strength was increased or decreased, the supply airflow rate remained unchanged but the supply air temperature was adjusted accordingly to maintain the same room air temperature. The change of the return outlet location was straightforward, as shown in Table 2. The operating mode indicated the system could operate not only in cooling mode but also in heating mode. Under the heating mode, the supply air temperature was higher than that of the room air. Thus, the ventilation system may not always create a stratified condition as one could always find in cooling mode. The "secondary heating" refers to the secondary hydronic heating system used in a building perimeter to offset the heat loss in winter. Thus, the heating is offered by both the hydronic heating system and the ventilation systems. This arrangement is to ensure that only one parameter is changed at a time in order to isolate the influence of changing other parameters.

Please note from Table 2 that our efforts focused not only on the cooling mode, but also on the heating mode. The TDV systems are more often used only for cooling. The systems used perimeter heating systems for heating mode. However, our investigation here was to explore if the systems could be used directly for heating as well. Hence, the total cases calculated by CFD to create the database were 6 types of spaces × 17 cases/space for a total of 102 cases. The database obtained contains the following information:

- Geometrical information
- Thermal boundary conditions
- Flow boundary conditions
- Spatial data on air distribution effectiveness, air velocity, air temperature, age of air in ASCII format with a resolution of 0.1 m (4 in.)

Summary of major findings for each type of indoor space, such as the averaged air distribution effectiveness, air velocity, and air temperature gradient, etc.

The information can be found in the final report of this ASHRAE research project (Jiang et al. 2009).

With the database, one can obtain very detailed spatial information on the ventilation system performance, such as air distribution effectiveness. The information can then be used to develop equations for calculating the air distribution effectiveness in the breathing zone. These equations are useful for designers so that they do not need to access the database but can estimate the air distribution effectiveness of the space being designed.

The equations were developed by using statistical analysis. The development first assumed that a dataset has normal distribution, which can be expressed with a normal probability plot. The normal probability plot is a graphic representing a linear relationship between the dataset and the expected normal value (McClave and Sincich 2003).

The normal probability plot, as shown in Figure 2, is well-known as a very useful tool to distinguish major parameters from minor parameters. The horizontal axis of the plot is the standardized effects and the vertical one is the normal score for each parameter. The straight line in Figure 2 is a line of normal

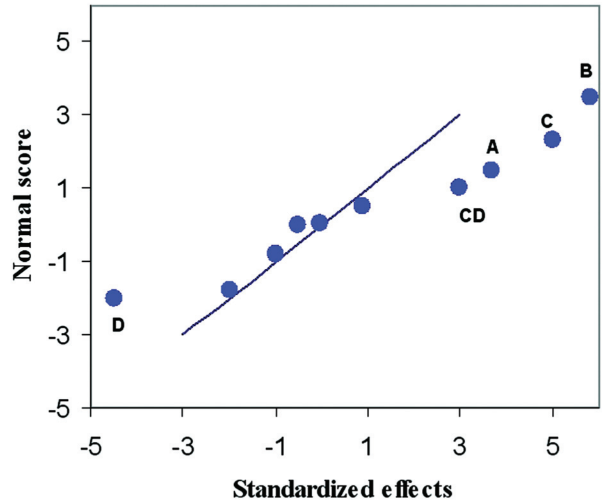

Figure 2 *An example of normal probability plot with variables of A, B, C, and D.*

distribution assuming that the dataset has normal distribution. The normal score is the expected standardized effects if the dataset has perfect normal distribution. In general, a distribution of the standardized effect is not exactly normal distribution. Thus, the location of the effect is off from the straight line according to its difference from the normal distribution. For example, if A, B, C, and D stand for diffuser type, supply air temperature, diffuser number, and thermal load, respectively, and Y is the air distribution effectiveness, the standardized effect is defined as

$$\text{The standardized effect} = \frac{\bar{Y}(+) - \bar{Y}(-)}{\sigma} \tag{3}$$

In Equation (3), $\bar{Y}(+)$ is the average of the air distribution effectiveness with a parameter greater than the mean value of the parameter. $\bar{Y}(-)$ is the average of the air distribution effectiveness for a parameter smaller than the mean value of the parameter. For instance, the supply air temperature, B, could be changed from 20°C (68°F) to 16°C (60.8°F) in design with a mean design temperature at 18°C (64.4°F). If the air distribution effectiveness increases with a decrease in the air supply temperature, the air distribution effectiveness for the supply air temperature higher than 18°C (64.4°F) is $Y(+)$ and that with B lower than 18°C (64.4°F) is $Y(-)$. Then

$$\bar{Y}(+) = \Sigma Y(+)/N_{higher} \tag{4}$$

$$\bar{Y}(-) = \Sigma Y(-)/N_{lower} \tag{5}$$

where N_{higher} and N_{lower} are the number of cases with the supply air temperature higher or lower than 18°C (64.4°F). The sign of Y means the increase or the decrease in the air distribution effectiveness as the supply air temperature is increased.

If the absolute standardized effect of a parameter is large, the parameter is important. Then the normal score would be either far from the center (0, 0) or far from the straight line. As shown in Figure 2, A, B, C, and D are important. Sometimes, the combination of the parameters, such as CD, as shown in the figure, can also be important. Figure 2 also shows that other parameters or their combinations not labeled are not important for the air distribution effectiveness, as they are either close to the center (0, 0) or close to the straight line. Then these important parameters should be used by regression to obtain an equation for determining the air distribution effectiveness as

$$\text{Air distribution effectiveness} = a_0 + a_1 A + a_2 B + a_3 C + a_4 D + a_5 CD. \tag{6}$$

RESEARCH RESULTS

This section will first discuss the impact of the thermal and flow conditions on the air distribution effectiveness of each space. Then the results from different space layouts will be discussed. Next, this study will use the most influential parameters to develop an equation calculating the air distribution effectiveness.

Due to limited space available, this paper only presents representative results selected from the investigation. Figure 3 shows the profiles of the air distribution effectiveness, E, for the classroom. The classroom was selected because the stratified air distribution systems with raised floor are commonly applied to classrooms and offices rather than to other spaces. Figure 3(a) compares the reference (L-UFAD) case with the TDV and H-UFAD cases. Very similar to what was reported in our companion paper (Lee et al. 2009), the TDV system had very high E in the lower part of the classroom. The H-UFAD system presented a lower E because of the strong mixing in the lower part of the classroom.

The impact of the number of diffusers on the E profiles is illustrated in Figure 3(b). The reference cases used 16 diffusers. When the number was reduced to 10, the supply air velocity increased significantly. As a result, the diffusers were no longer generating low height throws but high height throws and the E dropped significantly in the lower part of the room. On the other hand, by further increasing the diffuser number from the reference case, the supply air velocity would be reduced. But the reduction in the classroom had little influence on the E profile as the throws in the reference case were already low.

Figure 3(c) shows how the supply airflow rate and temperature cause different E distributions. Under the same cooling load, a lower supply airflow rate and temperature at the same time can lead to a higher E. However, the temperature data shows a very high risk of draft although it is not presented here due to space limitations in the paper. Again, by increasing the airflow rate and using a high air supply temperature, the E value becomes lower for the same reason explained in the previous paragraph.

Figure 3(d) describes the effect of heat load variation. In the study, the airflow rate to the classroom was unchanged. When the cooling load was decreased or increased, the supply air temperature was increased or decreased to maintain the same indoor air temperature. The figure shows that the E did not change much with the cooling load, although it decreased a little as the cooling load dropped. The effect of heat load variation was similar to that of changing the supply air temperature as shown in Figure 3(c). However, the magnitude of influence was much smaller.

The above results were for the first nine cases shown in Table 2. All of them were for cooling. Figure 4 further depicts the parametric studies for the ventilation systems under heating conditions. In order to find the differences between the cooling and heating modes, the reference case with the L-UFAD diffusers for cooling was again used in Figure 4(a). The heating case assumed a major heat loss through the building envelope so it needed heating. For the heating scenario, the supply airflow from the diffusers had an air temperature higher than that in the classroom. When the supply air temperature is high, the throws from the diffusers can reach a much higher height due to thermal buoyancy. Thus, it is not surprising to notice that the UFAD system turned into a well-mixing system with an E value close to 1.0.

Figure 4(b) explains how the location of the return outlet could affect the air distribution effectiveness. The figure compares two ventilation systems, TDV and L-UFAD. The TDV system had rather low air distribution effectiveness because the warm air with very low air velocity from TDV diffusers rises quickly to the ceiling before it mixes with room air. Generally, it is not recommended to use the TDV systems for heating mode. This study shows that the TDV system may be used for heating with careful design. The results indicate that the effect of the outlet location was negligible. This is because the airflow in the space was well stratified so that the indoor air quality in a horizontal direction in the upper part of the room did not change much.

Figure 4(c) shows the impact of the secondary heating system on the air distribution effectiveness. The performance of the L-UFAD system did not vary much with or without the secondary heating system since the space had many diffusers installed around the perimeter of the classroom. The warm jets from the diffusers could prevent the downward flow from the cold enclosure surfaces, which was similar to the function of the thermal plumes from the secondary heating system. The TDV system with the secondary heating performed poorly compared to the L-UFAD system.

The results presented in Figures 3 and 4 were for the classroom. For other spaces, the trend looked the same although the magnitude of the differences varied.

Figure 5 shows the averaged air distribution effectiveness in the breathing zone for all six spaces studied with standard conditions of supply airflow rate, diffuser number, supply air temperature, heat sources, return location in Table 2. Clearly, the TDV and L-UFAD systems had a higher E_z value than did

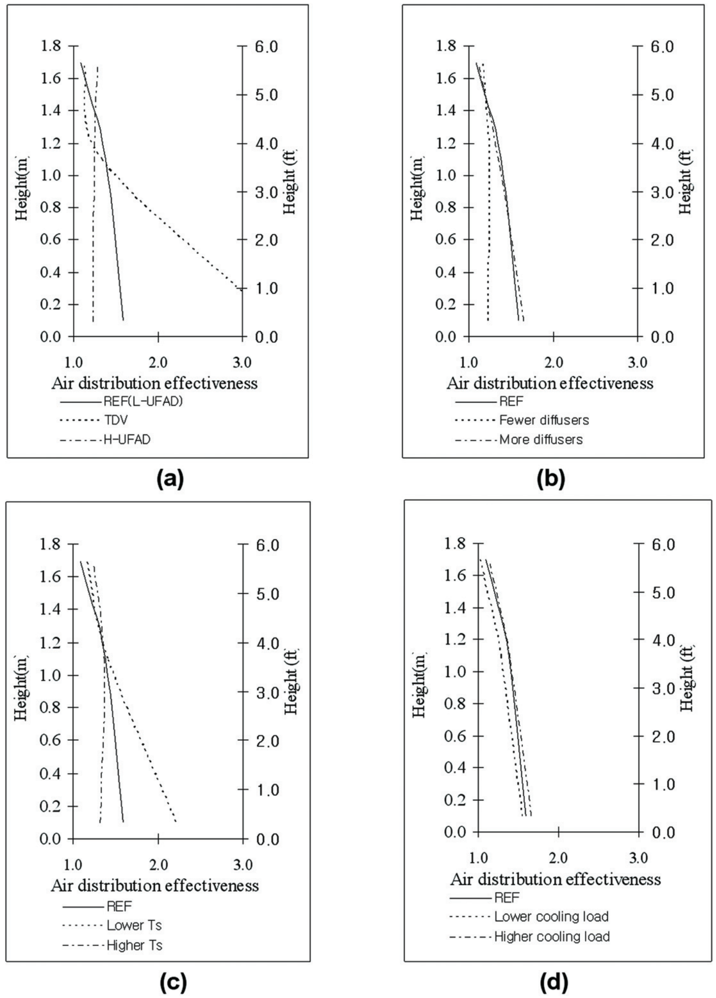

Figure 3 *Effects of some key parameters on vertical gradient of air distribution effectiveness for the classroom in cooling mode: (a) diffuser type, (b) diffuser number, (c) supply temperature, and (d) cooling load.*

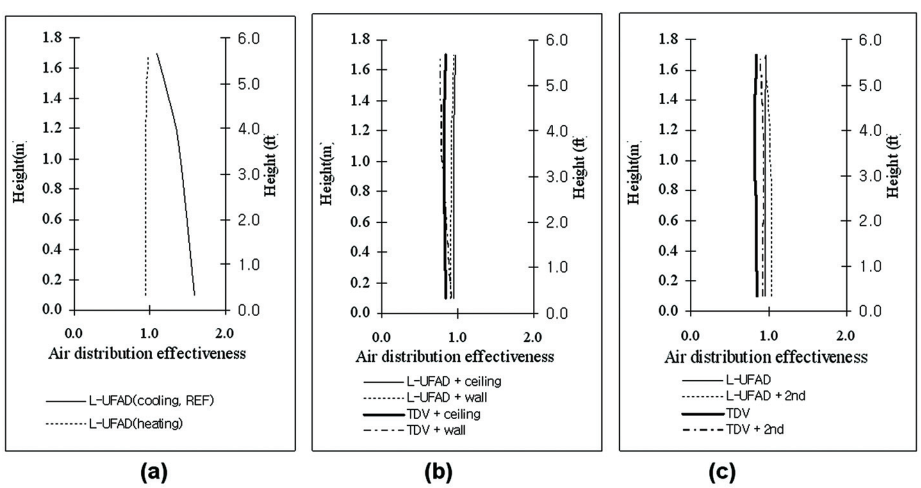

Figure 4 *Effects of some key parameters on vertical gradient of air distribution effectiveness for the classroom in heating mode: (a) cooling vs. heating, (b) return location, and (c) secondary heating system.*

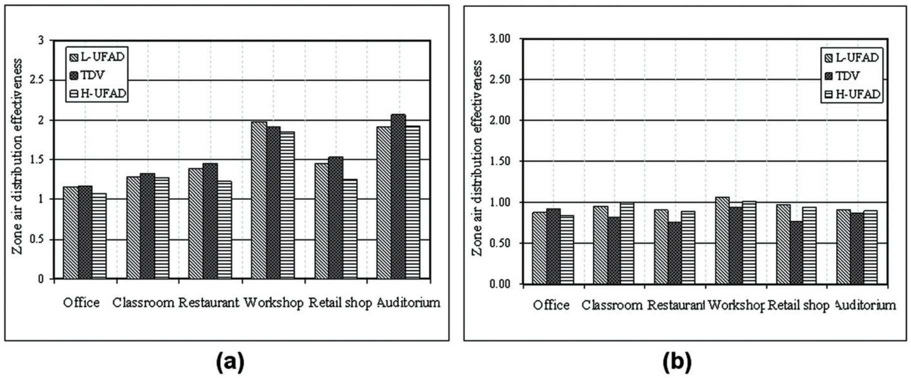

Figure 5 *Averaged air distribution effectiveness in breathing zone depending on diffuser type: (a) summer cooling and (b) winter heating.*

the H-UFAD system for cooling. The TDV system was a little better than the L-UFAD system. This is because the TDV system usually has a very high E_z in the lower part of the room. The higher the ceilings, the larger the air distribution effectiveness. This is well known for stratified ventilation systems. The results indicate that the averaged air distribution effectiveness in the breathing zone for cooling ranged from 1.15 to 1.98 for the L-UFAD system, 1.16 to 2.06 for the TDV system, and 1.06 to 1.92 for the H-UFAD system.

Please note that the H-UFAD system is not a well-mixing system for cooling unless the ceiling is very low such as in offices. The system can have an E_z value higher than 1.0. Particularly when the ceiling height is high, such as in auditoriums and workshops, the H-UFAD system does not mix the air from the lower part of a room with the air from the upper part.

When the TDV systems were used for heating, the supply air temperature from the diffusers was higher than that of the room air, and the air velocity from the diffusers was very low. Due to the thermal buoyancy, the warm and fresh air could travel quickly upwards to the upper part of the room. Thus, the TDV systems in Figure 5(b) showed low E_z in most indoor spaces except in the office, which had a space with only one TDV diffuser in a corner of the room. In the office, the downward cold air from the cold perimeter walls, which helps in mixing contaminated air with the warm and fresh air, is much stronger than that of other indoor spaces. On the other hand, the UFAD systems provide a strong mixing in the occupied zone even for heating mode. This is why the TDV systems generally have low E_z value compared to that from the UFAD systems for heating mode. Figure 5(b) has E_z in the breathing zone ranging from 0.75 to 1.05. When the E_z is smaller than 1.0, this implies that some fresh air was not mixed with the room air and went directly from an inlet to an outlet for the ventilation systems under the heating mode.

The results discussed were either for a single space (classroom) (Figures 3 and 4) or were under standard conditions for all spaces with a single value (Figure 5). In order to further identify the parameters that have a major influence on the air distribution effectiveness, it is necessary to include the data for all six kinds of spaces in the analysis. If the presentation were in the form shown in Figures 3 and 4, the figures would not be clear with so much data. Thus, a different form was used to analyze the entire database for the six kinds of spaces under the cooling operation mode, as depicted in Figure 6. The figure uses the averaged air distribution effectiveness in the breathing zone as the evaluation criterion.

Figure 6(a) confirms that the number of diffusers had a large impact on the zone air distribution effectiveness. For the reference cases, the averaged E values in the breathing zone varied from 1.15 to 1.98. With the same supply air temperature and total flow rate, the averaged E values increased with the diffuser number. With more diffusers, the discharging air velocity became smaller. Thus, the corresponding throw height was lower and the E was higher.

The lower the supply air temperature, the higher the E values in the breathing zone, as shown in Figure 6(b). This is because the lower supply air temperature helped to create a stronger stratified air distribution in the rooms. When the stratified flow became more stable, the clean air would stay more easily in the breathing zone. The impact of the supply air temperature on the air distribution effectiveness looked similar to that of the diffuser number.

Figure 6(c) indicates that the cooling load was also another important parameter influencing the averaged E value in the breathing zone. The cases with a higher cooling load had a higher averaged E value. This is because the higher cooling load required a lower supply air temperature since the total flow rate was the same. The lower supply air temperature created a stronger stratified flow due to thermal buoyancy.

Although the air change per hour (ACH) in an indoor space was not a major parameter in the original study, it in fact played an important role in determining the averaged E_z value. Figure 6(d) illustrates the influence of the air change rate on

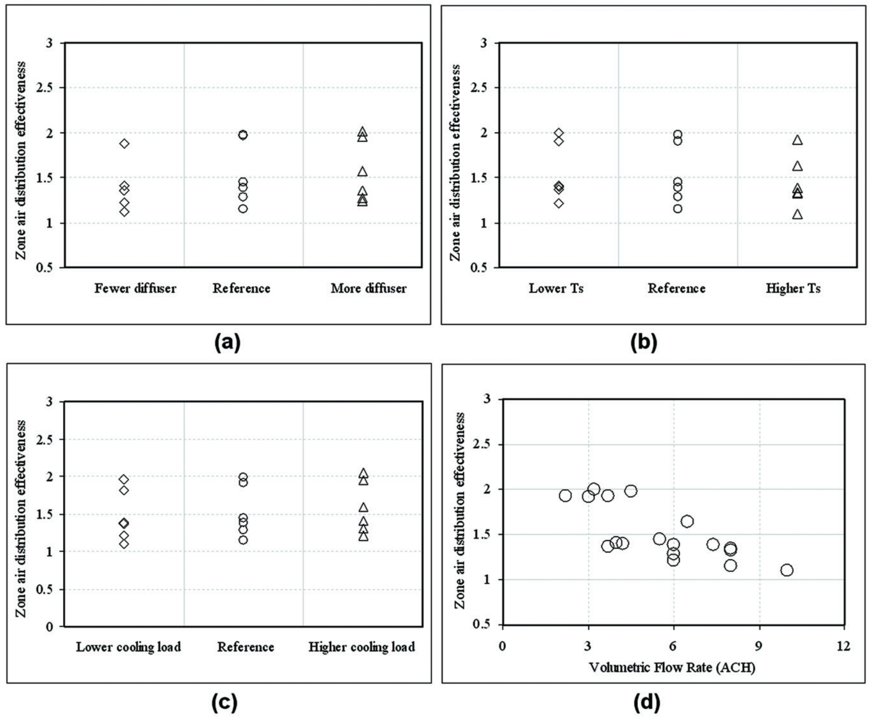

Figure 6 *The impact of different parameters on the averaged air distribution effectiveness in the breathing zone with the L-UFAD diffusers under cooling conditions: (a) diffuser number, (b) supply air temperature, (c) cooling load, and (d) total airflow rate.*

the E_z value. The air distribution effectiveness decreased as the air change rate increased. The figure is for all six spaces. The spaces with a high ceiling had a low air change rate. Figure 5(a) shows that high ceiling spaces had high E_z values. The two figures illustrate the same phenomenon. Therefore, the ACH is an alternative parameter for ceiling height.

Figure 7 plots the results presented in Figure 5(b) in a different fashion to further analyze the effect of the secondary heating systems on the E_z value under the heating operation mode. Again, the results confirm that the influence of the secondary heating system was negligible.

Table 3 summarizes the effect of different parameters on the air distribution effectiveness. The performance of the TDV system was similar to that of the L-UFAD system. The H-UFAD system, which had high height throws, had lower E values in the occupied zone compared to those of the TDV and L-UFAD systems for cooling mode. The more diffusers used, the higher the E would be. With a higher supply air temperature, the E value became lower. When the cooling load was increased, the E value would increase slightly. When the stratified systems were used for heating, the E values dropped significantly. The Secondary heating systems did not seem to have a major impact on the air distribution effectiveness. When the total flow rate in terms of ACH increased, the E value would decrease accordingly.

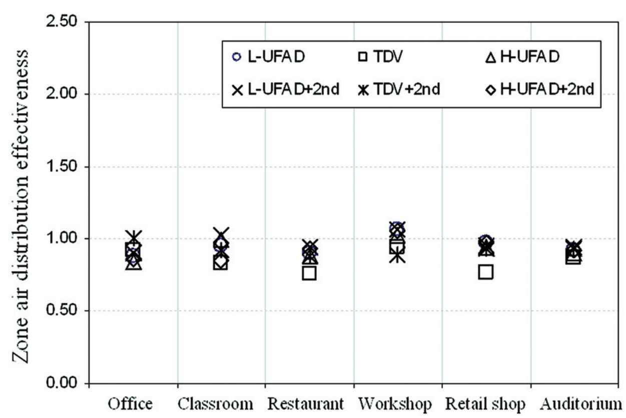

Figure 7 *The impact of the secondary heating system on the averaged air distribution effectiveness in the breathing zone in heating mode.*

Table 4 lists the zone air distribution effectiveness found in this study for cooling mode. Currently, Table 6-2 of ANSI/ASHRAE Standard 62.1-2004 (ASHRAE 2004) provides a single value of zone air distribution effectiveness of 1.2 for cooling mode for the stratified systems. However, the database from the parametric study indicates that the E_z could vary according to indoor space type and system type as indicated in

Table 3. Summary of the Effect of Different Parameters on the Air Distribution Effectiveness

Parameters	*E* Changes
Change L-UFAD to TDV	No difference
Higher throw from diffuser	Decreased strongly
More diffuser number	Increased strongly
Higher supply air temperature	Decreased strongly
Higher cooling load	Increased slightly
Changed cooling to heating	Decreased strongly
Used secondary heating	No difference
Higher total flow rate	Decreased strongly

Table 4. Zone Air Distribution Effectiveness for Different Indoor Spaces with UFAD and TDV Systems

	Zone Air Distribution Effectiveness (E_z)		
	L-UFAD Systems	H-UFAD Systems	TDV Systems
Offices	1.05 ~ 1.25	1.05	1.15
Classrooms	1.20 ~ 1.35	1.25	1.30
Restaurants	1.20 ~ 1.40	1.20	1.45
Workshops	1.60 ~ 2.00	1.85	1.90
Retail shops	1.25 ~ 1.60	1.25	1.50
Auditoriums	1.85 ~ 2.05	1.90	2.05

the table. The E_z for the TDV systems is not always equal to 1.2 as currently suggested in Standard 62. The E_z could be as high as 2.05 as in the cases shown in Table 4. The E_z for the L-UFAD systems could range from 1.05 to 2.0. The H-UFAD systems have a little lower E_z than the L-UFAD and TDV systems. We strongly recommend not using single E_z for the stratified air distribution systems. For simplicity, Table 4 should be used. In addition, designers are recommended to use the sets of equations that are introduced in the next section.

Development Equations to Calculate Air Distribution Effectiveness

Although the above results show the trends in how the *E* varied with different parameters, it is difficult to use the database quantitatively. This is because the parameters were dependent on each other. In other words, the parameters studied may have counter effects on each other so that it is not easy to estimate the *E* value for a new case. Thus, it is essential to establish a relationship that can quantitatively estimate the *E* value change with different parameters.

This investigation has studied the influence of several parameters on air distribution effectiveness, and a database of air distribution effectiveness has been established. The statistical analysis method can be used to find the influence of major parameters and the combinations of these parameters on air distribution effectiveness. This method can then develop a set of equations with which a designer can determine the air distribution effectiveness of his/her design.

Figure 8 shows a normal probability plot and a Pareto chart for air distribution effectiveness, where A is diffuser type, B is total flow rate (ACH), C is flow rate per diffuser (m^3/h or ft^3/min), D is supply air temperature (°C or °F), F is cooling load (W/m^2 or Btu/h·ft^2), and G is diffuser density (#/m^2 or #/ft^2). These parameters were studied and discussed in the previous section. The higher the standardized effect for a parameter, the stronger this would influence on the *E* value. The figure also includes the combined effect on the air distribution effectiveness from multiple parameters because the parameters might be inter-related. For example, CFG is the combined changes from flow rate per diffuser, cooling load, and diffuser density. Figure 8(a) indicates that one single parameter, B, and some combinations, CFG, BC, AF, BCD, and BD, are located far from the center (0, 0) and the straight line which expresses normal distribution. Since the variable, CFG, is far from the center and well off the line, it has the strongest influence on the *E* value. And since the variable AF is closest to the center and the straight line, its impact on the *E* would be the smallest among the six variables. Thus, these variables could be the dominate ones having a major impact on the air distribution effectiveness. Of course, one can include more variables, but their contribution to the changes in the *E* value would be minimal. The points without labels shown in Figure 8(a) are for the parameters or combined parameters, which had minimal impact on the *E* value. In order to develop an equation for *E* that can be easily used in design and include the most important variables, this study limited the number of variables to six.

By using the statistical method, the database, and the six variables identified, the following six equations have been developed for calculating the air distribution effectiveness:

$$E_z = 1.9 + 0.000257CFG + 0.0105BD + 0.000008BCD - 0.358B - 0.00025BC + 0.00591AF \text{ (SI)} \quad (7a)$$

$$E_z = 1.9 + 0.01489244CFG + 0.0058333BD + 0.00000755BCD - 0.5446667B - 0.00066639BC + 0.01864412AF \text{ (I-P)} \quad (7b)$$

$$E_{sitting} = 2.03 + 0.000183CFG + 0.00774BD + 0.000031BCD - 0.296B - 0.000682BC + 0.00588AF \text{ (SI)} \quad (8a)$$

$$E_{sitting} = 2.03 + 0.01060435CFG + 0.0043BD + 0.00002926BCD - 0.4336B - 0.00209507BC + 0.01854948\,AF \text{ (I-P)} \quad (8b)$$

$$E_{standing} = 1.66 + 0.000169CFG + 0.011BD + 0.000012BCD - 0.315B - 0.00033BC + 0.00476AF \text{ (SI)} \quad (9a)$$

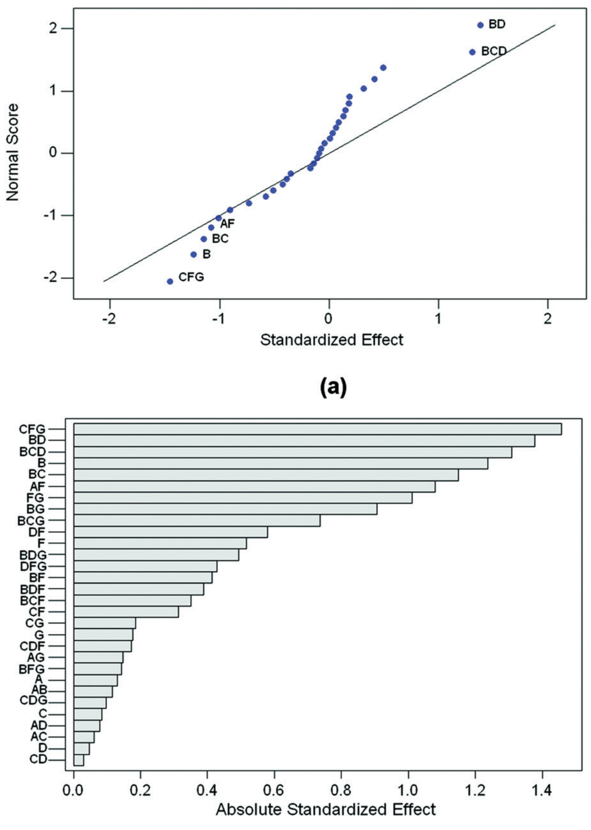

Figure 8 *(a) Normal probability plot and (b) Pareto chart.*

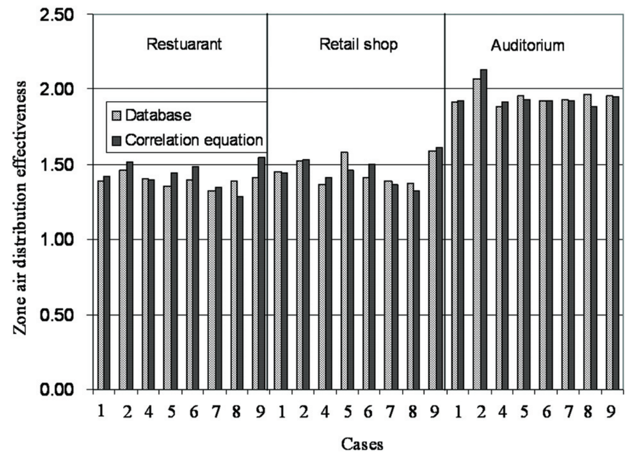

Figure 9 *Comparison of the zone air distribution effectiveness in the database and that calculated with the correlation equation.*

$$E_{standing} = 1.66 + 0.009793\text{CFG} + 0.006111\text{BD} + 0.000011\text{BCD} - 0.510556\text{ B} - 0.000923\text{ BC} + 0.015016\text{ AF (I-P)} \quad (9b)$$

where A is diffuser type (1 = L-UFAD diffuser, 2 = TDV diffuse), B is total flow rate (ACH), C is flow rate per diffuser (m^3/h or ft^3/min), D is supply air temperature (°C or °F), F is cooling load (W/m^2 or Btu/h·ft^2), and G is diffuser density (#/m^2 or #/ft^2).

Equation (7a) and (7b) are for the E_z, Equation (8a) and (8b) are for the E at breathing level of a sitting person, and Equation (9a) and (9b) are for the E at breathing level of a standing person, respectively. For example, if a space employs an L-UFAD system with an air change rate of 6 ACH, flow rate per diffuser of 160 m^3/h, supply temperature of 17.2°C, cooling load of 51.5 W/m^2, diffuser density of 0.1 #/m^2, then, A is 1.0, B is 6 ACH, C is 160 m^3/h, D is 17.2°C, F is 51.5 W/m^2, and G is 0.1 #/m^2. Thus, C·F·G is 824.0, B·D is 103.2, B·C·D is 16512.0, B is 6.0, B·C is 960.0, and A·F is 51.50 in Equation (7a), (8a), and (9a) in the SI unit. By substituting the variables for Equation (7a), (8a), and (9a) the predicted E values are 1.25 in breathing zone, 1.36 at breathing level of a sitting person, and 1.17 at breathing level of a standing person.

To verify whether the equations can estimate the air distribution effectiveness, they were tested for the entire database of the six indoor spaces investigated. For clarity, Figure 9 shows the E_z calculated by Equation (7a) or (7b) for the three selected spaces. The results for other spaces are similar although they are not presented due to the limited space available in this paper. Obviously, the equations can predict very well the E_z values of the TDV and L-UFAD systems. In most cases, the difference between the database and the prediction is less than 10%. This difference is certainly acceptable in ventilation system design, so the equations are validated.

This investigation has thus developed equations for calculating the averaged E in the breathing zone, the E at breathing level for sitting occupants, and the E at breathing level for standing occupants. Using the equations, a designer can predict the E for a room with the TDV or the L-UFAD system. The room does not have to be the same as in the database. Of course, the equations are valid within the range of the database.

CONCLUSION

This study investigated the influence of different design parameters on air distribution effectiveness. The parameters studied were space type, diffuser number, supply air temperature, cooling load, return location, and secondary heating system under both cooling and heating conditions. The results from the parametric study show that the performance of the TDV system was the same as the performance for the L-UFAD system. The throw height from a diffuser, diffuser number, supply air temperature, total flow rate, cooling load, and heating or cooling mode can have a major impact on the air distribution effectiveness. The influence from the secondary heating system on the air distribution effectiveness was minimal.

The parameter study had 102 cases that formed a database. The new recommended value of E_z with the L-UFAD systems is 1.05 ~ 1.35 for offices and classrooms, 1.2 ~ 1.6 for restaurants and retail shops, and 1.6 ~ 2.0 for workshops and auditoriums. The E_z for the TDV system is close to that for the L-UFAD system. However, the E_z for H-UFAD is slightly higher than 1.0 for classrooms, offices, restaurants, and retail shops, which have low ceilings and much higher than 1.0 for workshops and auditoriums, which have high ceilings. A statistical analysis was used to develop a set of equations to calculate different levels of air distribution effectiveness using the database. The equation development selected six parameters that are most important to air distribution effectiveness. By comparing the air distribution effectiveness calculated with that of the corresponding case in the database, the difference was acceptable for designing ventilation with the TDV and L-UFAD systems.

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