Diffuser selection method considering both thermal comfort and ventilation effectiveness in mixing ventilation

混合換気方式における快適性と換気効率を考慮した吹出口選定手法に関する研究

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Chapter 1

Introduction

1.1 Objectives of the research

Providing comfort and a healthy environment for occupants with minimum use of energy is the ultimate purpose of heating, ventilating, and air conditioning (HVAC) systems. Building ventilation directly affects the indoor air quality, and influences the health and productivity of the building's occupants. Among various types of ventilation, the best-known and most frequently used ventilation method is mixing ventilation. Mixing ventilation aims to dilute polluted and warmed/cooled room air with a cleaner and cooler/warmer supply of air to lower the contaminant concentration and provide an optimal temperature and indoor air quality. Mixing ventilation has been applied to a large variety of room types using different air diffusers and exhaust types with various HVAC systems; such as a variable air flow volume (VAV) system with constant supply temperature and constant air volume system with variable supply air temperature. One of the major challenges of mixing ventilation is its low effectiveness with regards to the exchange of air, which may occur owing to temperature stratification with all-air heating because the same mixing ventilation diffusers often provide both space cooling and heating.

In terms of thermal comfort, the impact of draft and temperature uniformity are measured based on the air distribution performance index (ADPI) (Miller et al. 1971). It is a widely accepted index that shows the performance of diffusers and is used for diffuser selection. The ADPI is defined as the percentage of an occupied zone falling into the acceptable velocity and temperature range. The region is determined by the local effective draft temperature (EDT), which combines the air temperature difference of the local and spacious average temperatures, and air speed. ADPI incorporates with the throw of selected terminal velocity and the characteristic length that describes a room geometry, and it provides design variables for selecting diffusers. The ADPI method is only valid for overhead air distribution systems under cooling operation (Krati et al 2008). However, in practice, the same mixing ventilation diffusers often provide space heating and cooling (Platt et al. 2010, Vakiloraya et al. 2014, Liu et al. 2015). This causes many issues during the heating period. Liu et al. (2016) recently expanded the ADPI concept in the heating mode and obtained ADPI values with recommended design criteria for various types of diffusers. They also updated the ADPI data with diffusers commonly available on the market nowadays under the cooling mode.

Current practice related to the air distribution design and diffuser selection relies on only the ADPI, which considers only temperature uniformity and draft. The impact of stratification and low ventilation effectiveness on all-air heating systems is taken into account by a correction factor in the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 62.1 (2010). This does not always result in proper diffuser selection. For example, when the throw is too short, the jet may detach from the ceiling, increasing the draft risk under the cooling regime. Moreover, a short throw length may cause inadequate mixing that results in a high temperature gradient and low air quality under the heating regime. It is necessary to have some momentum flow to obtain adequate mixing in the occupied zone. However, a very large supply jet momentum may generate a draft when the flow rate in the occupied zone is above a certain level. The temperature difference between the supply and return jets should also be restricted as a high-temperature difference may cause a draft and a very large temperature gradient may result in inefficient energy use. Few studies focused on ventilation effectiveness with mixing diffusers used for space heating have been conducted, although all-air-heating ventilation is widely used. A comprehensive design process for diffuser selection and positioning that considers both thermal comfort and ventilation effectiveness at the same time is needed.

In contrast, the ASHRAE Standard 62.1 (2010) specifies the minimum ventilation rate in different types of buildings. This required ventilation rate is adjusted by taking into account the impact of ventilation effectiveness. The ASHRAE Standard 129 (2002) specifies the standard procedure for measuring air change effectiveness and how the minimum ventilation rate in Standard 62.1 (2010) can be modified by air change effectiveness. However, this standard may not be practical for field measurements as it demands extensive measuring equipment or repetitive measurement. Specifically, the standard requires to measure air change effectiveness 25% of workstation or at least 10 locations in the test space. This results in low application of the ASHRAE standard 129, and the air change effectiveness is rarely measured in the field. Other standards, such as ISO 16000-8 and the standard developed by the Society of Heating, Air-Conditioning, and Sanitary Engineers of Japan also address such procedures. As the ventilation effectiveness may have a significant effect on both indoor air quality and building energy performance, it is important to know how to measure it both properly and effectively.

Thus, the three primary objectives of this study are summarized as follows:

(1) Define the operation range of commonly used diffusers

Define the operation range for selecting diffusers with an acceptable ADPI, air change effectiveness, and temperature effectiveness. Provide selection data for air distribution system design for the most common types of celling diffusers—linear slot diffuser, round ceiling diffuser, louvered face diffuser with no lip and perforated diffuser directional pattern (4 way)—in both cooling and heating regimes.

(2) Evaluate a method for improving ventilation effectiveness in the heating mode

Provide simple strategies to improve ventilation effectiveness in the heating regime while maintaining acceptable ADPI values. Evaluate the improvements and operation ranges of diffuser deflector adjustments, room-supply air temperature differences, and exhaust locations.

(3) Assess a procedure for evaluating air change effectiveness

Assess a procedure for evaluating the air change effectiveness of mixing ventilation by analyzing variances in local air change effectiveness. Examine the correlation of temperature effectiveness and air change effectiveness to suggest a simplified strategy for evaluating air change effectiveness that may be applied in actual buildings.

The author performed experimental measurements in a full-scale test room. Carbon dioxide (CO_2) tracer gas decay tests were conducted to measure the age of the air at multiple locations in the test room simultaneously with various types of ceiling diffusers/pattern adjustments at different airflow rates and internal loads. The presented study expands recent research on ADPI with various types of diffusers by Liu et al. (2016). The author also performed ADPI evaluations under conditions not considered in Liu's study.

1.2 Background

This section explains the theoretical and conceptual background of this research. The first section introduces ventilation types commonly utilized in the industry. The next sections explain indices used to evaluate ventilation. Among several indices, $T_{0.25}/L$ and ADPI, air change effectiveness, and temperature effectiveness are utilized to evaluate the performance of the diffusers with regards to thermal comfort, ventilation effectiveness, and heat removal efficiency, respectively. The chosen indices have been studied extensively by many researchers and are widely used in the industry. Detail explanations of $T_{0.25}/L$ and ADPI, air change effectiveness, and temperature effectiveness are provided in sections 1.2.2.3, 1.2.3 and 1.2.4, respectively. In addition, current practices of diffuser selection are introduced.

1.2.1 Ventilation methods

According to the Federation of European Heating, Ventilation and Air Conditioning Associations (REHVA) Guidebook No.2 (Muller et al. 2013), ventilation methods are categorized into three groups based on their air distribution patterns. The first ventilation method introduced in this section is mixing ventilation. Mixing ventilation refers to an air distribution pattern, rather than a ventilation system. It can also be called an air distribution pattern with a mixing effect or mixing air distribution. The purpose of mixing ventilation is to dilute polluted and warm/cool room air with cleaner and cooler/warmer supply air.

The second ventilation classification in REHVA is displacement ventilation. It can be called an air distribution pattern with a displacement effect or displacement air distribution. This ventilation pattern aims to replace but not mix polluted room air with clean air. The last method is unidirectional (piston) ventilation. Piston flow will take place when air distribution is controlled by the momentum flow. A very high flow rate is required to obtain such flow, and the air change rate can typically be 50 to 100 h^{-1}.

On the other hand, Cao et al. (2013) conducted a comprehensive review of scientific literature on air distribution systems and classified different ventilation systems according to specific requirements and assessment procedures. The ventilation methods were categorized into eight groups, i.e., mixing ventilation, displacement ventilation, personalized ventilation, hybrid air distribution, stratum ventilation, protected occupied zone ventilation, local exhaust ventilation and piston ventilation. In addition, five indices were introduced to assess the ventilation performance, including ventilation effectiveness, in terms of air exchange, pollutant removal, heat removal, exposure and air distribution. The study found that the assessment of ventilation effectiveness or efficiency should depend on the purpose of the ventilation system, and provided the basic framework regarding application of airflow distribution.

1.2.1.1 Mixing ventilation

Mixing ventilation is the most known and used ventilation method among the various types of strategies (Cao et al. 2013). In the design of an air distribution system, various types of supply air diffusers and return air inlets (exhaust) can be applied to create mixing in different space types (Muller et al. 2013) and can be applied to a large variety of room types. According to the ASHRAE Handbook-Fundamentals (2009), supply air diffusers (also called outlets) are categorized into five groups. Table 1-1 shows the classification of outlet types and examples of diffusers. Group A outlets are defined as outlets that are mounted in or near the ceiling that discharge air horizontally, and Group

E outlets are outlets that are mounted in or near the ceiling that project primary air vertically to the outlets utilized in mixing ventilation.

Table 1-1 Classification of outlet types and example of diffusers

	Definition	Outlet Examples
Group A	Outlets mounted in or near the ceiling that discharge air horizontally	high side wall grilles, side wall diffusers, ceiling diffusers, and linear ceiling diffusers
Group B	Outlets mounted in or near the floor that discharge air vertically in a nonspreading jet	floor registers, baseboard units, low sidewall units, and linear-type grilles in the floor or window sills
Group C	Outlets mounted in or near the floor that discharge air vertically in a spreading jet	floor diffusers, sidewall diffusers, linear type diffusers and other outlets installed in the floor or window
Group D	Outlets mounted in or near the floor that discharge air horizontally	baseboard and low sidewall registers
Group E	Outlets mounted in or near the ceiling that project primary air vertically	ceiling diffusers, linear grilles, sidewall diffusers and grilles

One of the most comprehensive mixing ventilation guides for mixing ventilation was published by REHVA; REHVA Guidebook No.19 “Guide on mixing air distribution design” (Muller et al. 2013). The guidebook gives an overview of the theory of mixing ventilation, design methods, and several case studies. Greater detail will be provided in the following sections.

1.2.1.2 Displacement ventilation

Displacement ventilation (DV) is one of the concepts utilized in the supply of conditioned air to and ventilation of buildings. Floor terminals or other lower mounted diffusers introduce air into a room at a low velocity. Figure 1-1 shows the concept of DV (Chen et al. 2003). As shown in the figure, as the air is heated by heat sources in the room, it rises, passes through the occupied zone, and is then exhausted at a high level. Cho et al. (2005) comprehensively reviewed research on displacement ventilation and categorized DV into three groups: 1) traditional displacement ventilation, 2) displacement ventilation with a chilled ceiling panel, and 3) displacement ventilation with a raised floor. The review was focused on temperature airflow and contaminant distribution, indoor air quality and comfort, and energy and cost. According to their review, the energy consumption of DV is similar to that of a mixing system, though it can sometimes be smaller or larger, depending on the control strategies. On the other hand, DV can achieve higher IAQ and comfort. According to ASHRAE Standard 62 (2016), the air change effectiveness of DV under the cooling mode is 1.2. However, the air change effectiveness under the heating mode is 0.7.

REHVA published the first version of its DV guide, which introduces the design guidance including diffuser selection with case studies. (Skistad et al. 2002). Recently, a revised guidebook, which simplifies and improves the practical design procedure, was published (Kosonen et al. 2017). In general, DV is suitable in tall rooms and where the supply air is cooler than ambient air. Furthermore, DV is not suitable for heating as warm supply air will rise due to buoyancy and will be extracted when it reaches the ceiling.

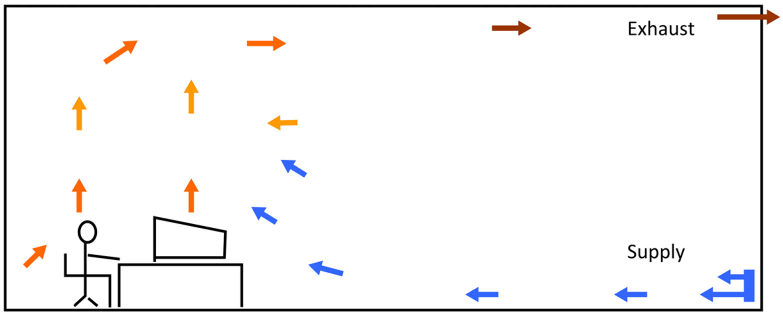


Figure 1-1 Sketch showing displacement ventilation. (Chen et al. 2003) (Prepared by author referring the reference)

1.2.2 Ventilation effectiveness

Four different standards/guidebooks on ventilation effectiveness are reviewed in this section. The publications include ASHRAE Standard 129 (2002) "Measuring Air-Change Effectiveness," REHVA Guidebook No.2 (Muller et al. 2013) "Ventilation Effectiveness," International Organization for Standardization (ISO) 16000-8 Part 8 (2007) "Determination of local mean age of air in buildings for characterizing ventilation conditions," the Society of Heating, Air-Conditioning and Sanitary Engineers of Japan (SHASE) Standard 115 (2010) "Field Measurement Methods for Ventilation Effectiveness in Rooms," and SHASE Standard 116 (2011) "Ventilation Rate Measurement of a Single Room Using Tracer Gas Technique."

All organization introduce the age of air concept based on tracer gas measurements. The age of air is measured by measuring the tracer gas concentration, whereby the tracer gas labels the indoor air with an inert or nonreactive gas (Dietz et al. 1986, Fisk et al. 1989, Fortmann et al. 1990; Harrje et al. 1990, Lagus and Persily 1985, Persily and Axley 1990, Sherman 1989, 1990, Sherman et al. 1980). Figure 1-2 shows the age of air concept. The age of air is defined as the average time that has elapsed since molecules of air in a given volume of air entered the building from the outside. In other words, according to the ISO, it describes the length of time the air at a specific location has on average spent within building.

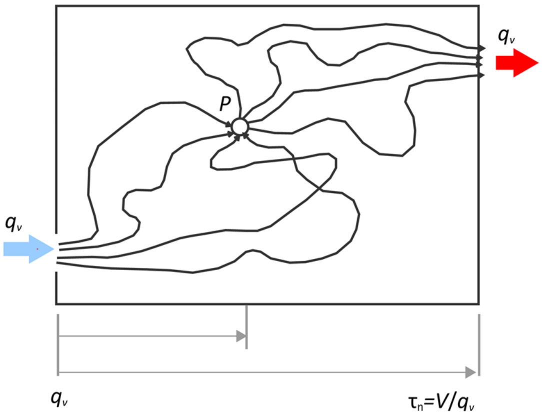


Figure 1-2 Definition of the age of air. (Muller et al. 2013)

(Prepared by author referring the reference)

The air at point *P* is a mixture of different air molecules that have spent different times in the room. The local mean age of air ($\bar{\tau}_\rho$) measures the quality of air at a given point. In the exhaust air stream, the local mean age of air is equal to the nominal time constant (τ_n).

$$\tau_n = \frac{V}{q_v} \qquad (1\text{-}1)$$

where V is the room air volume and q_v is the ventilation flow rate. When there is more than one exhaust outlet, the nominal time constant is also defined as the weighted average of the local mean age of air in the exhaust air stream

$$\tau_n = \frac{\sum_m (Q_{ex,m}\bar{\tau}_{ex,m})}{\sum_m Q_{ex,m}} \qquad (1\text{-}2)$$

where m is an identification number unique for each exhaust air stream, $Q_{ex,m}$ is the airflow rate in the exhaust airstream, and $\bar{\tau}_{ex,m}$ is the age of air in the exhaust airstream. The room mean age of air ($\langle\bar{\tau}\rangle$) is equal to the spatial average of the local mean ages of air ($\langle\bar{\tau}_p\rangle$).

Figure 1-3 shows $\langle\bar{\tau}\rangle$ and τ_n for four different types of airflow. Following the REVHA definition, $\langle\bar{\tau}\rangle$ is $\tau_n/2$ in an ideal piston flow, and $\langle\bar{\tau}\rangle$ is equal to τ_n in fully mixed airflow. If there is short-circuit flow from the supply to the exhaust, the local mean age of air will be low in the short-circuited zone and high in the stagnant zone. The air change time for all the air in the room ($\bar{\tau}_r$) is equal to twice $\langle\bar{\tau}\rangle$.

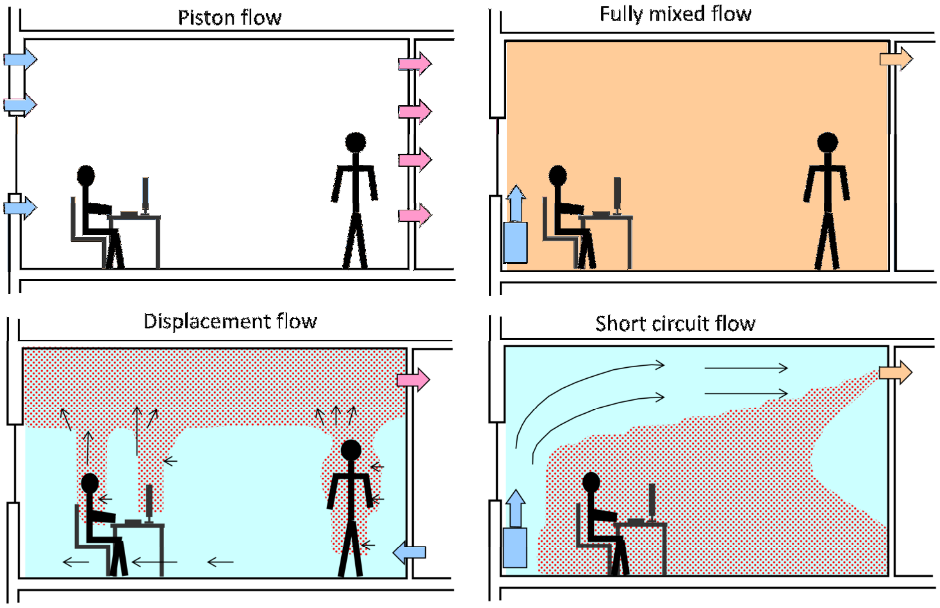


Figure 1-3. Room mean age of air and nominal time constant for different types of airflow. (Muller et al. 2013) (Prepared by author referring the reference)

As mentioned, the age of air is measured by measuring tracer gas concentration. The tracer step-down (decay) method and tracer step-up method are the common techniques for evaluating the age of air. In a decay test, the concentration of tracer gas at the start of the measurement is assumed to be uniform. The tracer gas concentration in the space then decreases at a rate that depends on the air-change rate. According to the ISO, the decay method can be used without problems up to an air change rate n of 10 [h^{-1}] in general. Following REHVA terms, from the decay test, the age of air at a point p in a space ($\bar{\tau}_p$) is given by

$$\bar{\tau}_p = \frac{1}{C_0}\int_0^{\infty} C_p(t)dt \qquad (1\text{-}3)$$

where C_0 is the concentration of the tracer gas at a time t = 0, and $C(t)$ is the concentration decay recorded at each point.

In a tracer step-up measurement, the tracer gas concentration is also assumed to be uniform at the beginning of the measurement. The tracer gas is injected into the outdoor air being delivered to the space at a constant rate. Tracer gas concentration increases to an equilibrium value C at a rate that depends on the air-change rate. The local age of air at a point p in the space ($\bar{\tau}_p$) is given by the following equation:

$$\bar{\tau}_p = \int_0^{\infty}\left[1 - \frac{C_p(t)}{C_{\infty}(t)}\right] \qquad (1\text{-}4)$$

The ISO guidebook also introduces two homogeneous emission methods: 1) the active homogeneous method and 2) the passive homogenous method. In the active homogeneous method, the tracer gas is fed at measured constant rates into zones by a suitable adjustable injection device and the local mean age of air is obtained from the quotient of the steady state concentration and injection rate per volume. On the other hand, in the passive homogeneous method, the tracer gas is emitted at a known constant rate into zones using diffusion sources. The local mean age of air is obtained from the quotient of the steady state concentration and the emission rate per unit volume. In the homogeneous emission method, $\bar{\tau}_p$ is calculated from the equation

$$\bar{\tau}_p = \frac{\varphi}{(q_V/V)} \qquad (1\text{-}5)$$

where φ is the measured trace gas concentration at steady state, q_V/V is the constant in the injection rate of a pure tracer gas in space, q_V is the injection rate of pure tracer gas, and V is the volume of the space. q_V is determined from

$$q_V = k_V \times V \qquad (1\text{-}6)$$

where k_V is a constant that can be estimated by the product of the anticipated air change rate and the desired tracer gas concentration at steady sate.

Although the same age of air concept is implemented in different publications, ventilation effectiveness indices are slightly different. In addition, the ISO only provides the evaluation method based on the age of air. Regarding the ventilation effectiveness indices, the ASHRAE standard defines the air change effectiveness (E). From the local mean age of air $\bar{\tau}_\rho$, the air change effectiveness of the test space E, is defined as

$$E = \frac{\tau_n}{\langle\bar{\tau}_v\rangle} \qquad (1\text{-}7)$$

where $\langle\bar{\tau}_v\rangle$ is the arithmetic average of the age of air measured at breathing level within a test space. Among other indices, this research consistently utilizes *E* by ASHRAE as the ventilation effectiveness index. A detailed explanation of this index is given in section 1.2.2.3. Similar to the ASHRAE standard (2002), the SHASE standard defines the standardized occupied zone concentration. The standardized occupied zone concentration (C_n) is defined as the inverse of *E* in the ASHRAE standard.

$$C_n = \frac{\langle\tau_v\rangle}{\tau_n} = \frac{C_a - C_{out}}{C_p - C_{out}} \qquad (1\text{-}8)$$

where $\langle\tau_v\rangle$ is the average age of air in the occupied zone, and τ_n is the nominal time constant on left side equation. C_a is the average occupied zone pollutant concentration under actual mixing conditions, C_p is the supplied air pollutant concentration, and C_{out} is pollutant concentration assuming perfect mixing on the right side equation.

In the REHVA guidebook, the air change efficiency (ε^a) is implemented. ε^a is defined as the ratio between the shortest possible air change time for the air in the room τ_n and τ_r. ε^a can also be described as the ratio between the lowest possible mean age of air τ_n /2 and $\langle\bar{\tau}\rangle$.

$$\varepsilon^a = \frac{\tau_n}{\bar{\tau}_r} \times 100 = \frac{\tau_n}{2\langle\bar{\tau}\rangle} \times 100 \ \ [\%\} \qquad (1\text{-}9)$$

The upper limit for this efficiency is 100 %, which occurs for ideal piston flow. It is worth noting that E and C_n are defined within air at breathing level (occupied zone) while $\mathcal{E}^a$ is defined with the entire room average including occupied zone. The REHVA guidebook also specifies the index that shows the conditions at a particular point. The local air change index is defined as the ratio between τ_n and $\bar{\tau}_p$. The local air change index (ε_p^a), is described as

$$\varepsilon_p^a = \frac{\tau_n}{\bar{\tau}_p} \times 100 \quad [\%\} \qquad (1\text{-}10)$$

Besides air change efficiency, REVHA defines contaminant removal effectiveness. The contaminant removal effectiveness (ε^c) is focused on the transport of pollution. It is a measure of how quickly an airborne contaminant is removed from the room and is defined as

$$\varepsilon^c = \frac{c_e}{c_{mean}} \qquad (1\text{-}11)$$

where c_e is the contaminant concentration in the exhaust and c_{mean} is the mean concentration of the contaminant in the room. The indices assume that the supplied air is not contaminated.

1.2.2.1 Perfect mixing

Figure 1-4 shows the definition of flow types based on ε^a and ε^c according to REHVA (Muller et al. 2013). ε^a evaluates how effective the supplied clean air replaces the room air. The piston flow whereby mixing does not occur between the supplied clean air and the room air is the most effective flow. This flow type has an ε^a value of 100%. Perfect mixing leads to an ε^a value of 50% and ε^c value of 1. Short circuit flow occurs when the ε^a value is lower than 50%.

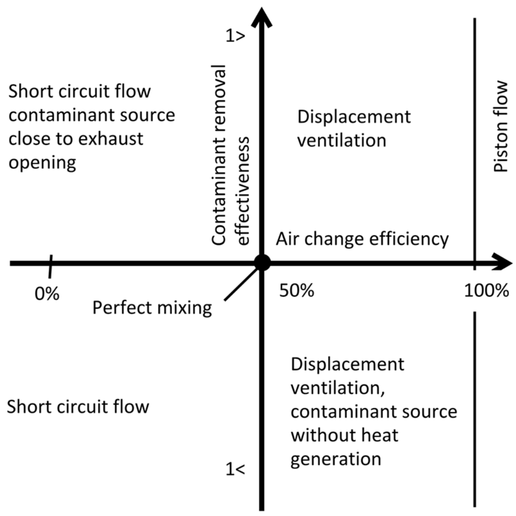


Figure 1-4 Definition of flow types based on air change efficiency and contaminant removal effectiveness. (Muller et al. 2013). (Prepared by author referring the reference)

On the other hand, according to ASHRAE Standard 129 (2002), perfect mixing is defined as a theoretical airflow distribution in which the concentration of all constituents in the air, and the age of air, are spatially uniform. Lastly, SHASE Standard 115 (2010) defines perfect mixing as simultaneous and equal distribution of supplied air. In other words, perfect mixing is a state in which a pollutant simultaneously spreads all over the space equally.

In this research, perfect mixing is assumed to occur when the air change effectiveness at all measured points is 1.0 ± 0.08. The value ±0.08 is based on the uncertainty in the tracer gas measurement assumed in ASHRAE Standard 129 (2002). It is assumed that perfect mixing can be achieved by using adequate mixing fans in the test space. It was confirmed that perfect mixing could be achieved by using three mixing fans as all the measured local air change effectiveness values in the perfect mixing test were in the defined range. More details about the procedure are given in Chapter 2.

On the contrary, short circuit flow is another important term in this research. Short circuit flow refers to entrainment flow with very poor mixing in the room as much of the supply air leaves the room without mixing with the room air. The ASHRAE Handbook (2009) mentions that poorly designed, installed, or operated mixing ventilation systems exhibit substantial short circuiting, especially ceiling-based systems in the heating mode (Offermann et al. 1989).

1.2.2.2 Measuring procedures

Table 1-2 summarizes the ventilation effectiveness indices, methods and required sampling points when evaluating space ventilation effectiveness in the ASHRAE, REHVA, ISO, and SHASE standards/guidebooks. The recommended measurement methods and sampling points are slightly different. The ASHRAE standard only recommends the use of the tracer gas step down and step-down method as the tracer gas measurement techniques, while the REHVA guidebook and SHASE standard introduce a pulse method. The ASHRAE standard introduces other tracer gas techniques, such as the constant concentration / injection method (Fortmann et al. 1990, Walker and Forest 1995, Walker and Wilson 1998) in its handbook (2009). On the other hand, the ISO mentions the homogeneous emission method, in addition to the tracer gas decay method. The tracer gas decay test is the most reliable tracer gas measurement technique as all standards/guideline recommend it. In terms of the sampling points, the ASHRAE standard stipulates that the local age of air of 25% of the workstations but not less than ten workstations and not less than the total work stations should be evaluated as a sampling point. On the other hand, the SHASE standard requires a minimum of three points or three repetitive measurement at one point in the target space. The measuring point should be near the center of the span (between columns) by the span or 10 m by 10 m. The ISO requires a minimum of three measurement points in order to gain information on variations that should not be close to the tracer gas sources (minimum 1 m distance) or close to an air supply terminal. Finally, REHVA only mentions that one or more places in the room or exhaust air may be evaluated, depending on the type of measurement. The sampling point requirements are diverse. However, the ASHRAE has the most stringent sampling point requirements as it requires 10 points at a minimum.

Table 1-2 Summary of ventilation effectiveness indices, tracer gas method, and sampling point specified in different standards/guidebooks.

	ASHRAE Standard 129	REHVA Guidebook No.2	ISO 16000-8	SHASE Standard 115
Indices	-Air change effectiveness	-Air change efficiency -Local air change index -Contaminant removal effectiveness -Local air quality index	N/a	-Standardized occupied zone concentration
Tracer gas Methods	-Tracer gas step-up -Tracer gas step-down	-Tracer gas step-up -Tracer gas step-down -Pulse -Homogeneous constant emission method	- Tracer gas decay - Homogeneous emission method	-Tracer gas step-up -Tracer gas step-down -Pulse
Sampling points	-Twenty-five percent of the workstations but not less than ten workstations and not less than the total work stations if the test space contains fewer than ten	-One or more places in the room or in exhaust air	- A minimum of three measurement points in order to gain information on variations. - The points should not be close to the tracer gas sources (minimum 1 m distance) or close to an air supply terminal	-Near the center of the span (between columns) by the span or 10 m by 10 m; a minimum of three points in the target space

1.2.2.3 Air change effectiveness

As explained in section 1.2.2, several indices for ventilation effectiveness have been proposed by different organizations and researchers. Among them, the age of air method for determining the air change effectiveness defined in ASHRAE Standard 129 is implemented in this research. The age of air is the average amount of time that has elapsed since the air molecules at the location and can be caluclated from Equations 1-3, 1-4 and 1-5. In the ASHRAE standard, the age of air at a point *i* in a space is defined as A_i, while REHVA defines it as the age of air at a point *p* in a space $\bar{\tau}_p$. This research will use the nomenclature favored by the ASHRAE. Following the steps illustrated in Appendix C of ASHRAE Standard 129, which focuses on the decay test, the corrected age of air ($A_{i,corr}$) is calculated by

$$A_{i,corr} = \frac{C_{i,avg}(t_{stop}-t_{start})}{C_i(t_{start})} + \frac{C_i(t_{stop})}{LC_i(t_{start})} \qquad (1\text{-}12)$$

where t_{stop} is the time of the final tracer gas measurement, t_{start} is the time when the tracer injection is stopped at the beginning of tracer gas decay, $C_{i,avg}$ is the time-averaged tracer gas concentration at location *i* between time t_{start} and t_{stop}, and *L* is the negative of the slope of the natural logarithm of concentration as a function of time calculated at the end of the tracer gas concentration decay. In this study, $A_{i,corr}$ is considered as A_i. Furthermore, t_{start} is determined as the time the mixing fans are stopped and t_{stop} as three times the nominal time constant. From A_i, the local air change effectiveness (E_i) is defined as

$$E_i = \frac{\tau_n}{A_i} \ [-] \qquad (1\text{-}13)$$

where τ_n is the nominal time constant. From the definition, τ_n is the weighted average of the age of air in the exhaust air stream (A_E). As the experiments in this research utilize a single exhaust, τ_n is equal to A_E. In addition, the arithmetic mean of E_i in the low measuring plane, high measuring plane, and overall test space are referred as E_{low}, E_{high}, and E, respectively, and will be described in detail in Chapters 2, 3, and 4. The A_E of each case is computed from the correlation between the exhaust air volume in the HVAC control system and the computed actual age of air in exhaust air stream, which is obtained from perfect mix tests.

1.2.3 Temperature effectiveness

Introducing warm air into a test space may result in thermal stratification under the ceiling due to a short circuit above the occupied zone caused by buoyancy effects. Liu's study (2015) suggests that the warm air from diffusers with small $T_{0.25}/L$ causes greater temperature gradients in the upper region of the occupied zone. Ventilation effectiveness might be low with such high thermal stratification, although the calculated ADPI is quite high. Temperature effectiveness (Etheridge et al. 1996), also defined as ventilation effectiveness for heat removal (Awbi et al. 1993) is implemented to evaluate the temperature gradient in the test space. Similar to the concept of the air change effectiveness, the temperature effectiveness ($\mathcal{E}_T$) is defined as

$$\varepsilon_T = \frac{T_S - T_E}{T_S - \langle T \rangle_0} \quad [-] \qquad (1\text{-}14)$$

where T_S is the supply air temperature, T_E is the exhaust air temperature, and $<T>_0$ is the average temperature in the occupied space (average of $T_{0.1}$, $T_{0.6}$, $T_{1.1}$, $T_{1.4}$, and $T_{1.8}$ is used, where the index indicates the vertical distance of the sensor from the floor in meters). In this research occupied space or zone is defined between floor and 1.8 m above floor as specified in ASHRAE Standard 55 (2013).

1.2.4 $T_{0.25}/L$ and *ADPI*

The ADPI is defined as the percentage of the occupied zone that maintains an acceptable velocity and temperature. The region with an acceptable velocity and temperature is determined by the local effective draft temperature (EDT) that combines air temperature difference and air speed (Rydberg et al. 1949, Straub et al. 1956). The EDT for the cooling condition is defined as

$$\text{EDT} = T_i - T_a - 8.0(V_i - 0.15) \quad [°C] \qquad (1\text{-}15)$$

where T_i is the temperature at the test point i, T_a is the spacious average temperature [°C], and V_i is the local air speed [m/s]. The EDT for the heating condition is defined as (Liu et al. 2015)

$$\text{EDT}\ (\theta) = T_i - T_a - 9.1(V_i - 0.15) \quad [°C] \qquad (1\text{-}16)$$

In addition, the ADPI incorporates the throw and the characteristic length, and it provides design variables for selecting diffusers. The throw (T) is defined as the distance from a diffuser to a point where the maximum velocity in the stream cross section has been reduced to a selected terminal velocity (ASHRAE Standard-70. 2006). The dominant diffuser property for air distribution is the supply jet throw length at which the jet velocity decreases to a selected terminal value of 0.25 m/s ($T_{0.25}$). The characteristic length *(L)* describes the room geometry using the distance at which the air jet travels. The ratio $T_{0.25}/L$ is a dimensionless number that characterizes a supply diffuser momentum (including capacity of a diffuser to mix/entrain surrounding air) for a given flow rate. $T_{0.25}/L$ has the largest impact on ADPI, and together, $T_{0.25}/L$ and ADPI are used in the diffuser selection guidelines that appear in the ASHRAE Handbook (2009). The handbook lists the relationships between the ADPI and dimensionless ratio of $T_{0.25}/L$ for various diffuser types at different thermal loads. One is able to design an HVAC terminal system, including selecting the diffuser and determining its layout, by ensuring the $T_{0.25}/L$ of the system gives an ADPI that is greater than 80%, which is the threshold of air distribution system designs.

As $T_{0.25}/L$ is a critical parameter determining air distribution, this research utilizes it to assess the performance of air distribution. $T_{0.25}$ is defined for different air flow rates in isothermal conditions from a manufacturer's catalogue data for several diffuser types including adjustable blade diffusers (ASHRAE. 2009). However, $T_{0.25}/L$ for a vertical flow is not defined in the literature. The throw of a warm vertical jet may significantly travel less than an isothermal jet as it is strongly influenced by the buoyancy force. The $T_{0.25}$ in a manufacturer's catalogue for vertical flow (isothermal flow) is modified by utilizing the empirical chart in the ASHRAE Handbook (2009) with measured supply air temperature and room air temperature. From this chart, the throw correction factor for vertical flow with regard to room supply air temperature differences is determined. In addition, the ceiling height is defined as L for vertical flow.

1.2.5 Diffuser selection methods

In mixing ventilation, diffuser selection, location, supply air volume, discharge velocity, and air temperature different result in air motion in the occupied zone. This section introduces diffuser selection methods recommended by the ASHRAE, REHVA and SHASE.

First of all, the ASHRAE recommends three methods that can be used to select diffusers: 1) appearance, flow rate, and sound data, 2) isovels (lines of constant velocity) and mapping, and 3) comfort criteria (ASHRAE Handbook 2009). It recommends the use of at least two methods when selecting diffusers. The last method, comfort criteria, introduces the ADPI diffuser selection method. Utilizing $T_{0.25}/L$ from manufactures' isothermal catalogue throw data at terminal velocity of 0.25 m/s $T_{0.25}$ and the dimensions available for the throw L on the diffusers, designers maximize space cooling comfort. ADPI method is most usable in space with ceiling heights between 2.4 m and 3 m (ASHRAE Handbook 2009). However, the ADPI diffuser selection method introduced in the ASHRAE Handbook is only valid for overhead air distribution systems under cooling operation (ASHRAE Standard 113. 2009). Liu et al. (2015) expanded the ADPI concept to the heating regime and obtained ADPI values with recommended design criteria for various types of diffusers under both the cooling and heating modes. Liu's study is reviewed in greater detail in section 1.3.2.2.

REHVA recommends the air distribution method. Figure 1-5 describes the design chart for air distribution in rooms (Nielsen 2007). The chart is based on the minimum and maximum allowable flow rates (q_o) of air supplied to the room, and the maximum temperature difference between return air and supply air (ΔT_o). The figure indicates that a flow rate is necessary to achieve a given air quality. The supply of flow with a momentum generates mixing in the occupied zone, and it characterize an air distribution. The drawback is that this flow may generate a draft when the flow rate is above a certain level. The temperature difference between return and supply air is also restricted as a too high temperature difference may either cause a draft in the occupied zone or create a too large temperature gradient in the room. The optimal air distribution system design is one that will give a sufficient supply of fresh air and a draft free air movement in the occupied zone.

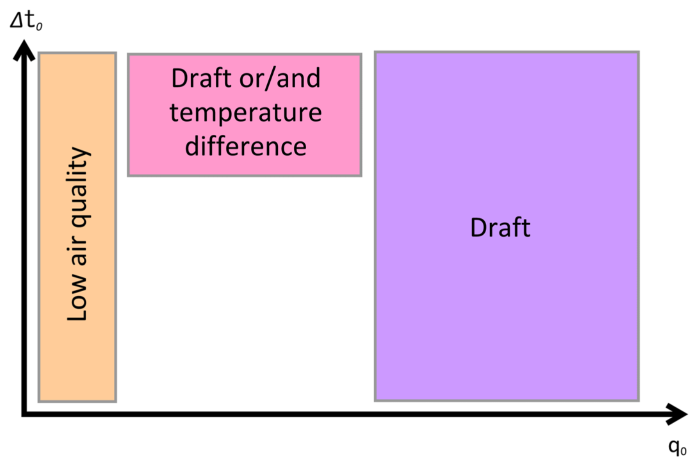


Figure 1-5 Design chart that indicates the restrictions on the flow rate and on the return and supply temperature difference. (Neilsen 2007)

(Prepared by author referring the reference)

REHVA also points out some aspects should be considered when selecting and locating diffusers: 1) equal air distribution in the room space without a draft, 2) throw length in the cooling mode condition by using the manufacture’s product data, 3) short circuit prevention, especially in the heating mode, 4) the presence of obstacles that might obstruct supplied air flow, and 5) the noise level, which should be according to the standards. In practice, air distribution in a room is often designed by product-related software or product-related design graphs. Figure 1-6 shows a typical design graph. The total pressure difference over the diffuser (Δp_t) is given as a function of the diffuser size (a_o) and volume flow (q_o). The sound pressure level (L_p) and the throw length (l_{02}) curves shown in the graph together with q_o and Δp_t are used to select the diffuser.

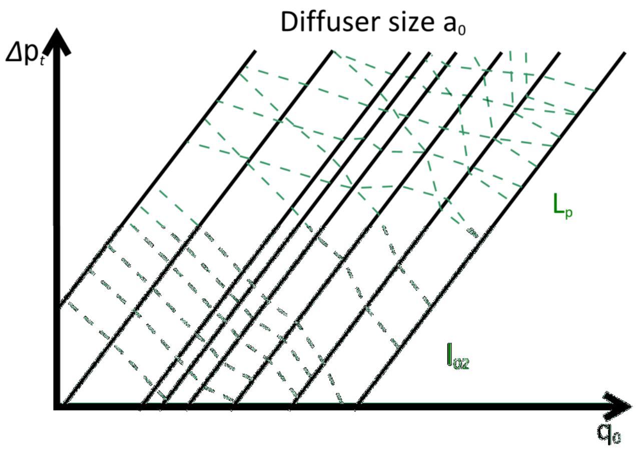


Figure 1-6 Graph for selecting a diffuser for a room. (Muller et al. 2013)

(Prepared by author referring the reference)

Finally, the SHASE Handbook (2010) introduces flow patterns in mixing ventilation, and provides diffuser selection guides including the ADPI method introduced by the ASHRAE. The SHASE suggests equations for determining the diameters and number of nozzle diffusers in a space based on the diffusion range of a free air jet (Nomura, 1960, Hirayama et al. 1961,). This method suggests that the air jet should reach the floor under the heating mode. It also introduces the selection guide for high side wall jet diffusers under the cooling mode (Kubota 1977). This method could be used to determine the height of diffusers in rooms with high side walls.

1.3. Literature review of related studies

This section reviews previous studies in the relevant literature. The section is divided into two subsections. Section 1.3.1 reviews studies related to the ventilation effectiveness of mixing ventilation. Section 1.3.2 then introduces previous studies on the ADPI. The most recent ADPI studies by Liu et al. are reviewed in section 3.2.2

1.3.1 Ventilation effectiveness in mixing ventilation

This section reviews previous studies related to ventilation effectiveness. Section 1.3.1.1 introduces studies related to the influence of inlet/outlet locations. Section 1.3.1.2 reviews the influence of internal objects and section 1.3.1.3 reviews ventilation effectiveness under the heating mode.

1.3.1.1 Influences of inlet/outlet locations

The impact of supply air diffuser (inlet) and exhaust (outlet) locations on air distribution and their ventilation effectiveness in a space have been investigated by many researchers (Shinha et al. 2000, Lee et al. 2007, Khan et al. 2006). Cao et al. (2013) summarized the numerous studies on mixing ventilation that considered locations of air flow inlets and outlets. Figure 1-7 shows their summary of mixing ventilation studies that considered inlet and outlet configurations (Boyle Son 1899, Clements 1975, Sandberg et al. 1986, Nielsen 1991, Sandberg et al. 1992, Awbi et al. 1993, Lee et al. 2004, Cao et al. 2010, Krajecik et al. 2012)

In addition, Sinha et al. (2000) compared the impact of different inlet and outlet locations by modelling their use using computational fluid dynamics. The study found that the most effective combination of inlet and outlet positioning is with the inlet near the floor and exhaust near the ceiling because the buoyancy force increases the intensity of recirculation in such a combination. When considering the position of air suppliers in the upper part of the room, Lee et al. (2007) experimentally compared high wall jet inlets and grill diffusers with typical ceiling diffusers. Their results show that the air inlet position and type are important determinants in the distribution of airborne contaminant concentrations. The ceiling diffuser overall produced more efficient ventilation than the wall jet air inlet. Overall, the ceiling diffuser produced more efficient ventilation than the wall jet air inlet. Unlike the air supply location, the air exhaust location has a small impact on the structure of room air flow in most room applications (Muller et al. 2013). This is because there is a rapid decay of velocity with increasing distance from the exhaust opening. However, the exhaust location may influence air change effectiveness and contaminant removal effectiveness. In Khan’s study (2006), the arrangements of wall inlets and outlets greatly influenced contaminant concentration. However, the influence of the outlet location was minimal with a ceiling diffuser inlet. As the air near the exhaust is not driven by jet momentum but by negative pressure in the air, the velocities near the exhausts are relatively small. Therefore, the effects of the exhaust location on room airflow pattern are relatively small in most applications.

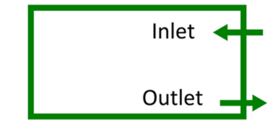
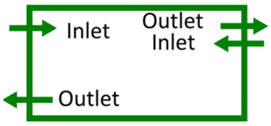
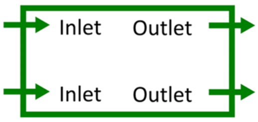
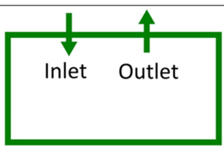
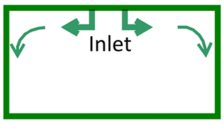
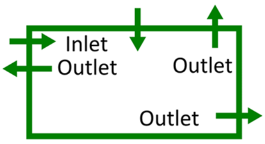
	Locations of airflow inlet and outlet	Methods	Main focus	Findings
Boyle Son (1899)		Investigation	Airflow distribution and indoor air quality.	- This is one of the very earliest forms of mechanical ventilation - Down draught ventilation by mechanical impulsion is pronounced by public health experts to be highly prejudicial to health.
Clements (1975)		Model experiment	Airflow pattern and the effect of Archimedes number.	- The air pattern is almost a function of the Archimedes number. As Ar increases, the jet deflection from the horizontal increases.
Sandberg et al. (1986)		Experimental study	Air exchange efficiency and contaminant exposure.	- The ceiling-to-floor system gives rise to a comparatively rapid exchange of the air with heating. However, the evacuation of the contaminant is delayed.
Nielsen (1991)		Modeling and experimental study	Simplified design method.	- Simplified design models work well with simple geometry - Measurements show significant deviation.
Sandberg et al. (1992)		Experimental study	The effect of Archimedes number on the airflow distribution.	- The critical supply Archimedes number at which the jet breaks away from the surface as soon as it leaves the nozzle was just below 0.03.
Awbi and Gan (1993)		Numerical study	Air distribution and ventilation effectiveness.	- Air distribution systems should be different for heating and cooling in order to achieve a comfortable room environment.
Lee and Awbi (2004)		Experimental and numerical study	The effects of partitions on the room air quality as well as ventilation performance.	- Increasing the partition gap underneath from 0%H to 10%H causes an overall improvement in the air change efficiency.
Cao et al. (2010)		Experimental study	Maximum velocity decay in the air distribution via attached plane jet.	- Attached plane jet can be used as an effective method to avoid draught in mixing ventilation conditions.
Kraj cík et al. (2012)		Experimental study	Air distribution and ventilation effectiveness	- The ventilation effectiveness varied between 0.4 and 1.2, where 1 is complete mixing, which depends on the position of air terminal devices.

Figure 1-7 Summary of mixing ventilation studies regarding inlet and outlet configurations. (Cao et al. 2013) (Prepared by author referring the references)

1.3.1.2 Influences of internal objects

Space partitioning such as cubicles and internal objects such as furniture or even occupants may also affect effectives of air distribution (Shaw et al. 1993, Lee et al. 2004, Wu et al. 2015). Shaw et al. (1993) showed the presence of cubicles (with partition height of 1.9 m in a space with total height of 2.9 m) had no significant effect on the air distribution patterns. They also found that the effect of the layout of a cubicle on the ventilation effectiveness is very small. A study on the impact of cubicle height by Lee (2004) shows that internal partition up to 60% of the room height has a very small impact on the air distribution. Partitions up of 80% of the room height have a significant impact on the room flow. When considering occupants, Wu et al. (2015) conducted a test with real walking occupants in a test chamber and analyzed their impact on CO_2 concentration and temperatures distribution in the space with three different ventilation methods (stratum, displacement, and mixing ventilation). The study found that short term-walking did not change the temperature or CO_2 concentration profiles. However, mixing occurred when occupants walked for longer periods of time.

1.3.1.3 Ventilation effectiveness under the heating mode

The studies mentioned in section 1.3.1.1 and 1.3.1.2 showed that many factors may influence the supply air distribution in mixing ventilation with an emphasis on cooling. In mixing ventilation, the overall effectiveness of air distribution is slightly better or worse than that with perfect mixing. Fewer studies have focused on the ventilation effectiveness of mixing diffusers used for space heating. Air distribution with all-air-heating presents major challenges for mixing air distribution. Researchers found low ventilation effectiveness under heating condition (Fisk et al. 1997, Offermann et al. 1989, Krajcik et al. 2012, Tomasi et al. 2013, Novoselac et al. 2003). Fisk et al. (1997) conducted experiments that used overhead all-air-heating systems with minimum supply air flow rates of typical VAV systems. The air change effectiveness was significantly lower than 1.0 in each experiment. The measured air change effectiveness was in range of 0.69–0.91 with a mean value of 0.81. Offermann et al. (1989) measured ventilation effectiveness and ADPI under heating conditions with recommended minimum ventilation rates while considering different supply and return air positions. For the ceiling supply/return configuration, ventilation effectiveness was 0.73 when the temperature difference of supply air temperature and room average temperature was 8 °C. It was even lower, 0.66, when the difference was 13 °C. Short-circuit flow from the supply to exhaust was apparent in each configuration. Krajcik et al. (2012) and Tomsai et al. (2013) measured air change efficiency and temperature effectiveness in a test chamber with various combinations of radiant floor heating and mixing ventilation. All-air heating systems often produce stagnant air in the occupied space of the room with relatively uniform low temperature in this stagnant zone. It was found that with stagnant cold air in occupied zone, the fresh hot supply air short circuit in the upper part of the room causes very poor ventilation effectiveness in the occupied space. The aforementioned low ventilation effectiveness under heating conditions is considered into outdoor intake volume required by building standard. ASHRAE Standard 62.1 (2010) states that a zone air distribution effectiveness of 1.0 can be achieved when the ceiling supply of warm air is less than 8 °C above the space temperature and the supply air jet throw with a velocity of 0.8 m/s ($T_{0.8}$) reaches the lower part of the room (that is 1.4 m above the floor level). When this 0.8 m/s jet throw does not reach the lower part of the room or when the supply-room temperature difference is larger than 8 °C, the nominal ventilation effectiveness is 0.8.

1.3.2 ADPI

The following subsections review two categories of ADPI studies. Those related to legacy applications of the ADPI to the cooling mode, which are reviewed in section 1.3.2.1. Section 1.3.2.1 reviews recent studies on the ADPI that re-construct the ADPI based on commonly used diffusers and its application under the heating mode.

1.3.2.1 Legacy ADPI

The legacy ADPI method first applied to the cooling mode was developed by Miller and Nash (1971). They derived it from a subjective response to the air temperature difference and velocity draft proposed by Houghten et al. (1938). As introduced in section 1.2.4, the ADPI is determined by the maximum air speed and EDT. The ADPI is the percentage of testing points falling into the range of the EDT at an acceptable air speed. The EDT is calculated from Equation 1-14, which was first introduced by Rydberg et al. (1949) then modified by Straub (1956) in a discussion of the paper by Koestel et al. (1955). The acceptable range of the EDT under the cooling mode is between -1.7 °C and 1.1 °C with an air speed less than or equal to 0.35 m/s. The lower boundary of the EDT, -1.7 °C, has a good agreement with Houghten's data for 80% of the occupants reporting comfort (Nevins et al. 1972). On other hand, the upper limit of EDT, 1.1 °C is satisfactory, as indicated by Koestel et al. (1955). Furthermore, a maximum acceptable velocity of 0.35 m/s was recommended by Nevins and Miller (1972).

The ADPI diffuser/ grille selection guidance is broadly applied with the characteristic length introduced in section 1.2.4 and internal thermal load (Miller et al. 1969, Miller et al. 1970; Miller 1971 Miller et al. 1971; Miller et al. 1972; Miller 1979). The ADPI of each diffuser/ grille type is described as a function of the ratio of jet throw length T_v to room characteristic length L, T_v/L, and room load (ASHRAE - HVAC Application 2007; ASHRAE-Fundamentals 2009). As shown in Figure 1-8, most studies conducted under the cooling condition found that the ADPI has a concave-like shape via the $T_{0.25}/L$. The figure illustrates that ADPI is low when $T_{0.25}/L$ is small and large, and there is a point where a maximum ADPI is achieved (Miller et al. 1969, Miller et al. 1970; Miller 1971 Miller et al. 1971; Miller et al. 1972; Miller 1979).

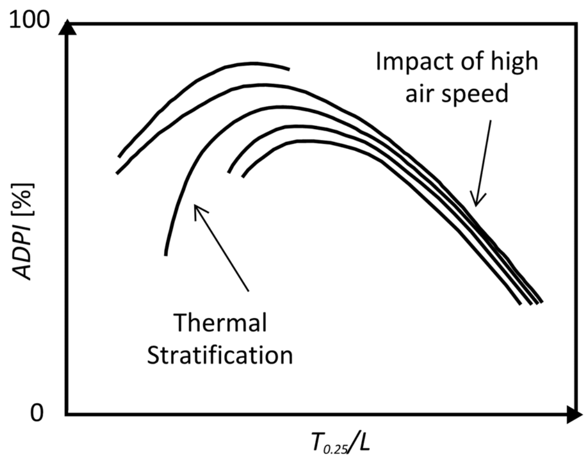


Figure 1-8 General ADPI profiles vs. $T_{0.25}/L$ under the cooling mode. (Miller et al. 1971) (Prepared by author referring the reference)

As an indicator of occupant comfort (John 2012), the ADPI method has been used to evaluate thermal comfort along with the predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) that involve four additional factors: (1) the mean radiant temperature, (2) the relative humidity, (3) the metabolic rate, and (4) clothing insulation. Chung and Lee (1996) reported an evaluation of thermal comfort under three ventilation patterns in terms of PPD and ADPI. The results revealed that ADPI may not be an adequate representation of draft risk. Other studies investigated experimentally the jet-flow characteristics from a high sidewall grille and a linear diffuser (Chow et al. 1994, Chow et al. 1996). The ADPI was found to be linearly related to PPD for the high sidewall grill. Several studies have attempted to correlate ADPI with other parameters, such as Archimede's number, jet momentum, and supply air temperature (Chow et al. 1996; Rutman et al. 2005; Ng et al. 2008; Corgnati et al. 2009; Gao and Lee 2009). The ADPI method has also been employed to optimize the design of a floor-based air-conditioner (Corgnati et al. 2009).

1.3.2.2 Updated ADPI and ADPI applications under the heating mode

The aforementioned legacy ADPI diffuser selection guide was mainly developed in the 1970s, when HVAC systems that deliver warm air in the heating mode were not common. However, the all-air heating HVAC systems are much more common these days, and the systems have to meet both heating and cooling requirements. (Krarti 2008; Platt et al. 2010; Vakiloroaya et al. 2014). Even though ASHRAE Standard-113 (2009) states that "the ADPI method for mixing systems should be applied to traditional overhead air distribution systems under cooling operation only," the same diffusers are used for heating in the winter. The lack of an ADPI method for the heating mode often causes underperformance of all-air delivery systems when they are used for heating. Moreover, in the ASHRAE Handbook (2009), only seven diffuses are available. Since the 1970s, more diffusers have become available, and it has become more popular to utilize various diffusers depending on the situation.

Against this background, Liu et al. (2016) developed an ADPI method for the heating mode by deriving an EDT formula for the heating mode. Then, they conducted extensive experiments under both cooling and heating modes in terms of ADPI to update the database. The experiments were conducted with 16 of the most current diffuser types, applications that included both heating and cooling operations, and building loads that are typical for modern buildings. Figure 1-9 illustrates examples of the 16 types of diffusers. The figure shows some manufactures' models. However, the intention of this figure is to provide an idea of diffuser types and thus it is not exhaustive.

To develop an ADPI for the heating mode, they first predicted the ADPI curves via $T_{0.25}/L$ for the heating mode as shown in Figure 1-10. When $T_{0.25}/L$ is small, two phenomena, "stagnant flow" and "thermal stratification," may be observed when warm air is introduced into the space. "Stagnant flow," which is also refereed to as "short circuit" flow, is a phenomenon whereby the supply air short circuits above the occupied zone, and "thermal stratification" is whereby the supply air reaches the occupied zone but creates a great vertical temperature difference in the space. "Stagnant flow" still creates a high ADPI because the warm air stagnates above the occupied zone, and as the occupied zone is less affected by the supply air, the stagnated air has a relatively uniform velocity and temperature. When $T_{0.25}/L$ increases, a large amount of air leads to an unacceptable air speed or "no impact of high speed." Then, they derived an EDT under the heating mode (Equation 1-16) by plotting the thermal acceptance range of the PMV (Fanger, 1970) for the cooling mode and heating mode with a metabolic rate of 1.15 Met., clothing insulation of 0.5 Clo., and relative humidity of 50%. In addition, they set the following three criteria for the ADPI for the heating mode; 1) an EDT ranging from -2.2 °C to 2 °C, 2) a local air speed less than or equal to 0.35 m/s, and 3) an overall vertical temperature gradient lower than 3 °C/m, and a local maximum temperature difference between 0.1 m to 1.1 m is less than 3 °C.

As stated in ASHRAE Standard 55 (2010), the air speed criterion is less than or equal to 0.35 m/s for warm air. Occupants generally accept an air speed greater than 0.2 m/s. Other important criteria are the vertical air temperature difference and thermal stratification. ASHRAE Standard 55 (2010) allows a maximum vertical air temperature of 3°C between 1.1 m and 0.1 m.

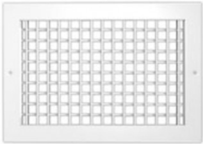
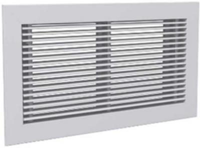
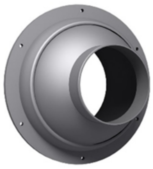
	Terminal device	Typical installation	Sample pictures	Manufacture's models of sample pictures
1	Adjustable blade grilles	High sidewall		Nailor: Model Series - 51D
2	Fixed blade grills	High sidewall		Price: RCG - Reversible Core Grille
3	Linear bar grilles	High sidewall/Sill		Price: LBP - Linear Bar Grille
4	Nozzles	High sidewall		Price: ND - Nozzle Diffuser
5	Round ceiling diffuser	Ceiling		Metalaire: Series 3000 - Round Adjustable Diffusers
6	Square ceiling diffuser	Ceiling		Price: SCD - Square Cone Diffuser
7	Perforated diffusers-round pattern	Ceiling		Price: PDMC - Perforated Face Supply Diffuser, Modular Core
8	Perforated diffusers-directional pattern (4-way)	Ceiling		Titus: PCS-Steel perforated neck-mounted curved blade deflectors

Figure 1-9 Sample pictures of diffuser types

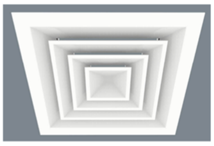
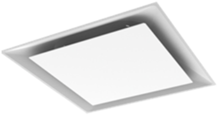
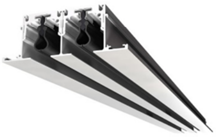
	Terminal device	Typical installation	Sample pictures	Manufacture's models of sample pictures
9	Louvered face diffusers-with lip on deflector blade	Ceiling		Metalaire: Series 5500DD - Aluminum Architectural Fixed Louver Face Diffuser
10	Louvered face diffusers-without lip on deflector blade	Ceiling		Metalaire: Series 5000 - Aluminum Directional Diffuser
11	Plaque face diffusers	Ceiling		Titus: OMNI- Steel Plaque Face Diffuser
12	Linear slot diffusers	Ceiling		Price: SDS - Linear Slot Diffuser
13	T-bar slot diffusers	Ceiling		Price: TBD7 - T-Bar Diffuser
14	Swirl diffusers	Ceiling		Titus: TSW-Steel square, swirl face diffuser
15	N-slot diffusers	Ceiling		Krueger: PTBSC-Steel Plenum Slot Diffuser

Figure 1-9 Sample pictures of diffuser types (continued)

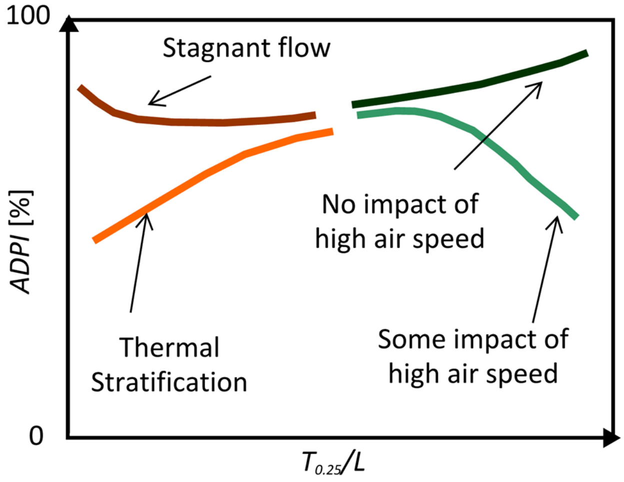


Figure 1-10 General ADPI profiles as functions of $T_{0.25}/L$ for the heating mode. (Liu et al. 2016) (Prepared by author referring the reference)

Table 1-3 show the findings of Liu's research (2016). The table shows the acceptable air distribution (ADPI > 80%) at certain ranges of $T_{0.25}/L$ with 15 commercially available types of diffusers under the cooling mode with two different loads of 25 and 50 W/m^2 and heating mode at a load of 35–40W/m^2. The large acceptable ranges of $T_{0.25}/L$ for several of the 15 types of diffusers implies that those diffusers have more flexibility in HVAC design than others. It turns out that linear bar grilles and linear slot ceiling diffusers have the largest range for the heating condition. HVAC design should take into consideration the operation of the HVAC system under both cooling and heating conditions. Therefore, this investigation compares the effects of blade angle and deflector adjustment on ADPI values. The findings would guide manufacturers in improving the design of their products to generate a more satisfactory air distribution.

Table 1-3 Ranges of $T_{0.25}/L$ for various diffuser types when the ADPI is higher than 80% under the cooling and heating conditions. (Liu et al. 2016)

(Prepared by author referring the reference)

Diffuser type	Mode	Loads [W/m^2]	Range of $T_{0.25}/L$	$T_{0.25}/L$ for maximum ADPI
Adjustable blade grilles - 45° up	Cooling	25	0.5-1.1	0.8
		50	0.6-1.1	0.9
	Heating	35-40	0.6-1.0	1.0
Adjustable blade grilles - 0°	Cooling	25	1.2-2.2	1.8
		50	1.7-2.2	2.0
	Heating	35-40	1.1-2.2	1.8
Adjustable blade grilles - 45° down	Cooling	25	-	0.9
		50	-	1.0
	Heating	35-40	0.6-0.8	0.7
Fixed blade grills - 15° up	Cooling	25	1.1-2.2	1.1
		50	1.2-2.2	1.9
	Heating	35-40	1.2-2.2	2.2
Fixed blade grills - 15° down	Cooling	25	1.6-2.2	1.9
		50	1.9-2.2	1.9
	Heating	35-40	1.2-2.2	1.3
Linear bar grilles - high sidewall	Cooling	25	0.8-1.5	1.3
		50	1.0-1.5	1.0
	Heating	35-40	0.7-1.5	1.3
Linear bar grilles - sill	Cooling	25	0.8-2.1	2.1
		50	1.3-2.1	1.8
	Heating	35-40	0.6-2.1	1.8
Nozzles - high sidewall	Cooling	25	0.9-2.2	0.9
		50	0.9-2.2	0.9
	Heating	35-40	1.2-1.9	1.5
Round ceiling diffuser	Cooling	25	0.5-2.3	1.8
		50	0.7-2.3	2.2
	Heating	35-40	1.4-2.1	2.1

Table 1-3 Ranges of $T_{0.25}/L$ for various diffuser types when the ADPI is higher than 80% under the cooling and heating conditions. (Liu et al. 2016) (continued)

(Prepared by author referring the reference)

Diffuser type	Mode	Loads	Range of $T_{0.25}/L$	$T_{0.25}/L$ for maximum ADPI
Square ceiling diffuser	Cooling	25	1.0-2.6	2.1
		50	1.0-2.6	2.6
	Heating	35-40	2.1-2.5	2.5
Perforated diffusers - round pattern	Cooling	25	0.6-2.4	1.9
		50	0.8-2.4	1.9
	Heating	35-40	2.0-2.5	2.5
Perforated diffusers - directional pattern	Cooling	25	0.7-3.0	3.0
		50	0.7-3.0	3.0
	Heating	35-40	2.4-2.9	2.4
Louvered face diffusers - with lips	Cooling	25	1.4-4.7	3.0
		50	1.4-4.7	1.5
	Heating	35-40	2.6-3.3	3.0
Louvered face diffusers - without lips	Cooling	25	1.0-3.3	3.3
		50	1.0-3.3	1.0
	Heating	35-40	2.4-3.3	3.3
Plaque face diffusers	Cooling	25	1.0-2.2	2.2
		50	1.0-2.2	1.3
	Heating	35-40	2.1-2.6	2.3
Linear slot diffusers	Cooling	25	1.1-3.4	2.8
		50	1.1-3.5	2.5
	Heating	35-40	2.1-3.4	2.9
T-bar slot diffusers	Cooling	25	1.2-2.2	1.2
		50	1.2-1.8	1.5
	Heating	35-40	1.8-2.1	2.1
Swirl diffusers	Cooling	25	0.4-1.7	1.7
		50	0.4-1.7	1.7
	Heating	35-40	1.4-1.7	1.7
N-slot diffusers	Cooling	25	1.4-2.4	1.8
		50	1.4-2.4	1.8
	Heating	35-40	1.5-2.4	2.1

1.4 Structure of the present thesis

The contents and structure of the present thesis are shown in Figure 1-11. This thesis consists of the following five chapters:

In Chapter 1, “Introduction,” the objectives of this research are formulated with a background description and review of relevant literature. This chapter introduces current ventilation methods, indices related to ventilation effectiveness, and diffuser selection methods. A review of previous studies on ventilation effectiveness in terms of mixing ventilation and ADPI is also provided.

In Chapter 2, “Experimental Study on Air Change Effectiveness in Mixing Ventilation,” experimental measurements in a full-scale test room with various types of diffusers, air flow rates and internal loads to evaluate the air change effectiveness and the temperature effectiveness are described. These experiments were conducted for both heating and cooling cycles, though there was an emphasis on the heating mode. All diffusers tested showed similar results, although each diffuser had a unique shape. The ranges of air change effectiveness and temperature effectiveness were examined within the recommended range of $T_{0.25}/L$ regarding ADPI. The studies provided fundamental diffuser performance data that considers both thermal comfort and ventilation effectiveness.

In Chapter 3, “Improving Ventilation Effectiveness under the Heating Mode,” experimental measurements of the ADPI, air change effectiveness, and temperature effectiveness in a same full-scale test room are described. The experiments examine simple strategies for overcoming the challenges of poor ventilation under heating conditions while maintaining an acceptable ADPI. The strategies include the use of diffuser deflector adjustment with linear slot diffusers and adjustable blades grills, room-supply air temperature difference with vertical flow of linear slot diffusers, and exhaust locations with adjustable blade grills. In addition, it provides new diffuser selection data in the form of ADPI values for diffusers with a vertical jet projection (vertical flow).

In Chapter 4, “Assessing Measuring Procedure for Ventilation Effectiveness,” extensive experiments on mixing ventilation that were conducted in the same test chamber are described. The experiments examined the vertical, horizontal and overall variance of local air change effectiveness to evaluate whether a more practical evaluation of ventilation effectiveness would be possible if fewer measuring points were used. Furthermore, they analyzed the correlation of thermal effectiveness and air change effectiveness as an alternative method for interpreting air change efficiency.

In Chapter 5, “Conclusive summary,” the results described in each chapter are summarized.

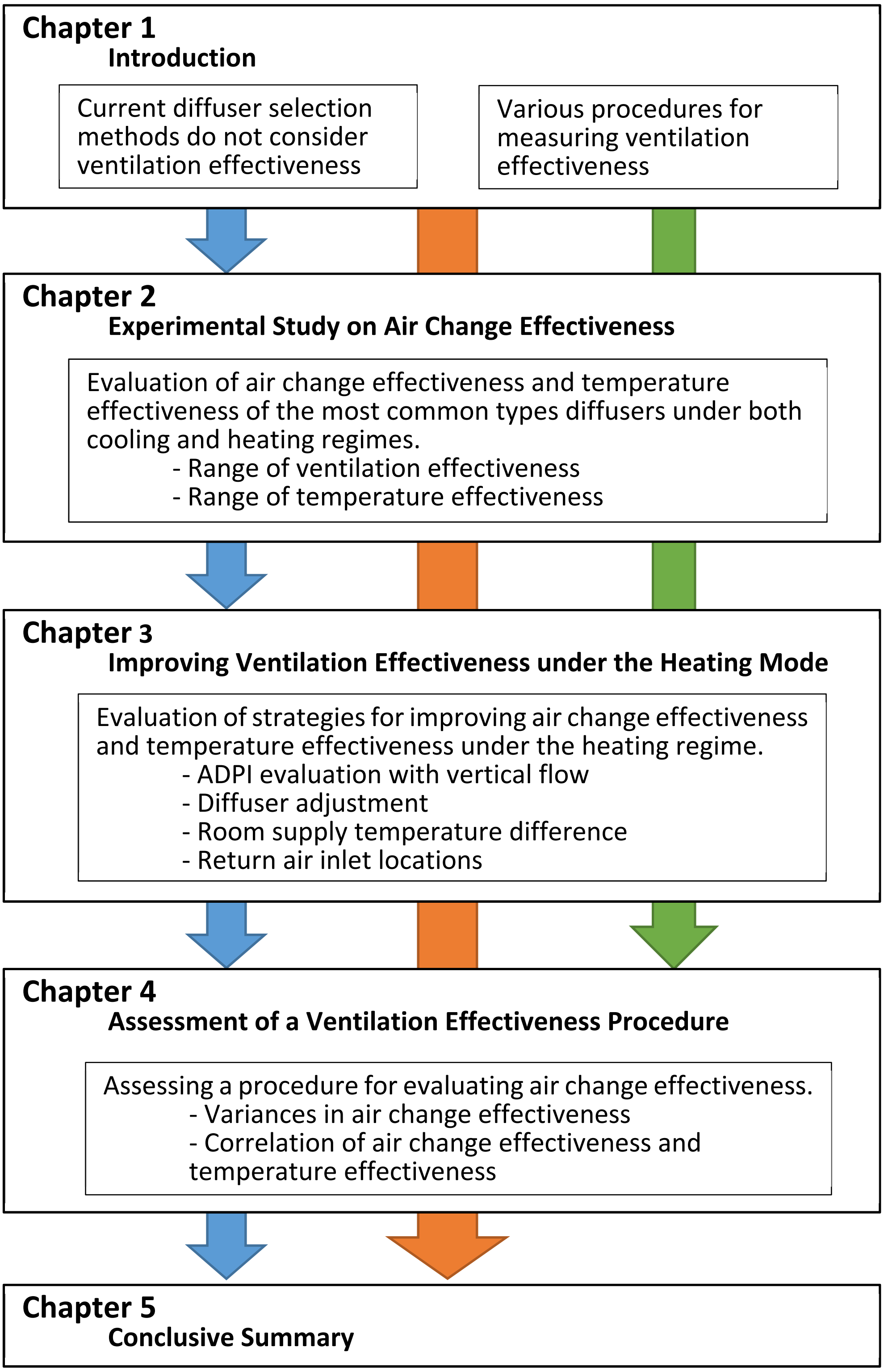


Figure 1-11 Structure of the research

Nomenclature

A_i	Age of air at a location *i* (ASHRAE)
$A_{i,corr}$	Corrected age of air
A_E	Age of air in exhaust air stream (ASHRAE)
ADPI	Air diffusion performance index
C_a	Average occupied zone pollutant concentration
C_e	Contaminant concentration in the exhaust
$C_{i,avg}$	Time-averaged tracer gas concentration at location *i* between t_{start} and t_{stop}
C_n.	Standardized occupied zone concentration (SHASE)
C_{mean}	Mean concentration of contaminant in the room
C_{out}	Pollutant concentration assuming perfect mixing
C_p	Supplied air pollutant concentration
$C(t)$	Decayed concentration measured at time t
C_0	Initial concentration of tracer gas at the time $t = 0$
DV	Displacement ventilation
EDT	Effective draft temperature
E	Air change effectiveness, arithmetic mean of E_i in occupied zone
E_i	Local air change effectiveness
E_{low}	Air Change effectiveness, arithmetic mean in low plane
E_{high}	Air change effectiveness, arithmetic mean in high plane
L	Characteristic length
L_p	Sound pressure level
PMV	Predicted mean vote
PPD	Predicted percentage of dissatisfied

$Q_{ex,m}$	Airflow rate in exhaust airstream
q_o	Minimum and maximum allowable flow rate
q_v	Ventilation flow rate
T_a	Spacious average temperature (EDT)
T_{EA}	Exhaust air temperature
T_i	Temperature at a location *i*
T_{SA}	Supply air temperature
$T_{0.25}$	Terminal velocity value of 0.25 m/s
$<T>_0$	Average temperature in occupied space
t_{stop}	Time of the final tracer gas measurement
t_{start}	Tracer injection is stopped at the beginning of tracer gas decay
V	Room air volume
V_i	Local air speed (EDT)
$\langle\bar{\tau}\rangle$	Room mean age of air
τ_n	Nominal time constant
$\bar{\tau}_\rho$	Local mean age of air at location *p* (REHVA)
$\langle\bar{\tau}_p\rangle$	Spatial average of the local mean ages of air
τ_r	Actual air change time.
$\langle\bar{\tau}_v\rangle$	Arithmetic average of the age of air measured at breathing level
ε^a	Air change efficiency
ε^c	Contaminant removal effectiveness
ε_p^a	Local air change index
ε_T	Temperature effectiveness

Φ	Measured trace gas concentration at steady state
Δp_t	Total pressure difference over the diffuser
ΔT_o	Maximum temperature difference between return air and supply air
ΔT	Exhaust air and supply air difference, T_{EA}-T_{SA}

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Chapter 2

Experimental Study on Air Change Effectiveness

2.1 Introduction

The use of the ADPI as diffuser selection method has been updated and expanded to heating by Liu et al. recently. However, this new updated cooling and heating ADPI concept considers only temperature uniformity and drafts caused by high velocity. The ADPI does not take into account the impact of thermal stratification and low ventilation effectiveness in all-air heating systems. This may be taken into account by just a single correction factor in ASHRAE Standard 62.1 (2010). Combining the ADPI with this correction factor does not always result in the optimal diffuser selection that considers both thermal comfort and ventilation effectiveness. For example, when the throw is too short, the jet may detach from the ceiling and increase draft risk under cooling conditions. Furthermore, a short throw length may cause inadequate mixing, resulting in a high temperature gradient and low air quality under heating conditions. It is necessary to have some momentum of flow to obtain adequate mixing in the occupied zone. However, a very large supply jet momentum may generate a draft when the flow rate is above a certain level. The temperature difference between return and supply jets should also be restricted, as a high-temperature difference may cause either a draft and/or a large vertical temperature stratification that results in inefficient energy use. Few studies have focused on ventilation effectiveness with mixing diffusers used for space heating, although all-air-heating ventilation is widely used (Muller et al. 2013). A comprehensive design process for diffuser selection and positioning that considers both thermal comfort and ventilation effectiveness at the same time is needed.

Therefore, the objective of this chapter is to provide data for guiding the design of air distribution systems for several of the most common types of ceiling diffusers when used for both the (1) cooling and (2) heating regimes. The studied diffuser types are as follows: linear slot diffuser, round ceiling diffuser, louvered face diffuser with no lip and perforated diffuser directional pattern (4-way). The study defines the operation range for diffuser selection when considering a high ADPI, ventilation effectiveness, and temperature effectiveness.

2.2 Methodology

The study utilized an experimental test room with various diffusers. Section 2.2.1 introduces the test chamber and diffusers used in the experiments described in this chapter. Section 2.2.2 explains the tracer gas decay test methodology. Then, the calibration of experiments, including the uncertainty in experiments, is examined in 2.2.3. Finally, Section 2.2.4 shows the experimental matrix.

2.2.1 Test chamber and tested diffusers

The study used experimental measurements taken in a full-scale test room with dimensions of 5.5 m × 4.5 m × 2.7 m and a sophisticated HVAC control system (Figure 2-1) at the University of Texas at Austin, USA. Detailed technical specifications for this indoor environment related research facility can be found in a previous publication (Liu et al. 2014). Figure 2-2 shows the chamber geometry and sensor positions. The experimental setup allowed different diffuser mounting positions: round ceiling, louvered face and perforated diffusers, and linear slot diffusers. Cooled panels with a total area of 10.8 m^2 covered one of the room walls, simulating a cold window surface in winter. The panels were connected to the dedicated chiller system, and the temperature of the panels was adjusted to simulate various heating loads. To simulate cooling loads, when the diffuser supplied cooled air, adjustable electric heaters were installed throughout the test room to mimic the cooling load of the room occupants, computers, lighting fixtures and floor heat patches due to transmitted solar radiation. The electric heaters were controlled to achieve the target supply and exhaust air temperature difference ($\Delta T = T_{SA} - T_{EA}$). The electric heaters were turned off under heating conditions, and the cooled panels were turned off under cooling conditions.

Figure 2-3 shows the four different tested diffusers, i.e., the linear slot diffuser (Price: model SDS75), round ceiling diffuser (Metalaire: model 3000-1), louvered face diffuser with no lip (Metalaire: model 5000-1), and perforated diffuser directional pattern (4-way) (Titus: model PCS). Linear slot diffusers and perforated diffuser directional pattern (4-way) can be made to provide various airflow patterns by adjusting their deflectors. The deflectors were adjusted to blowout horizontally for this experiment. For the linear slot diffusers, by-pass leakage was sealed (Figure 2a-2) to prevent air bypass, which is typical for slot diffuser (Liu et al. 2015), and achieve an ideal flow pattern.

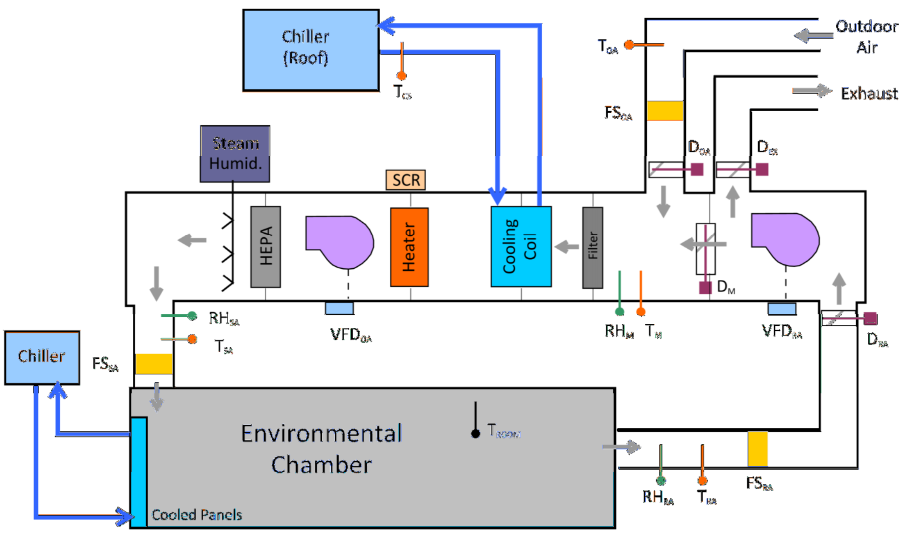


Figure 2-1 HVAC system for environmental control

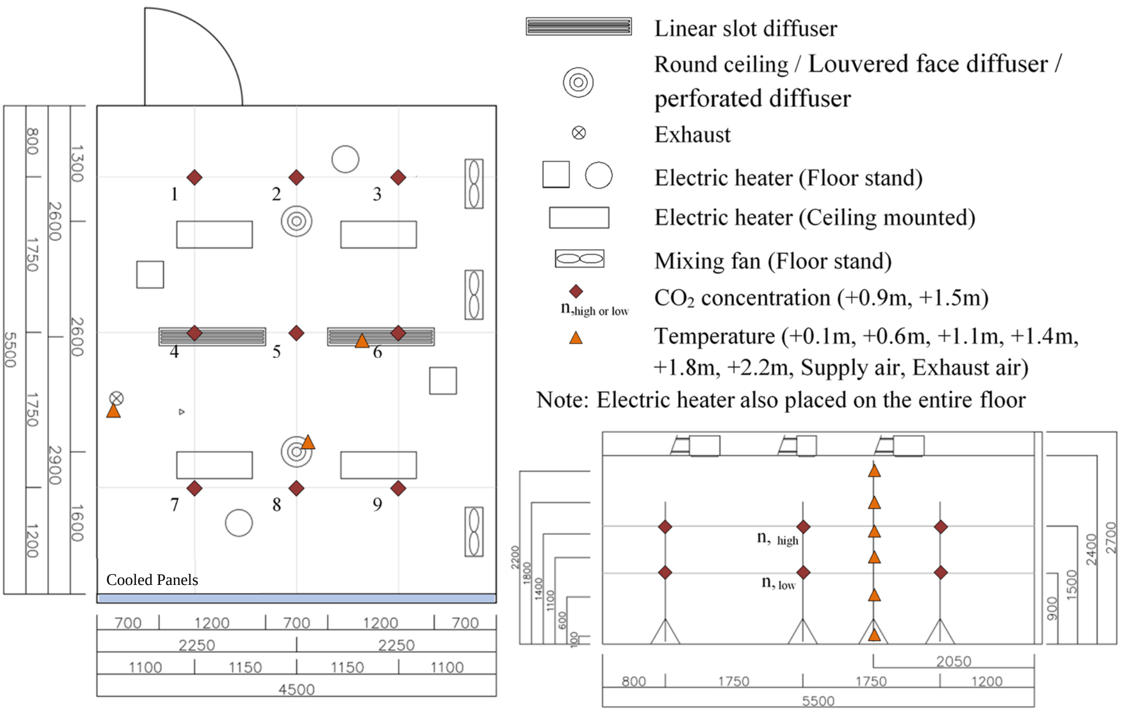


Figure 2-2 Experimental setup of the test room: the chamber geometry, diffuser locations, and sensor positions

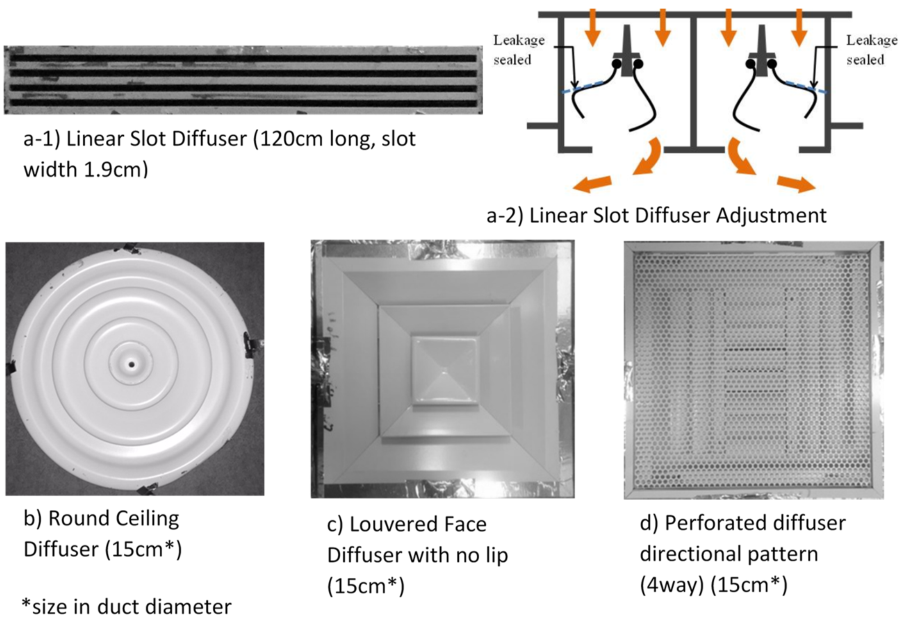


Figure 2-3 Tested diffuser types and flow adjustments

2.2.2 Tracer gas decay test

The tracer gas decay test using CO_2 as tracer gas was conducted to measure air change effectiveness. The decay of the CO_2 concentration was measured by in-situ sensors (TELAIR model 7001, range 0 to 4000ppm voltage output, Accuracy: ± 50 ppm or 5 %) at 18 locations simultaneously. The sensors were positioned across the room at two different horizontal planes, 0.9 m (low measuring plane) and 1.5 m (high measuring plane), above the floor (Figure 2-2 provides the exact sensor locations). The tracer gas was injected into the supply air duct and the gas spread though the target space. Three powerful mixing fans (Lasko Box Fan, Model # B20200, 55 cm × 11 cm × 57 cm) ensured the tracer gas mixed throughout the space while as it was released into the space. Tracer gas injection was stopped after the concentration of the gas in the space increased to around 2000 ppm. The mixing fans were stopped a little after the termination of tracer gas injection to make sure the remaining gas in the supply air duct flowed to the target space. The concentrations of the tracer gas at the measuring points were monitored during the experiment to confirm that the tracer gas was equally distributed and that all injected gas was supplied to the space before stopping the mixing fan. In addition, vertical temperature distributions at 0.1 m, 0.6 m, 1.1 m, 1.4 m, and 1.8 m ($T_{0.1}$, $T_{0.6}$, $T_{1.1}$, $T_{1.4}$ and $T_{1.8}$), and the supply and exhaust temperatures were measured. The vertical temperature distribution was simultaneously measured at one location. The Air change effectiveness and temperature effectiveness were utilized as indices of experiments described in this chapter. Refer to Chapter 1 section 1.2.2.3 and section 1.2.3 for more details about the indices.

2.2.3 Pre-tests – calibration and experimental set-up checks

Before conducting the set of full-scale experiments, the validity of the experimental procedure and the set-up for local age of air measurements were determined through a group of perfect mix tests and several repetitive experiments. In the perfect mixing test, the air change effectiveness and variances in the local air change effectiveness were analyzed. In addition, the uncertainty in the set of experiments was evaluated through the results of repetitive experiments and an uncertainty analysis.

2.2.3.1 Perfect mix tests

The perfect mix tests were conducted as control tests to provide reference results. A perfect mix was achieved by placing three additional mixing fans (Lasko Box Fan, Model# B20200, 55 cm × 11 cm × 57 cm) in the room, which ensured that air mixing would occur throughout the space (Figure 2-2), and the mixing fans were operated throughout the experiments. A total of eight cases with different air flow rates were studied. Table 2-1 shows the air change effectiveness (E), standard deviation (STDV) of local air change effectiveness (E_i), and temperature effectiveness ($\mathcal{E}_T$) in each experiment. V in the table indicates the air flow rate of exhaust calculated from the measured age of exhaust air for each experiment. The E value was close to 1.00 with a minimal standard deviation in all cases. $\mathcal{E}_T$ was also close to 1.0 for all experiments. These results reveal that the experiments were well controlled, and the local CO_2 decay measurement (needed for local age of air calculation) was valid.

Table 2-1 Results of the perfect mix test

Case	V [m^3/h]	E [-]	STDV of E_i [-]	ε_T [-]
P-1	69	0.98	0.01	1.02
P-2	152	0.98	0.02	1.02
P-3	221	1.00	0.02	1.02
P-4	301	0.98	0.02	1.02
P-5	381	1.01	0.02	1.00
P-6	390	0.98	0.03	1.01
P-7	474	1.00	0.04	1.02
P-8	566	0.97	0.04	1.00

2.2.3.2 Uncertainties in the measurements

The uncertainty in measurements of ventilation effectiveness depends on several factors such as an accuracy of the instruments used, air flow adjustment, pressure balancing, etc. In general, ASHRAE Standard 129 (2002) discusses various factors that cause significant measurement errors. The total uncertainty in the measured values of E was assumed to be approximately ±16%, and it mentioned that this can be considered as maximum uncertainty in the measured value. Cui et al. (2015) showed the uncertainty of the CO_2 tracer gas decay method for measuring air change was rate related to sensors and the calculation method. The uncertainties related to various in-situ CO_2 sensors were 5% for most of them, 5% for multi-points calculations, and 12% for two-point calculations. The total uncertainty also included the contribution of the experimental procedure and overall experimental set-up. The accuracy can be assessed by comparing repeated experiments. Table 2-2 shows the uncertainty in air change effectiveness calculations from seven sets of repeated experiments. The local air change effectiveness

(E_i), air change effectiveness at the low measuring plane (E_{low}), air change effectiveness at the high measuring plane (E_{high}), and the overall air change effectiveness (E) were compared. Differences between repeated experiments were computed as a percentage. All difference values in percentage per E_i, E_{low}, E_{high} and E were rearranged in ascending order, respectively. The minimum, 5th, 25th, 50th, 75th, and 95th percentile, maximum, and average values are shown in the table. The uncertainty of E_i was 6% on average and 14% in 95th percentile. Overall, the uncertainty of E was 6% on average and at most 11%. E_{low} and E_{high} had similar averages and maxima.

Table 2-2 Uncertainty in air change effectiveness

	Min.	5th	25th	50th	75th	95th	Max.	Ave.
E_i (N = 122)	0	1	3	5	9	14	19	6
E_{low} (N = 7)	0	-	-	7	-	-	12	6
E_{high} (N = 7)	1	-	-	4	-	-	11	5
E (N = 7)	0	-	-	4	-	-	11	6

* N = Number of the data compared

** #th in percentile

2.2.4 Experimental matrix

Table 2-3 shows the experimental conditions. The experiments were conducted under both heating and cooling conditions with four different diffusers, $T_{0.25}/L$, and internal loads. Each set of experiments had subsets of various $T_{0.25}/L$ values. For example, cases 1-9 had subsets of $T_{0.25}/L$ as 1.2 (case 1), 1.6 (case 2), and 1.8 (case 3).

The air change rates are also shown in parenthesis for reference. Overall, 85 experiments were conducted: 59 cases for heating and 26 cases for cooling. The internal loads (ΔT) indicate differences between supply air temperature (T_{SA}) and exhaust air temperature (T_{EA}). The cooled panel temperature mimicked a cooled window in the experiment where the diffuser operated in the heating mode (Figure 2-2). Furthermore, electric heaters associated with devices that mimicked internal heat gains (indoor occupants by cylinder heaters, computers by box heaters, lamps by ceiling heaters, and floor heat by floor heater respectively) were controlled to achieve the target ΔT. The electric heaters were turned off in heating conditions, and the cooled panels were turned off in cooling conditions. Mixing fans were used as CO_2 was injected into the test space, and they were stopped after sufficient CO_2 had been injected as described in section 2.2.2.

Table 2-3 Experimental conditions

Case #	Diffusers	$T_{0.25}/L$ [-] (Air change rate [h^{-1}])	Internal load ($\Delta T=T_{EA}-T_{SA}$) [°C]
Heating Conditions			
1–9	Linear slot diffusers (2 slots)	1.2 (1.1), 1.6 (2.1), 1.8 (3.0), 1.9 (3.3), 2.3 (4.5), 2.6 (5.8), 2.9 (6.9), 2.9 (7.2), 3.2 (8.6)	-5
10–17	Linear slot diffusers (4 slots)	0.9 (2.1), 1.1 (2.7), 1.4 (3.2), 1.9 (4.5), 2.2 (5.8), 2.5 (6.9), 2.5 (7.2), 2.7 (8.7)	-5
18–23	Linear slot diffusers (2 slots)	1.6 (2.1), 2.0 (3.3), 2.6 (5.7), 2.6 (5.8), 2.9 (7.2), 3.2 (8.6)	-2
24–29	Linear slot diffusers (4 slots)	0.9 (2.1), 1.4 (3.3), 1.9 (4.5), 2.2 (5.8), 2.5 (7.2), 2.7 (8.6)	-2
30–35	Round ceiling diffusers	0.7 (2.1), 1.0 (3.3), 1.2 (4.5), 1.4 (5.8), 1.7 (7.2), 2.0 (8.6)	-8
36–41	Round ceiling diffusers	0.7 (2.1), 1.0 (3.8), 1.2 (4.6), 1.4 (5.8), 1.7 (7.2), 2.0 (8.6)	-5
42–48	Round ceiling diffusers	0.7 (2.1), 0.7 (2.2), 0.9 (3.3), 1.2 (4.4), 1.4 (5.7), 1.7 (7.2), 2.0 (8.6)	-2
49–53	Louvered face diffusers without lip	1.8 (3.0), 2.0 (3.9), 2.1 (4.2), 2.6 (6.3), 3.5 (9.4)	-5
54–59	Perforated diffusers directional pattern (4-way)	0.8 (2.1), 1.4 (3.3), 1.8 (4.5), 2.2 (5.8), 2.4 (7.2), 2.7 (8.6)	-2
Cooling Conditions			
60–64	Linear slot diffusers (2 slots)	1.6 (2.1), 1.9 (3.3), 2.3 (4.5), 2.6 (5.8), 3.2 (8.6)	8
65–69	Linear slot diffusers (4 slots)	0.9 (2.1), 1.4 (3.3), 1.9 (4.5), 2.2 (5.8), 2.7 (8.6)	8
70–75	Round ceiling diffusers	0.8 (2.3), 1.0 (3.3), 1.1 (4.4), 1.4 (5.5), 1.9(8.3), 2.0 (8.6)	8
76–80	Louvered face diffusers without lip	1.7 (2.2), 1.9 (3.3), 2.1 (4.4), 2.5 (5.8), 3.3 (8.6)	8
81–85	Perforated diffusers directional pattern (4-way)	0.8 (2.1), 1.4 (3.3), 1.8 (4.5), 2.2 (5.8), 2.7 (8.6)	8

2.3. Results

This section examines the results of experiments in terms of the performance indices explained in section 2.2.2. The results are summarized for diffusers operating under (1) heating conditions and (2) cooling conditions.

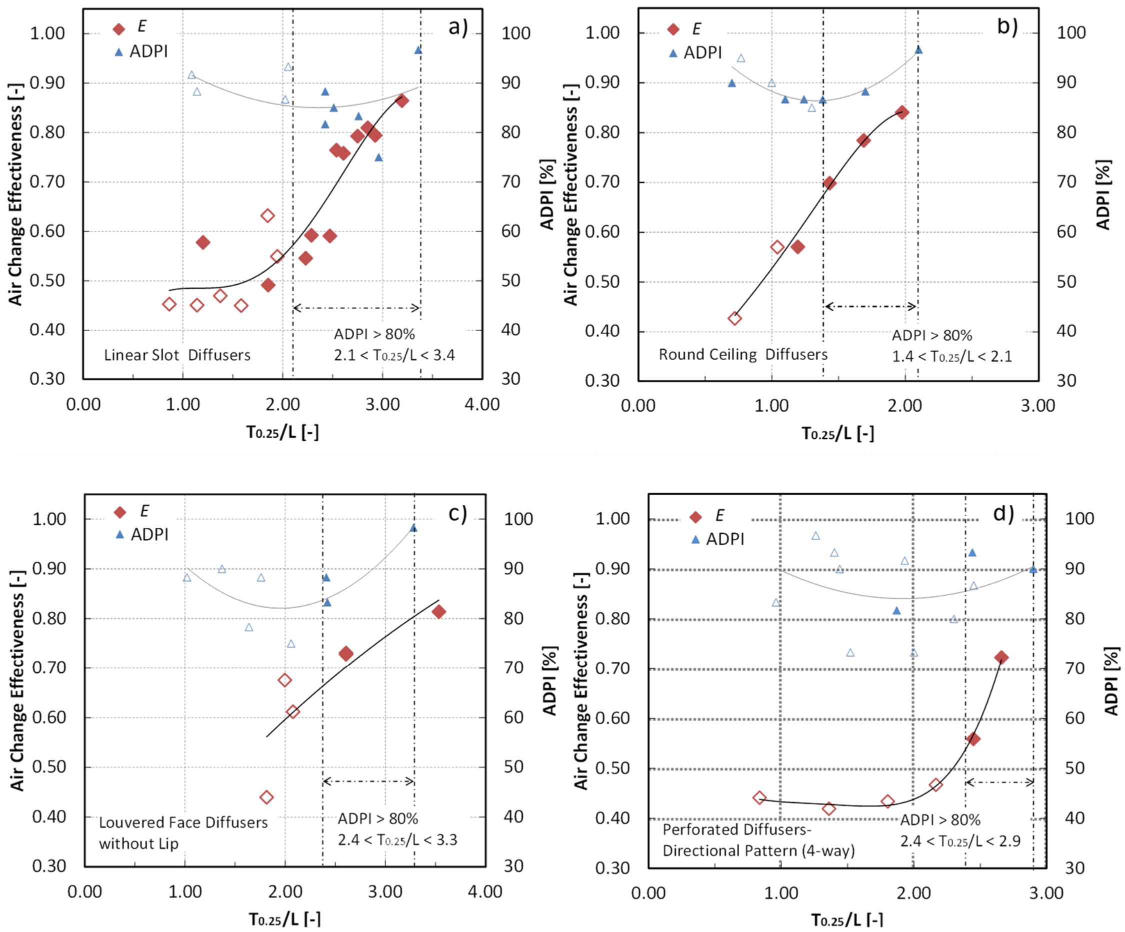
2.3.1 Heating experiments

This section shows the results related to the *E* and $\mathcal{E}_T$ under heating conditions as well as the influences of different internal loads on each diffuser.

2.3.1.1 Air change effectiveness and temperature effectiveness under heating conditions

Figure 2-4 shows the results of *E* under heating conditions with an internal load (room- supply temperature difference) of ΔT = -5 °C . Besides *E*, Figure 2-4 shows the ADPI value from the Liu and Novoselac study (2016) on the right side of the y-axis for reference. For the ADPI value, the unfilled marks indicate high thermal stratification, with a vertical temperature gradient higher than 3 °C per meter. As analyzed in Liu's study (2016), the highest ADPI was found for lower and higher $T_{0.25}/L$ values. The recommended range of $T_{0.25}/L$ with regards to the ADPI (ADPI higher than 80%, herein after the recommended range) under the heating mode excluded the high temperature gradient. A thorough detailed analysis and discussion of the results of the ADPI can be found in Liu's study (2015, 2016). The recommended range is also indicated in the graph. For the *E* value, the solid marks indicate that the T_{SA} values were at most 8 °C higher than the average occupied space temperature ($<T>_0$). The unfilled marks indicate that T_{SA} was at least 8 °C above $<T>_0$.

The results reveal that the value of E within the recommended range was from 0.56 to 0.87. A similar tendency was found among all diffusers that E significantly decreased when $T_{0.25}/L$ was smaller than the recommended range. At the same time, thermal stratification was also high in most cases as T_{SA} was 8 °C or higher than $<T>_0$. The smallest E was approximately 0.42 among all tested diffusers. Linear slot diffusers and perforated diffusers directional pattern (4way) had the lower E values at the minimum $T_{0.25}/L$ within the recommended range. Linear slot diffusers had the highest E at the maximum $T_{0.25}/L$ within recommended range. E sharply increased as $T_{0.25}/L$ increased. To maintain E around 0.8 as noted in ASHRAE Standard 62.1, $T_{0.25}/L$ should remain higher than 2.7 for linear slot diffusers, 1.7 for round ceiling diffusers, and 3.2 for louvered face diffuser without lips. Perforated diffusers directional pattern (4way) did not achieve E of 0.8, and the maximum air change effectiveness found was 0.72.



* For E) mark with fill: T_{SA} higher than 8 °C above $<T>_0$, mark without fill: T_{SA} higher than 8 °C or larger above $<T>_0$

** For ADPI): mark with fill: vertical temperature gradient lower than 3°C /m, mark without fill: higher than 3°C/m

Figure 2-4 $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and *ADPI* from Liu's (2016) experiments (right side y-axis) under heating conditions with ΔT = -5 °C. a) Linear slot diffusers: cases 1–17; b) round ceiling diffusers: cases 36–41, c) louvered face diffusers without lip: cases 49–53, d) perforated diffusers directional pattern (4-way): cases 54–59

Figure 2-5 shows the results of ε_T under heating conditions with ΔT = -5 °C. Overall, the value of ε_T within the recommended range was from 0.56 to 0.75. The range of ε_T was very similar between the linear slot, round ceiling, and louvered face diffusers. ε_T for the perforated diffuser was slightly smaller than those of three diffusers. Similar to the E, ε_T increased as $T_{0.25}/L$ increased within the recommended rage. However, ε_T slightly increased as $T_{0.25}/L$ decreased below the recommended value. The results indicate that when $T_{0.25}/L$ became smaller than the recommended range, the dominant factor that characterized the mixture of the space gradually changed from the supply jet from the diffuser to the down draft caused by the cold wall surface. This down draft caused high thermal stratification in the space. However, ΔT also slightly increased as $T_{0.25}/L$ decreased. As ε_T shows the ratio of T_{SA} - T_{EA} and T_{SA} - $\langle T \rangle_0$, the different ratio of the gradual increase of T_{SA} - T_{EA} and T_{SA} - $\langle T \rangle_0$ caused a slight increase in ε_T.

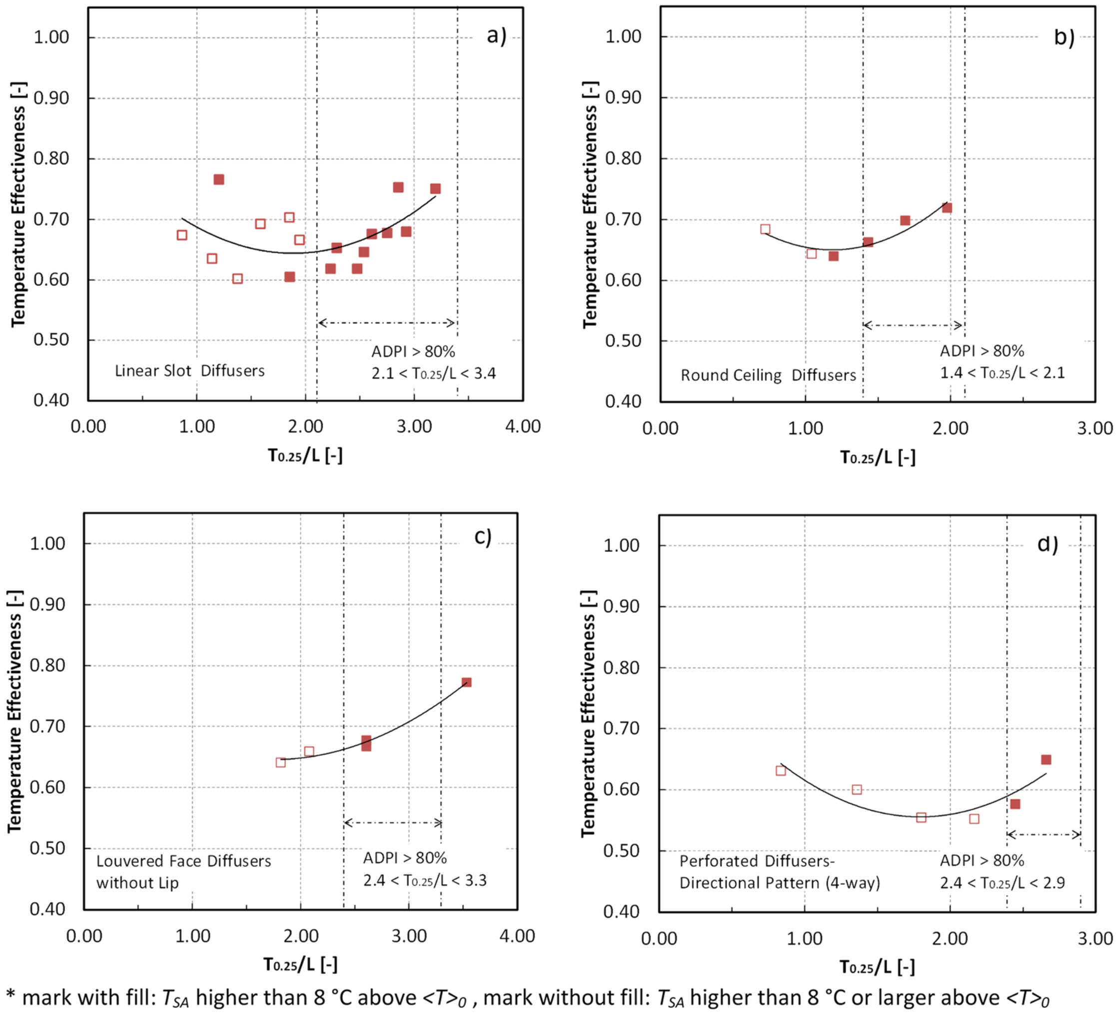


Figure 2-5 $T_{0.25}/L$ vs. Temperature effectiveness under heating conditions with ΔT = -5 °C. a) Linear slot diffusers: cases 1–17; b) round ceiling diffusers: cases 36–41; c) louvered face diffusers without lip: cases 49–53; d)perforated diffusers directional pattern (4-way): cases 54–59

2.3.1.2 Effect of internal load on air change effectiveness and temperature effectiveness

Figure 2-6 shows E and $\mathcal{E}_T$ of linear slot diffusers under the heating conditions with low and medium internal load that required room-supply temperature differences of-2 and -5 °C, respectively. For the same $T_{0.25}/L$, both E and $\mathcal{E}_T$ were higher under low load than under medium load. The largest values of the E were 0.87 and 1, and they achieved under the medium load and low load conditions. For the low load conditions, E was in the 0.65 to 1.00 range and $\mathcal{E}_T$ varied from 0.77 to 0.95 within the recommended range. The differences between the low and medium load conditions were minimal when $T_{0.25}/L$ was less than 1.5. A possible explanation for this is that, for cases where $T_{0.25}/L$ was less than 1.5, down draft from the cold wall surface became the dominant force, which did not make significant changes to E of the space. When $T_{0.25}/L$ was greater than 2.5, E was close to 1. However, $\mathcal{E}_T$ was approximately 0.8 when $T_{0.25}/L$ was 2.5 and it increased as $T_{0.25}/L$ exceeded 2.5. An increase in $\mathcal{E}_T$ was also observed in the low load condition when $T_{0.25}/L$ was between 1.5 to 2.5; the increase in $\mathcal{E}_T$ with higher $T_{0.25}/L$ for low load condition than in the medium load condition may be due to a smaller buoyancy effect that caused better mixing under the low load condition.

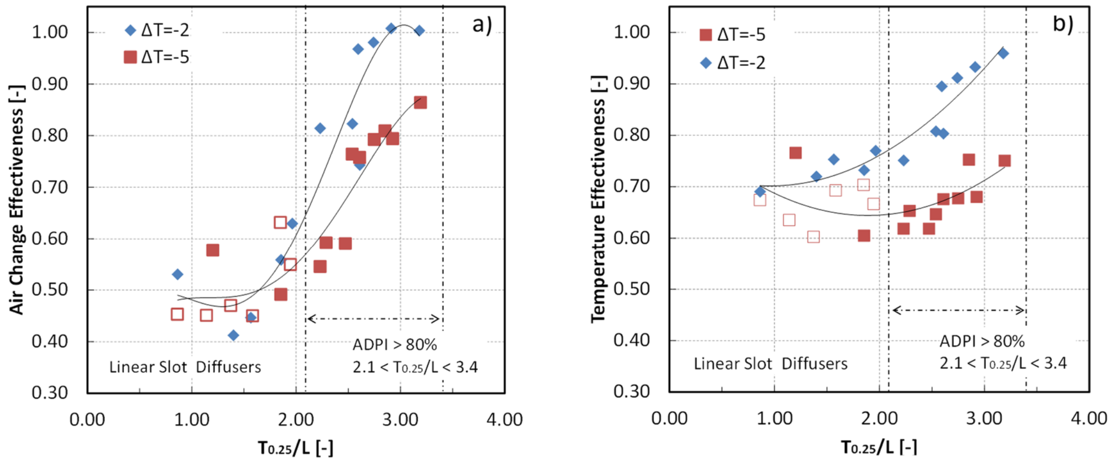
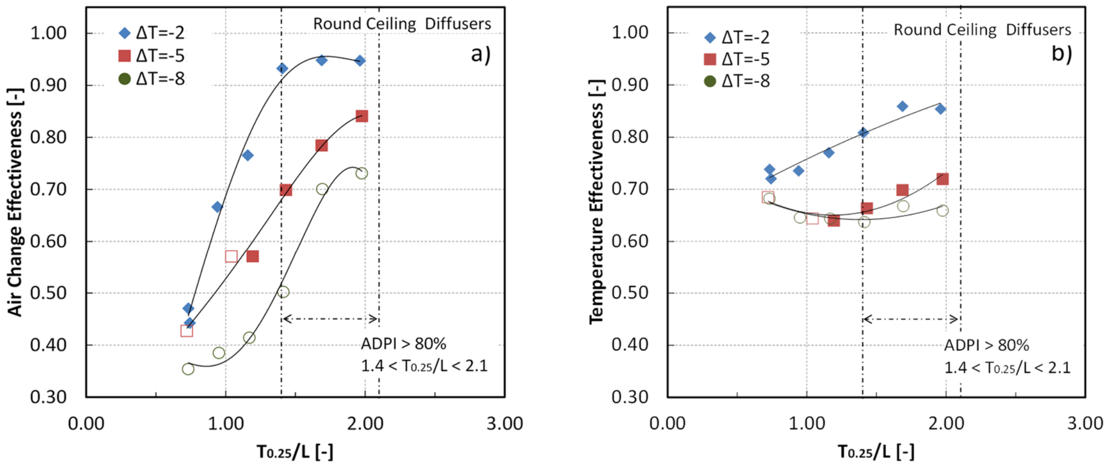


Figure 2-6 $T_{0.25}/L$ vs. air change effectiveness and temperature effectiveness of linear slot diffusers under heating conditions with different ΔT (cases 1–17 and cases 18–29). a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis), b) $T_{0.25}/L$ vs. temperature effectiveness

Figure 2-7 shows E and ε_T for round ceiling diffusers under heating conditions with low, medium, and high internal loads at room temperature differences of ΔT = -2, -5 and -8 °C, respectively. Both E and ε_T were higher under low load conditions than under medium load conditions and higher under medium load condition than under high load conditions for the same $T_{0.25}/L$ within the recommended range. The largest values of E were 0.95, 0.85 and 0.73 under low, medium, and high load conditions, respectively. Under low load conditions, E was near 1, which is within the recommended range. E sharply decreased when $T_{0.25}/L$ decreased such that it was outside the recommended range. ε_T slightly increased (from 0.80 to 0.86 within the recommended range) while E stayed close to 1. The lowest E was about 0.35 at a $T_{0.25}/L$ value of 0.6 under high load conditions. The difference between various ΔT values was minimal around a $T_{0.25}/L$ value of 0.6. It was interpreted as the transition point. $\mathcal{E}_T$ of the medium and high load conditions were almost same as when $T_{0.25}/L$ was smaller than the recommended range.



* mark with fill: T_{SA} higher than 8 °C above <T>$_0$, mark without fill: T_{SA} higher than 8 °C or larger above <T>$_0$

Figure 2-7 $T_{0.25}/L$ vs. Air change effectiveness and temperature effectiveness of round ceiling diffusers under heating conditions with different ΔT (cases 30–35, cases 36–41, and cases 42–48). a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis), b) $T_{0.25}/L$ vs. temperature effectiveness

2.3.2 Air change effectiveness and temperature effectiveness under cooling conditions

Figure 2-8 shows the results of *E* under cooling conditions with internal loads of ΔT = 8 °C. The right side of the y axis with the triangle plots show ADPI values from Liu's study (2016) for reference. The recommended range is also indicated on the graph. The results revealed that *E* was in the range of 0.98 to 1.16 within the recommended range. *E* was greater than or equal to 1 under all cooling conditions. *E* was slightly larger when $T_{0.25}/L$ decreased. This might be because the jet from the diffuser detaches from the ceiling and dives into the occupant zone when $T_{0.25}/L$ is small. As E_i was measured in an occupant zone, this jet detachment affect produced slightly higher *E* values. However, it should be noted that a short throw length tends to decrease ADPI by increasing draft risk.

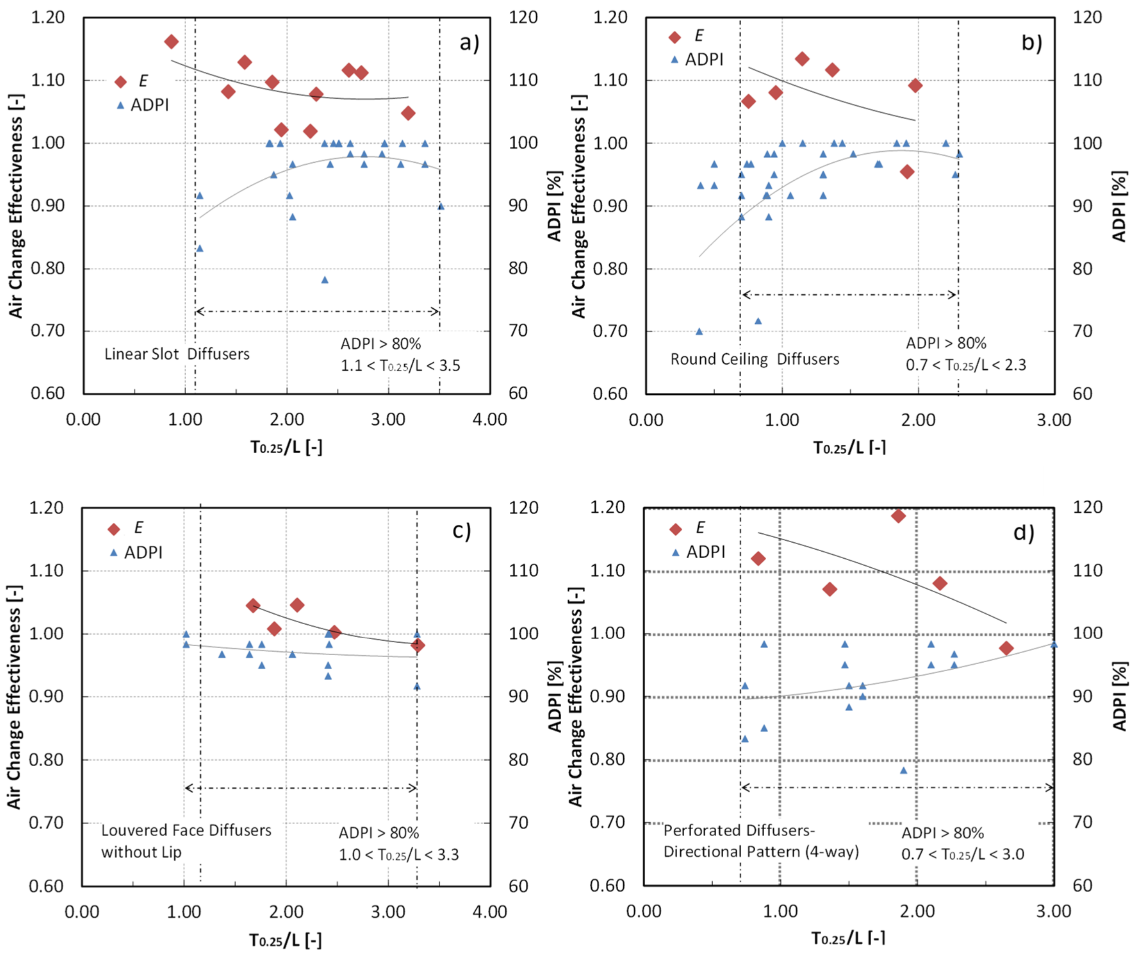


Figure 2-8 T0.25/L vs. air change effectiveness (left side y-axis) and ADPI from Liu's (2016) experiments (right side y-axis) under cooling conditions with ΔT = 8 °C. a) Linear slot diffusers: cases 60–69, b) round ceiling diffusers: cases 70–75, c) louvered face diffusers without lip: cases 76–80, d) perforated diffusers directional pattern (4-way): cases 81–85

Figure 2-9 displays the results of $\mathcal{E}_T$ under cooling conditions with an internal load of $\Delta T = 8$ °C. Overall, $\mathcal{E}_T$ was in the range of 0.92 to 1.11 within the recommended range. Similar to E, $\mathcal{E}_T$ was slightly higher when $T_{0.25}/L$ was small, and it decreased as

$T_{0.25}/L$ increased. The results for round ceiling diffusers show the smallest slope among the four tested diffusers. ε_T was close to 1 for every measured $T_{0.25}/L$.

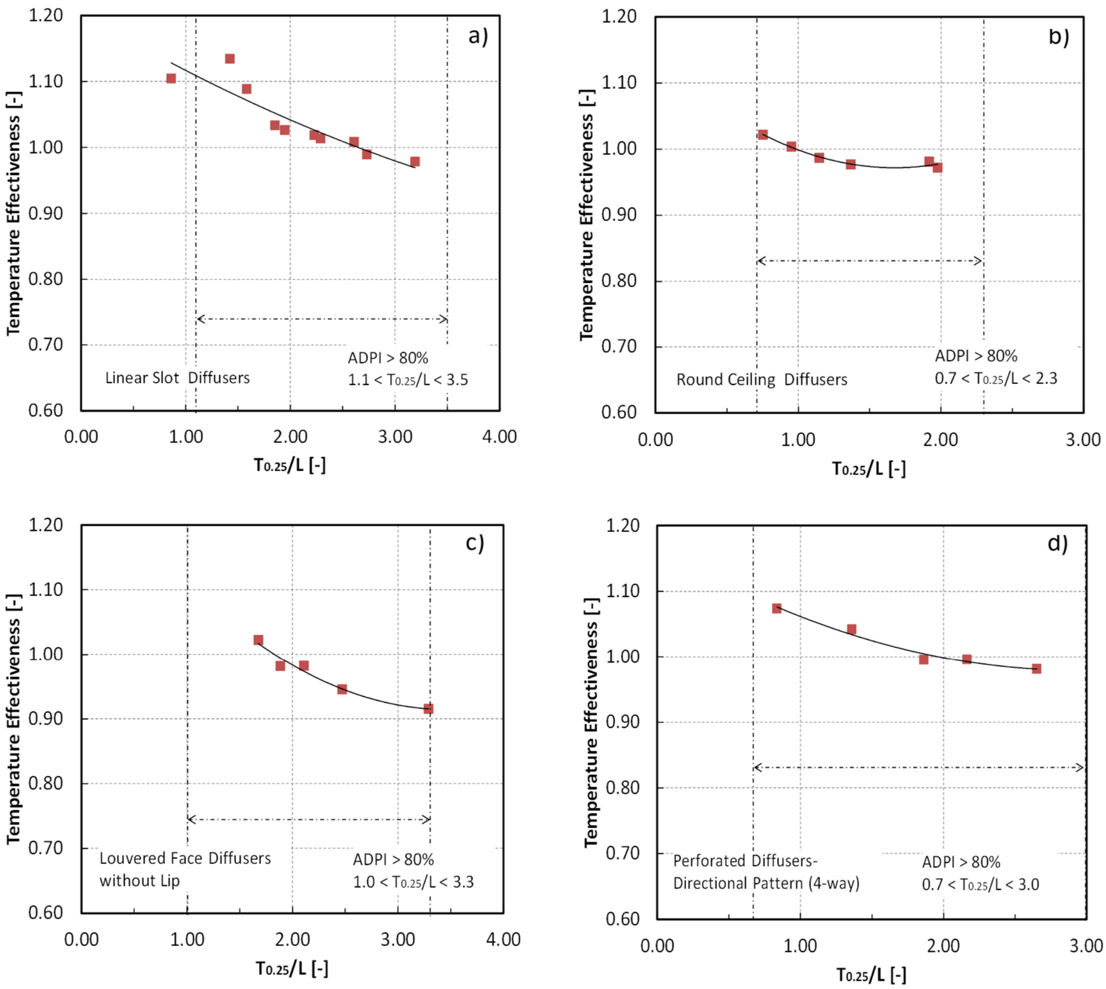


Figure 2-9 $T_{0.25}/L$ vs temperature effectiveness under cooling conditions with ΔT = 8 °C. a) Linear slot diffusers: cases 60–69, b) round ceiling diffusers: cases 70–75, c) louvered face diffusers without a lip: cases 76–80, d) perforated diffusers directional pattern (4-way): Cases 81–85

2.4 Discussion

Tables 2-4 and 2-5 provide the ranges of E and $\mathcal{E}_T$ within the recommended range of $T_{0.25}/L$ in terms of ADPI for the heating mode and cooling mode. The tables are intended to to provide HVAC designers with guidance for selecting diffusers and air flow rates for air systems that provide both heating and cooling. The data shown for heating in the table are from the cases in which the typical internal load is ΔT = -5 °C (medium loads condition). The range of loads [W/m^2] for E and $\mathcal{E}_T$ within the recommended range of $T_{0.25}/L$ were calculated from ΔT and the air flow rate of the exhaust air stream. The experiments in this study used a slightly different range of internal loads than the previous experiments for the ADPI study (Liu et al. 2016). This is because the experiments aimed to control room supply temperature difference (ΔT) to examine effects of thermal stratification on the E. This target ΔT produced slightly lower loads than those used in the previous ADPI study. However, the results between ADPI, E, and $\mathcal{E}_T$ in Table 2-4 are still comparable, because the recommended range of ADPI is valid for smaller loads (smaller that 30–40 W/m^2, indicated in Table 2-4) as the ADPI only increases with a decrease in the thermal loads.

The results in Tables 2-4 and 2-5 reveal that all analyzed diffusers have similar performance when considering loads and $T_{0.25}/L$. Both E and $\mathcal{E}_T$ increased as $T_{0.25}/L$ increased under heating conditions and slightly decreased as $T_{0.25}/L$ increased under cooling conditions. The results also revealed that it is possible for E to decreases to less than 0.8, as mentioned in SHRAE Standard 62.1 (2010), even when T_{SA} is at most 8 °C higher than $\langle T \rangle_0$. A wider range of $T_{0.25}/L$ was allowed under cooling conditions than heating conditions, meaning that a smaller range of $T_{0.25}/L$ value is accepted under heating conditions than would be accepted under cooling conditions. However, the air change effectiveness may significantly decrease when $T_{0.25}/L$ is small under heating conditions. The perforated diffuser directional pattern (4-way) showed slightly lower E and ε_T under heating conditions. The linear slot diffusers had a greater range for air change effectiveness under heating conditions. For cooling condition, E and ε_T were close to or higher than 1 (within recommended range) regardless of the diffuser type.

Mixing ventilation systems are utilized in various HVAC systems such as a valuable air volume (VAV) system with constant supply temperature or constant air volume (CAV) system with variable supply air temperature. In most buildings, the same all-air system is used for both heating and cooling, and the cooling load is the dominant factor for sizing the coils, fans, ducts and diffusers; consequently, the diffusers are usually selected with consideration of only the cooling mode. However, the range capable of achieving good mixing under the heating condition is not as wide as that in the cooling mode. This chapter provides fundamental data on diffuser performance under both heating and cooling conditions. HVAC system designers should carefully select not only the diffusers but also an air flow rate, a supply air temperature, and a control sequence that can achieve better air change effectiveness and thermal comfort with optimal use of energy under both the cooling and heating modes.

Table 2-4 Ranges of air change effectiveness and temperature effectiveness within the range of recommended $T_{0.25}/L$ in terms of ADPI (Heating)

Diffuser type	Recommended range of $T_{0.25}/L$ regarding ADPI		Range of E and $\mathcal{E}_T$ within recommended $T_{0.25}/L$ regarding ADPI		
	$T_{0.25}$ / L [-]	Loads [W/m^2]	E [-]	$\mathcal{E}_T$ [-]	Loads [W/m^2]
Linear slot diffusers	2.1–3.4	30–40	0.57–0.87	0.65–0.75	20–31
Round ceiling diffusers	1.4–2.1	30–40	0.68–0.85	0.66–0.72	24-30
Louvered face diffusers without lip	2.4–3.3	30–40	0.66–0.81	0.65–0.74	26–33
Perforated diffusers directional pattern (4-way)	2.4–2.9	30–40	0.56–0.72	0.58–0.65	27–30

Table 2-5 Ranges of air change effectiveness and temperature effectiveness within the range of recommended $T_{0.25}/L$ in terms of ADPI (Cooling)

Diffuser type	Recommended range of $T_{0.25}/L$ regarding ADPI		Range of E and $\mathcal{E}_T$ within recommended $T_{0.25}/L$ regarding ADPI		
	$T_{0.25}$ / L [-]	Loads [W/m^2]	E [-]	$\mathcal{E}_T$ [-]	Loads [W/m^2]
Linear slot diffusers	1.1–3.5	25–50	1.12–1.05	1.11–0.98	14–69
Round ceiling diffusers	0.5–2.3	25–50	1.08–1.03	1.03–0.97	17–66
Louvered face diffuser with no lip	1.0–3.3	25–50	1.05–0.98	1.03–0.92	17–62
Perforated diffuser directional pattern (4way)	0.7–3.0	25–50	1.16–0.98	1.07–0.98	17–63

2.5. Conclusion

Experimental measurements in a full-scale test room were conducted with various types of diffusers, air flow rates, and internal loads to evaluate the air change effectiveness and the temperature effectiveness. These experiments were conducted on both heating and cooling cycles, though there was an emphasis on the heating mode. All diffusers tested showed similar results, although each diffuser had a unique shape. Under the heating mode, the ranges of air change effectiveness E and temperature effectiveness ε_T were 0.56 to 0.87 and 0.58 to 0.75, respectively, falling within the recommended range of $T_{0.25}/L$ with regard to the ADPI. A significant decrease in E was found to occur when $T_{0.25}/L$ was small. Both E and ε_T increased as ΔT became close to isothermal flow. Under the cooling mode, the ranges of E and ε_T were 0.98 to 1.12 and 0.92 to 1.11, respectively, falling within the recommended range. Relatively good mixing was found under cooling conditions.

The studies provided fundamental diffuser performance data that considers both thermal comfort and ventilation effectiveness. The range capable of achieving good mixing under the heating condition was significantly smaller than the range for the cooling mode. Not just diffusers, but also factors such as air flow rate and supply air temperature should be carefully designed in all-air heating and cooling systems in order to achieve good mixing and thermal comfort.

Nomenclature

ADPI	Air Diffusion Performance Index
$T_{0.25}$	Terminal Velocity Value of 0.25 m/s
L	Characteristic Length
E	Air Change Effectiveness, Arithmetic Mean of E_i in Occupied Zone
E_i	Local Air Change Effectiveness
E_{low}	Air Change Effectiveness, Arithmetic Mean in low plane
E_{high}	Air Change Effectiveness, Arithmetic Mean in high plane
ε_T	Temperature Effectiveness
T_{SA}	Supply Air Temperature
T_{EA}	Exhaust Air Temperature
T_i	Temperature at a Location *i*
$<T>_0$	Average Temperature in Occupied Space
ΔT	Exhaust Air and Supply Air Difference, T_{EA}-T_{SA}
STDV	Standard deviation
VAV	Valuable Air Volume
CAV	Constant Air Volume

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Chapter 3

Improving Ventilation Effectiveness under the Heating Mode

3.1 Introduction

In Chapter 2, extensive experiments on air change effectiveness (*E*) in mixing ventilation were conducted. These data combined with the results from the recent ADPI study (Liu et al., 2016) provided comprehensive data set on diffuser performance considering both the uniformity of the temperature field and range of *E* in the both the cooling and heating applications. *E* and temperature effectiveness ($\mathcal{E}_T$) were slightly higher than 1.0 in the cooling applications. However, in the heating application, *E* significantly decreased at small $T_{0.25}/L$ even though ADPI was within the acceptable range (ADPI higher than 80%); this low *E* value is due to the short circuit of the supply air. *E* within the acceptable ADPI range of $T_{0.25}/L$ (recommended range) was 0.56 to 0.87. This short circuiting of hot air puts the performances of many diffusers in a range that is lower than the specified value in ASHRAE standard 62.1, $E > 0.8$.

To overcome the challenges of poor ventilation under heating conditions, this chapter examines simple strategies that may improve *E* and $\mathcal{E}_T$ under heating conditions, while maintaining an acceptable ADPI; specifically, it evaluates the impacts of diffuser deflector adjustment in linear slot diffusers and adjustable blades grills, room-supply air temperature difference of linear slot diffusers with vertical flow, and exhaust locations of adjustable blade grills. In addition, it provides new diffuser selection data in the form of the ADPI for diffusers with a vertical jet projection (vertical flow) categorized as Group E in the ASHRAE Handbook (2013).

3.2 Methodologies

The first part of the methodology section describes the diffusers tested in this chapter. The second part explains the two series of experiments that were conducted: (1)

experiments related to ADPI and (2) experiments related to E and $\mathcal{E}_T$. These two series of experiments were conducted in the test room described in Chapter 2. The testing methodology for the ADPI experiments followed the procedure in Liu's study and the Chapter 2 procedure for E and $\mathcal{E}_T$ experiments. The testing methodologies followed previous study procedures described in greater detail in (Liu et al, 2015, 2016, 2017) and Chapter 2.

3.2.1 Tested diffusers

Figure 3-1a shows the linear slot diffusers (Model SDS75, frame size 190 mm×1,200 mm, Price Industries, Inc.) with vertical flow used in the experiments. The diffusers provide various airflow patterns when their deflectors are adjusted. The same diffuser used in Chapter 2 was utilized; however the deflector positions were different. The performance of linear slot diffusers with horizontal projection can be found in Chpater 2. The vertical flow allows supply of primary air directly to the occupied space and may increase E and $\mathcal{E}_T$. However, a higher air velocity in the occupied space may compromise the ADPI.

Figure 3-1b shows the adjustable blade diffusers (Model 51DV, frame size: 150 mm × 600 mm, Nailor HVAC, Inc.) used as high side wall diffusers. This specific model is a good representative of all high side wall adjustable blade diffusers (Liu et al., 2016). By adjusting their blade angles, the diffusers can permit different airflow directions. To evaluate the impact of adjustments on E and $\mathcal{E}_T$, blades angle were set as follows: 0° horizontal, 45° upward, and 45° downward. The ADPI with each adjustments was determined from previous studies (Liu et al., 2016). In previous studies, the 45°upward projection allowed supply air to easily attch to the ceiling and slide along the ceiling owing to the Coanda effect, resulting in a better ADPI under cooling conditions (Liu et al., 2016). However, it may cause higher thermal stratifications and lower *E* under heating conditions. The 45° downward projection directly supplies air to the occupied zone, which may increase *E*. However, it may also cause a lower ADPI because of higher velocity in the occupied space. The 0° horizontal blade position was considered as the nominal setting, and different exhaust locations were tested with the nominal setting.

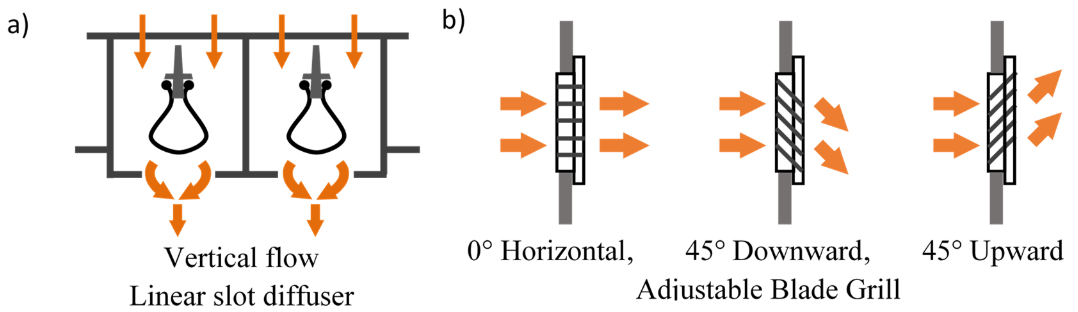


Figure 3-1 Diffusers used in this study: a) Linear slot diffuser with vertical flow, b) Adjustable blade grill with 0° horizontal, 45° upward, and 45° downward adjustment

3.2.2 ADPI measurements

The ADPI measurements were conducted in a test chamber located at the Center for Energy and Environmental Resources at the University of Texas at Austin, USA with the size of 5.5 m × 4.5 m × 2.7 m; it was the same test chamber described in Chapter 2. Figure 3-2 illustrates experimental setups of the test room for ADPI measurements. ADPI measurements were conducted under the same chamber and setups as our previous studies (Liu et al., 2015, 2016, 2017). The heating load was simulated by adjusting the temperature of the cooled wall connected to a dedicated chiller, which together mimicked an exterior wall or window in winter conditions.

Although particle image velocimetry (PIV) measurements (Cao et al., 2014) can be used to better visualize the airflow fields, velocity and temperature at 60 locations in the occupied zone at four different heights above the floor (0.1 m, 0.6 m, 1.1 m, and 1.7 m) were measured with twelve hot-sphere anemometers (HT-400, SENSOR, Poland, accuracy: ±0.03 m/s+-3 %, temperature: ±0.2 °C). The measurements were repeated five times for each experiment to obtain 60 locations with the twelve available sensors. At the same time, the vertical temperature (0.1 m, 0.6 m, 1.1 m, 1.4 m, 1.8 m, 2.2 m) was measured at five different locations by thermistors (Model 44033, OMEGA, Accuracy: ±0.1 °C). Furthermore, supply and exhaust air temperatures were monitored during the experiments to ensure the stability of the chamber conditions.

The uncertainty of an ADPI measurement depends on the accuracy of the instruments used and several factors related to the experimental setup; the detailed procedure for calculating this uncertainty is described in our previous study (Liu et al., 205). Owing to the high accuracy of the velocity and temperature sensors in the studies, the uncertainty was ±2.7 % for the absolute value of the ADPI.

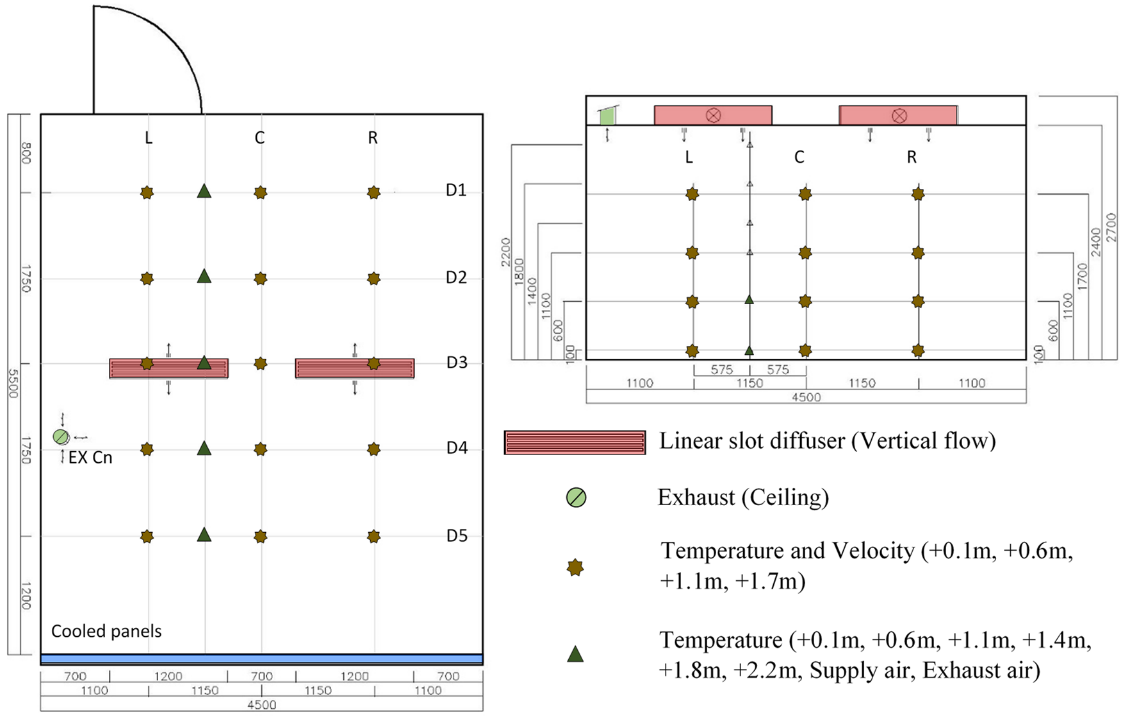


Figure 3-2 Experimental setup of a test room for ADPI measurements. (Chamber geometry, temperature, and velocity measurement locations)

3.2.2.1 Experimental matrix for ADPI measurements

Table 3-1 shows the experimental conditions for the ADPI measurements. Linear slot diffusers with two and four slots with vertical flow were tested. Each setup consisted of several cases with various $T_{0.25}/L$ (related to air change rate). For instance, cases (A.1–6) had the same experimental setup but different $T_{0.25}/L$ values, such as 0.5 in case A.1 and 0.8 in case A.2. The $T_{0.25}$ value was derived as described in Chapter 1 section 1.2.4. All experiments were conducted under heating conditions.

Table 3-1 Experimental conditions for the ADPI measurements

Case #	Supply diffusers		$T_{0.25}/L$ [-] (Air change rate [h^{-1}])	Temperature difference between supply and exhaust $\Delta T = T_{EA} - T_{SA}$ [°C]	Return air inlet location
	Type	Adjustment			
A.1–6	Linear slot diffuser (2 slots)	Vertical	0.5 (2.1), 0.8 (3.3), 1.1 (4.5), 1.4 (5.8), 1.7 (7.2), 2.0 (8.6)	-5	EX Cn
A.7–12	Linear slot diffuser (4 slots)		0.4 (2.1), 0.7 (3.3), 0.9 (4.5), 1.2 (5.8), 1.4 (7.2), 1.8 (8.6)	-5	
A.13–18	Linear slot diffuser (2 slots)		0.6 (2.1), 1.0 (3.3), 1.3 (4.5), 1.6 (5.8), 1.9 (7.2), 2.2 (8.6)	-2	
A.19–24	Linear slot diffuser (4 slots)		0.5 (2.1), 0.8(3.3), 1.1 (4.5), 1.4 (5.8), 1.7 (7.2), 1.9 (8.6)	-2	

3.2.3 Air change and temperature effectiveness measurements

Figure 3-3 illustrates the experimental setup for the E and $\mathcal{E}_T$ measurements. The experimental setup included adjustable blade diffusers with high side wall positions and 2-slot and 4-slot linear slot diffusers with vertical flow. The dimensions of the plenum box for an adjustable blade diffuser are also described in Figure 3-3. Furthermore, Figure 3-3 shows the five specific exhaust locations used to evaluate the impact of the exhaust locations. There were three locations for ceiling mounted positions, EX Cd, EX Cn, and EX Cw, and two locations near floor, EX Fd and EX Fw.

The tracer gas decay test using CO_2 as the tracer gas was conducted to measure Ei and calculate E. The measuring point and procedure described in Chapter 2 was utilized. The CO_2 concentration was measured at 18 locations simultaneously and vertical temperature distributions and supply and exhaust temperatures were measured. The uncertainty in the measurements of E has also been discussed in Chapter 2. From repetitive experiments, the uncertainty in the local *Ei* was 6% on average with a maximum of 14% and that in the *E* measurement was 6% on average with a maximum of 11 %.

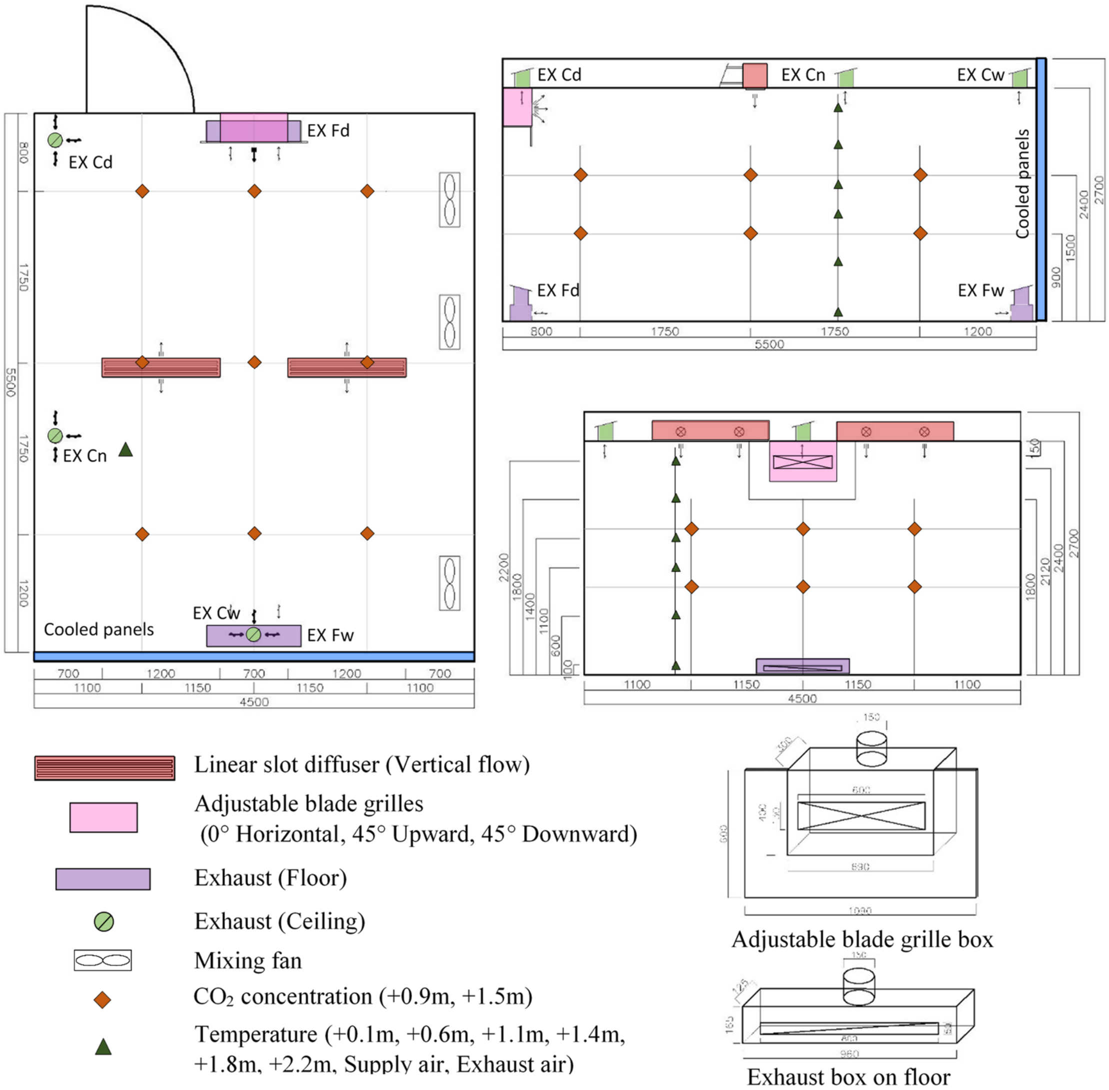


Figure 3-3 Experimental setup of a test room for Air change effectiveness and Temperature effectiveness measurements (Chamber, adjustable blade grille box and exhaust box geometry, temperature and CO_2 concentration measurement locations)

3.2.3.1 Experimental matrix for air change and temperature effectiveness measurements

Table 3-2 shows the experimental conditions for the E and $\mathcal{E}_T$ measurements. The experiments were conducted with 2-slot and 4-slot linear slot diffusers with vertical flow for low and medium internal loads (room-supply temperature difference of ΔT = -2 °C and ΔT = -5 °C, respectively), and adjustable blade diffusers with three different adjustments under heating conditions. In addition, five different exhaust locations with nominal settings (0° Horizontal) of adjustable blade diffusers were tested.

Table 3-2 Experimental conditions for air change effectiveness and temperature effectiveness measurements

Case #	Supply Diffusers		$T_{0.25}/L$ [-] (Air change rate [h^{-1}])	Temperature difference between supply and exhaust ($\Delta T=T_{EA}-T_{SA}$) [°C]	Return air Inlet Location
	Type	Adjustment			
B.1–8	Linear slot diffuser (2 slots)	Vertical	0.5 (2.1), 0.7 (3.0), 0.8 (3.3), 1.1 (4.4), 1.5 (5.8), 1.7 (6.9), 1.7 (7.2), 2.0 (8.6)	-5	EX Cn
B.9–17	Linear slot diffuser (4 slots)	Vertical	0.4 (2.1), 0.6 (3.0), 0.7 (3.3), 0.9 (4.4), 0.9 (4.5), 1.2 (5.8), 1.4 (6.9), 1.5 (7.2), 1.8 (8.6)	-5	EX Cn
B.18–23	Linear slot diffuser (2 slots)	Vertical	0.6 (2.1), 1.0 (3.3), 1.3 (4.5), 1.6 (5.8), 1.9 (7.2), 2.2 (8.6)	-2	EX Cn
B.24–29	Linear slot diffuser (4 slots)	Vertical	0.5 (2.1), 0.8(3.3), 1.1 (4.5), 1.4 (5.8), 1.7 (7.2), 2.0 (8.6)	-2	EX Cn
B.30–36	Adjustable Blade Grill	0° Horizontal	0.8 (1.6), 0.9(2.1), 1.2 (3.3), 1.4 (4.5), 1.6 (5.8), 1.9 (7.2), 2.1 (8.6)	-5	EX Cn
B.37–43	Adjustable blade grill	45° upward	0.3 (1.6), 0.4(2.1), 0.5 (3.3), 0.7 (4.5), 0.8 (5.8), 0.8 (7.2), 0.9 (8.6)	-5	EX Cn
B.44–50	Adjustable blade grill	45° downward	0.3 (1.6), 0.4(2.1), 0.5 (3.3), 0.7 (4.5), 0.8 (5.8), 0.8 (7.2), 0.9 (8.6)	-5	EX Cn
B.51–57	Adjustable blade grill	0° horizontal	0.8 (1.6), 0.9(2.1), 1.2 (3.3), 1.4 (4.5), 1.6 (5.8), 1.9 (7.2), 2.1 (8.6)	-5	EX Cw
B.58–63	Adjustable blade grill	0° horizontal	0.8 (1.6), 0.9(2.1), 1.2 (3.3), 1.4 (4.5), 1.6 (5.8), 1.9 (7.2), 2.1 (8.6)	-5	EX Cd
B.64–71	Adjustable blade grill	0° horizontal	0.8 (1.6), 0.9(2.1), 1.2 (3.3), 1.4 (4.5), 1.6 (5.8), 1.9 (7.2), 2.1 (8.6)	-5	EX Fw
B.72–78	Adjustable blade grill	0° horizontal	0.8 (1.6), 0.9(2.1), 1.2 (3.3), 1.4 (4.5), 1.6 (5.8), 1.9 (7.2), 2.1 (8.6)	-5	EX Fd

3.3. Results

This section is divided presents the results from two series of experiments: 1) experiments with linear slot diffusers with vertical flow and 2) experiments with adjustable blade diffusers with different deflector angles and exhaust locations.

3.3.1 Vertical flow with linear slot diffusers

The first part examines the range of $T_{0.25}/L$ that can achieve an ADPI higher than 80% and the associated E and $\mathcal{E}_T$. Then, the ADPI, E and $\mathcal{E}_T$ results with a lower ΔT are discussed. Finally, the section examines some results of velocity and temperature measurements from the ADPI experiments.

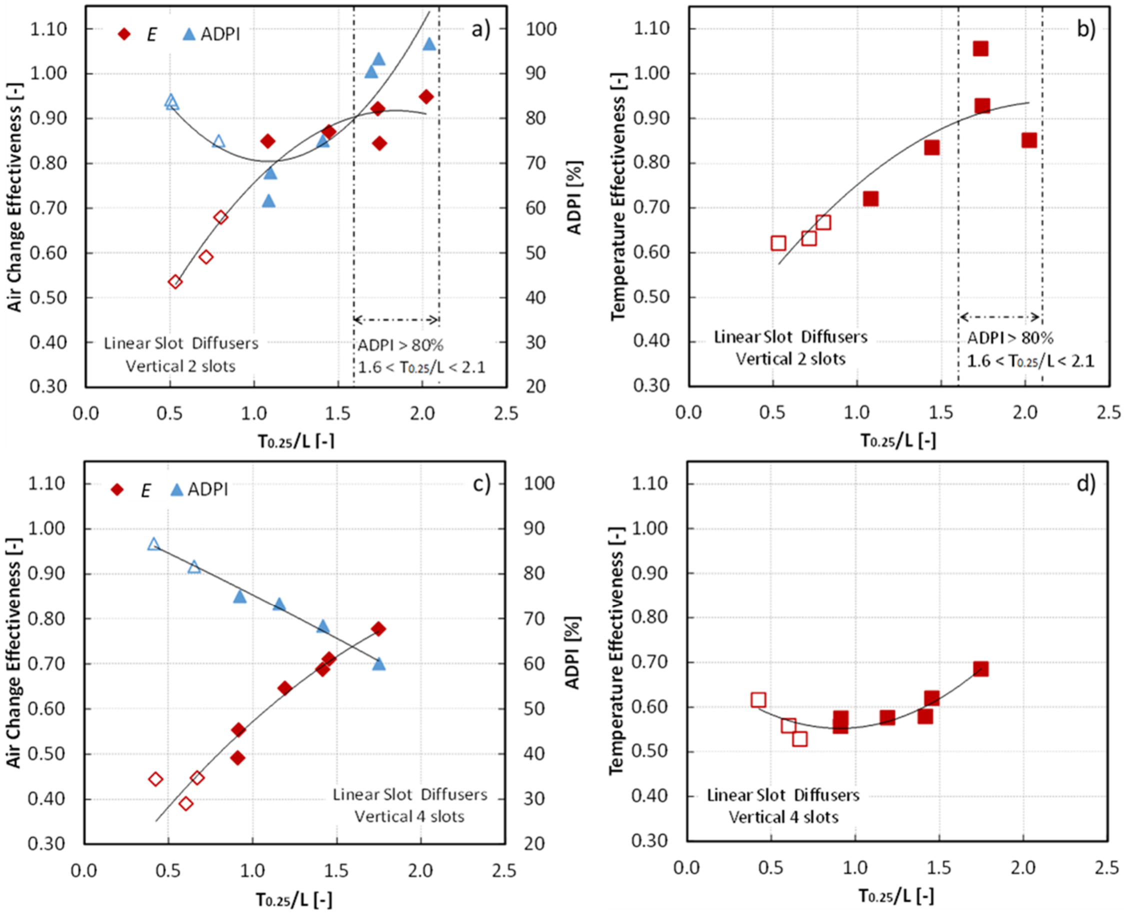
3.3.1.1 ADPI, air change effectiveness and temperature effectiveness

Figure 3-4 shows the results of ADPI, E, and $\mathcal{E}_T$ with ΔT = -5 °C. Figure 3-4 displays the results of 2-slots diffusers (Figure 3-4a and 3-4b) and 4-slots diffusers (Figure 3-4c and 3-4d). The solid marks indicate that the supply air temperature (T_{SA}) was less than 8 °C higher than the average occupied space temperature ($<T>_0$). The dashed marks indicate that the T_{SA} value was 8 °C or higher than $<T>_0$.

With 2-slots linear slot diffusers (Figure 3-4a), a higher ADPI was found for lower and higher $T_{0.25}/L$ values. Although the ADPI was high for lower $T_{0.25}/L$ values, E and $\mathcal{E}_T$ were low. The range of $T_{0.25}/L$ that can achieve an ADPI higher than 80 % with a T_{SA} less than 8 °C above $<T>_0$ (herein after recommended range) was 1.6–2.1. E and $\mathcal{E}_T$ were approximately 0.9 within the recommended range. The small momentum of supply air at low $T_{0.25}/L$ values could not provide enough mixing in the space and resulted in high thermal stratification and a low ADPI. Conversely, a strong vertical momentum increased the overall mixing performance of the space, resulting in higher ADPI, E and $\mathcal{E}_T$ values. However, the excessive air speed may result in discomfort that is asociated with a draft in the area below the diffuser.

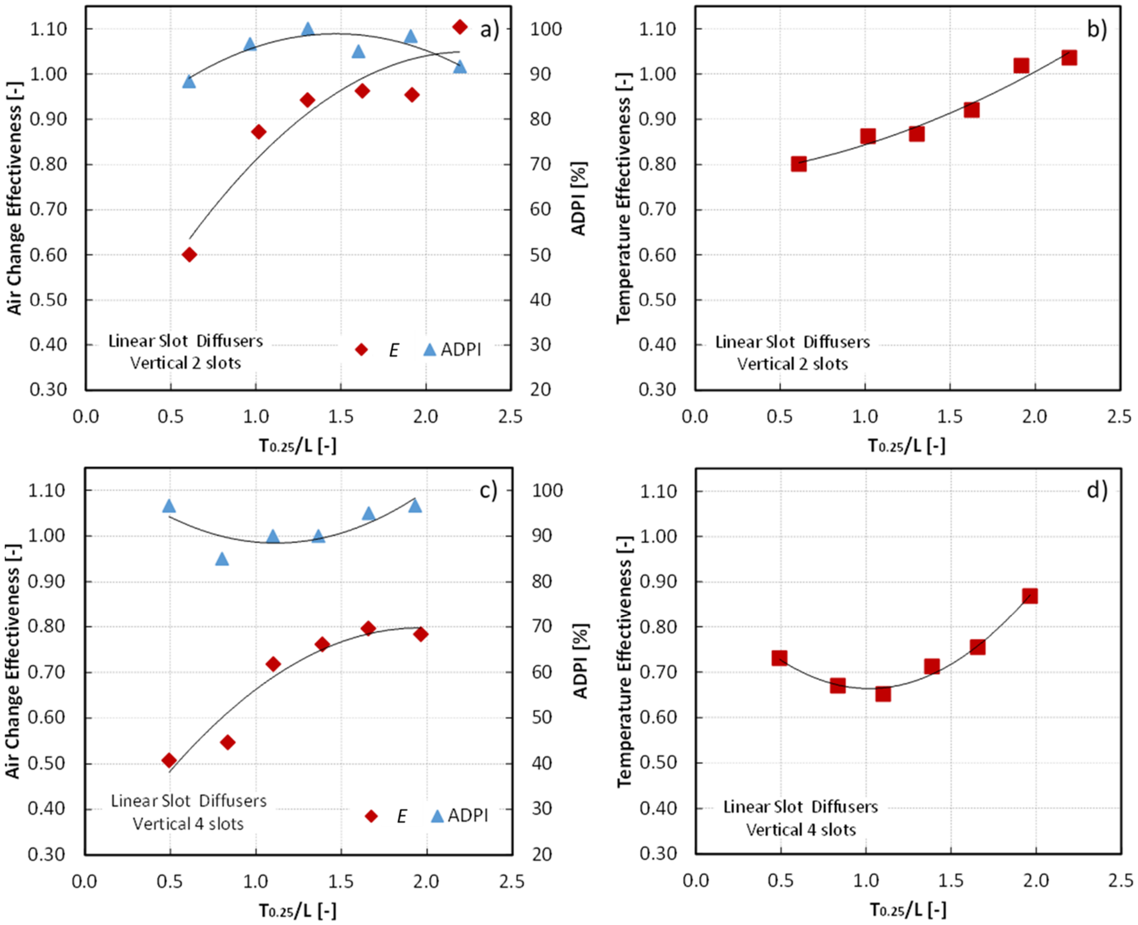
Different from the results for 2-slot diffusers, the ADPI was decreased as $T_{0.25}/L$ increased with 4-slot diffusers (Figure 3-4c). An ADPI higher than 80 % was found only for lower $T_{0.25}/L$ values at which high thermal stratification resulted in lower E and $\mathcal{E}_T$ values. On the contrary, higher $T_{0.25}/L$ improved both E and $\mathcal{E}_T$ owing to an increased mixing effect. The 4-slot diffusers performed quite differently from 2-slot diffusers with regards to the ADPI. This will be further discussed in the later results section with descriptions of room air velocity and temperature fileds.

Figure 4-5 shows the results of ADPI, E, and $\mathcal{E}_T$ measurements with $\Delta T = -2$. The ADPI was higher than 80 % for both 2-slot and 4-slot diffusers within the tested ranges of $T_{0.25}/L$, implying that a small ΔT enhanced ADPI significantly. The highest E was greater with 2-slot diffusers than with 4-slot diffusers: E was 1.1 with 2-slot diffusers and 0.8 with 4-slots diffusers. The highest $\mathcal{E}_T$ was also greater with 2-slot diffusers than with 4-slot diffusers: 1.05 with 2-slot diffusers and 0.9 with 4-slot diffusers.



* mark with fill: T_{SA} higher than 8 °C above $<T>_0$, mark without fill: T_{SA} higher than 8 °C or larger above $<T>_0$
** $T_{0.25}/L$ is based on corrected $T_{0.25}$ from $<T>_0$ and T_{SA}.

Figure 3-4 Linear slot diffusers with vertical flow under ΔT = -5 °C. a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 2-slot diffusers: cases a.1–6 and cases B.1–8, b) $T_{0.25}/L$ vs. temperature effectiveness with 2-slot diffusers: cases a.1–6 and cases B.1–8, c) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 4-slot diffusers: Cases A.7–12 and cases B.9–17, d) $T_{0.25}/L$ vs. temperature effectiveness with 4-slot diffusers: cases a.7–12 and cases b.9–17



* $T_{0.25}/L$ is based on corrected $T_{0.25}$ from $<T>_0$ and T_{SA}.

Figure 3-5 Linear slot diffusers with vertical flow under ΔT = -2 °C. a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 2-slot diffusers: Cases A.13–18 and cases B.18–23, b) $T_{0.25}/L$ vs. temperature effectiveness with 2-slot diffusers: Cases A.13–18 and cases B.18–23, c) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 4-slot diffusers: Cases A.24–29 and cases B.24–29, d) $T_{0.25}/L$ vs. temperature effectiveness with 4-slot diffusers: Cases A.24–29 and cases B.24–29

3.3.1.2 Room air velocity and temperature

Figure 3-6 shows the results of velocity and temperature fields with ADPI measurements (cases A.1, A.3, A.5, and A.12). The temperature fields are shown with temperature differences between supply air and point of measurements *i* ($\Delta T_{SA\text{-}i} = T_{SA}$-$T_i$). For case A.1 (2 slots, $T_{0.25}/L$: 0.5 (2.1 h^{-1})), the air velocity below 1.7 m was less than 0.25 m/s. The temperature stratification (maximum temperature difference within occupied space) was less than 2 °C. For case A.3 (2 slots, $T_{0.25}/L$: 1.1 (4.5 h^{-1})), the jet from the diffuser (velocity higher than 0.25 m/s) reached 1.7 m. The temperature stratification was 4 °C between 0.1 m and 1.7 m. Because of the bouyancy effect and the weak jet from diffuser, the jet from the diffuser could not reach the bottom end in the higher thermal stratification. It was considered that the thermal stratification in the occupied zone led to a low ADPI. In case A.5 (2 slots, $T_{0.25}/L$: 1.7 (7.2 h^{-1})), the jet from the diffuser reached 0.6 m, and the temperature stratification was the lowest among the shown results. Finally, with case A.12 (4 slots, $T_{0.25}/L$: 1.8 (8.6 h^{-1})), the jet from diffuser reached 1.1 m. The temperature stratification was high and the difference was about 3.5 °C. Although case 12 had almost the same $T_{0.25}/L$ as case A.5 calculated by the method in Chapter 1, the velocity and tempereture field results were different. The calculated correction factor did not accurately normalize the throw of the 2-slot and 4-slot diffusers as the emprical table in ASHRAE handbook (2013) used a supply opening aspect ratio of 1.0 and assumed air flow along with the perimeter wall. It was also considered that the different diffusers' widths affected the obtained temperature fields.

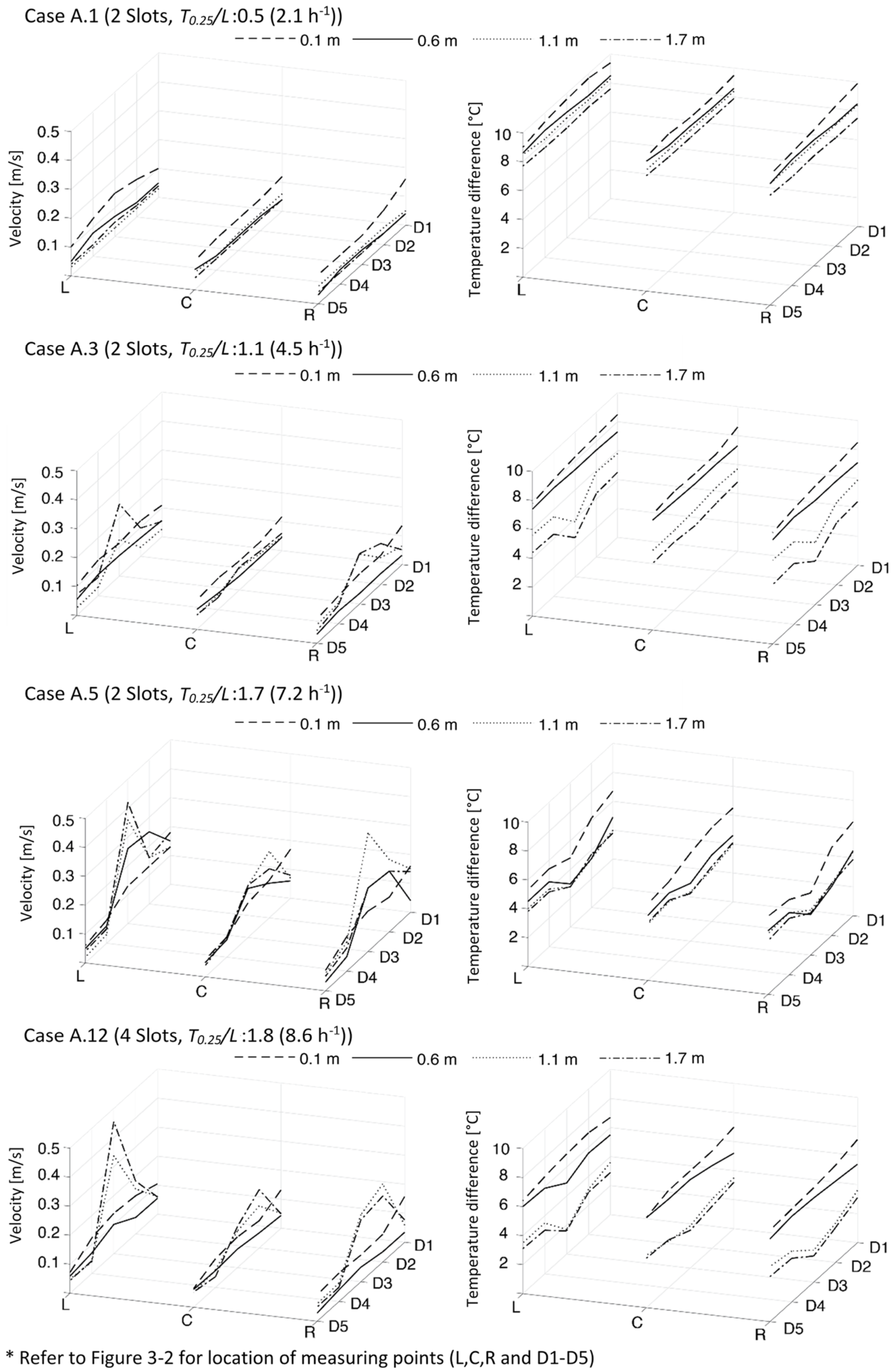


Figure 3-6 Velocity and temperature fields in the ADPI measurements (cases A.1, A.3, A.5, and A.12)

3.3.2 Adjustable blade diffusers with high side wall supply

This section examines the results of adjustable blade diffusers. The first paragraph examines different blade angles. The results are combined with ADPI measurements from a previous study (Liu et al., 2016). The next paragraph discusses the results of different exhaust locations with nominal adjustment.

3.3.2.1 ADPI, air change effectiveness and temperature effectiveness

Figure 3-7 shows the results of ADPI, E ,and $\mathcal{E}_T$ measurements with different blade angles: 0° horizontal, 45° downward, and 45°upward. The ADPI values shown in Figures 3-7a, 3-7c, and 3-7e on the right side of the y-axis were extracted from a previous study (Liu et al., 2016). The solid marks indicate that T_{SA}, is at most 8 °C higher than $<T>_0$. The unfilled marks indicate that T_{SA} is 8 °C or higher than $<T>_0$.

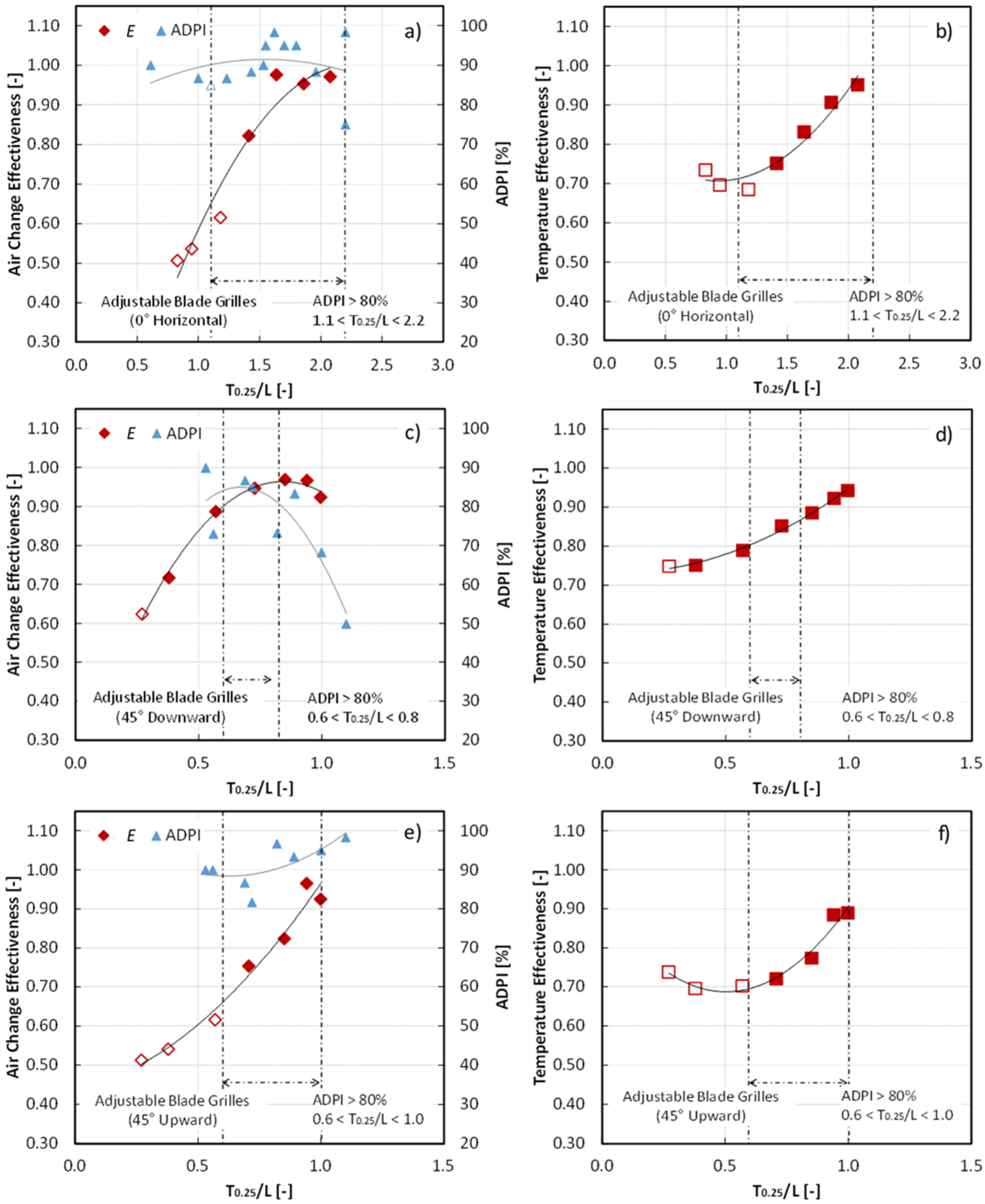
With the 0° horizontal adjustment in Figures 3-7a and 3-7b, the ranges of E and $\mathcal{E}_T$ within the recommended ranges were 0.65-0.98, and 0.71-0.95, respectively. E was significantly decreased when $T_{0.25}/L$ was small. Even within recommended range, E and $\mathcal{E}_T$ were approximately 0.6 and 0.7, respecitvely. The thermal stratification for such conditions was quite high as T_{SA} was 8 °C or higher than $<T>_0$. With the 45° downward adjustments in Figure 3-7c and 3-7d, the ranges of E and $\mathcal{E}_T$ within the recommended ranges were 0.92–0.97 and 0.81–0.90, respectively. The ADPI changed inversely with E and $\mathcal{E}_T$ when $T_{0.25}/L$ increased, as downward jets from the diffuser with higher speed increased draft discomfort in the occupied zone while the jets improved the mixing effect that improves ventilation effectiveness. With the 45° upward adjustments in Figure 3-7e and 3-7f, the ranges of E and $\mathcal{E}_T$ within the recommended ranges were similar to those of the 0° horizontal adjustment: 0.68–0.97 and 0.70–0.90, respectively.

E significantly decreased when $T_{0.25}/L$ was smaller than the recommended range as the low momentum of the diffuser jet could not support the mixing of the supplied air in the occupied space.

3.3.2.2 Impact of exhaust locations with nominal adjustment

Figure 3-8 shows the results of *E* and $\mathcal{E}_T$ with different exhaust locations with the 0° horizontal adjustment. Figure 3-8a shows *E*, and Figure3-8b shows $\mathcal{E}_T$. The vertical dashed lines display the recommended range with EX Cn.

The results reveal that *E* and $\mathcal{E}_T$ were almost equivalent to those for EX Cd. The EX Cw location yielded lower *E* and $\mathcal{E}_T$ than the EX Cn location. The maximum *E* and $\mathcal{E}_T$ were 0.7 because short circuits occurred when the exhaust was located at the opposite side of the diffuser. With near floor exhaust (EX Fw and EX Fd), *E* and $\mathcal{E}_T$ were significantly higher than for the ceiling mounted exhaust, especially at low $T_{0.25}/L$. With EX Fw, *E* and $\mathcal{E}_T$ were greater than 1.0 when $T_{0.25}/L$ was lower than 1.7. *E* and $\mathcal{E}_T$ slightly decreased as $T_{0.25}/L$ increased. As the exhaust was located on the opposite side of the diffuser, short circuit flow may have increased once the jet from the diffuser reached the opposite side of the wall. With Ex Fd, *E* and $\mathcal{E}_T$ were higher than 1.0 and were not sensitive to $T_{0.25}/L$. The results suggest that a near floor exhaust significantly improves ventilation effectiveness and air distribution performance compared to a ceiling mounted exhaust. A higher supply airflow rate (higher $T/L_{0.25}$) may not be helpful for effective air distribution for the floor mounted exhaust.



* mark with fill: T_{SA} higher than 8 °C above $\langle T \rangle_0$, mark without fill: T_{SA} higher than 8 °C or larger above $\langle T \rangle_0$

** $T_{0.25}/L$ is based on corrected $T_{0.25}$ from $\langle T \rangle_0$ and T_{SA}.

Figure 3-7 Adjustable blade grill with different blade angles (cases B.30–36, cases B.37–43 and cases B.44–50, and ADPI from a previous study (Liu et al., 2016). a), c) and e): $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis), b), d) and f): $T_{0.25}/L$ vs. temperature effectiveness

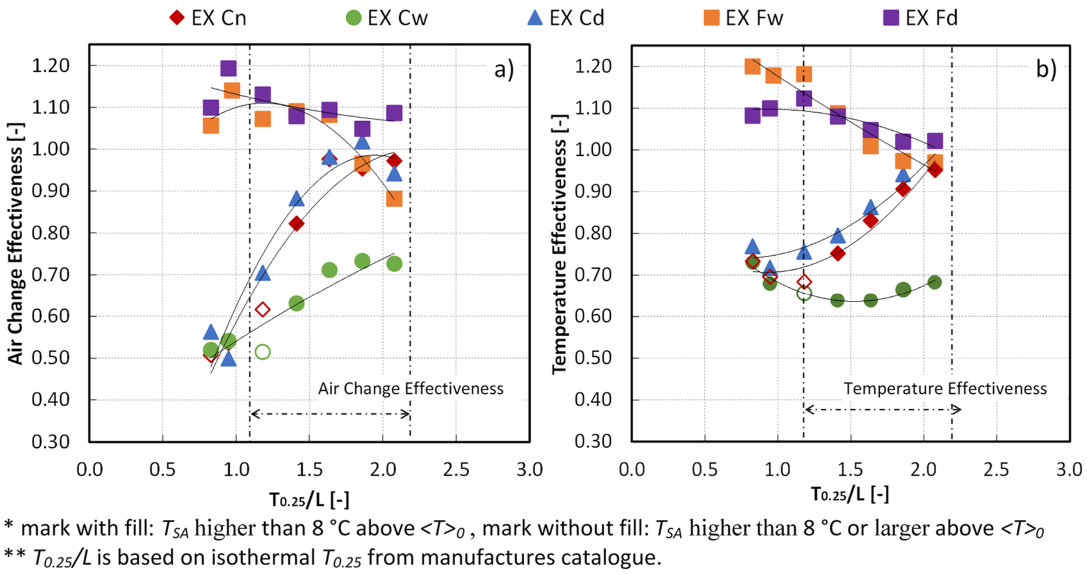


Figure 3-8 Adjustable blade grill (0° horizontal adjustment) with different exhaust locations (cases B.30–36, cases B.51–57, cases B.58–63, cases B.64–71, and cases B.72–78). a) $T_{0.25}/L$ vs. air change effectiveness, b) $T_{0.25}/L$ vs. temperature effectiveness

3.3 Discussion

This section discusses the results of the experiments and the improvements in E and $\mathcal{E}_T$. The fist section summarizes the ranges of E and $\mathcal{E}_T$ within recommended ranges for tested diffusers. The next section discusses the possible measures to improve E and $\mathcal{E}_T$ while maintaining an acceptable ADPI.

3.3.1 Range of air change effectiveness and temperature effectiveness

Table 3-3 provides a summary of E and $\mathcal{E}_T$ within the recommended range. This table intends to update the design guideline in Chapter 2. The data shown in the table are from the experiments with ΔT = -5 °C. The ranges of heating loads were calculated from ΔT and air flow rates of the exhaust air streams. The 4-slot linear slot diffusers with a vertical flow are not covered as the recommended range is determined in this study.

E and ε_T were higher than 0.8 for 2-slot linear slot diffusers with vertical flow. With adjustable blade diffusers, both E and ε_T increased with an increase in $T_{0.25}/L$. The results also reveal that E can be less than 0.8 under 0° horizontal and 45° upward conditions. E and ε_T were higher than 0.8 under 45° downward conditions, and the recommended range was shorter in 45° downward than in 45° upward conditions.

Table 3-3 Ranges of air change effectiveness and temperature effectiveness within recommended range of $T_{0.25}/L$ in terms of ADPI

Diffuser type		Recommended range of $T_{0.25}/L$ with regards to ADPI		Range of E and ε_T within recommended $T_{0.25}/L$ with regards to ADPI		
		$T_{0.25}/L$	Loads	E	ε_T	Loads
		[-]	[W/m^2]	[-]	[-]	[W/m^2]
Linear slot diffusers	Vertical 2 slots	1.4–2.1	12–31	0.87–0.95	0.84–0.95	10–27
Adjustable blade grill	0° horizontal	1.1–2.2*	35–40*	0.65–0.98	0.71–0.95	9–36
	45° downward	0.6–0.8*	35–40*	0.92–0.97	0.81–0.90	9–33
	45° upward	0.6–1.0*	35–40*	0.68–0.97	0.70–0.90	9–35

* Results extracted from previous studies (Liu et al., 2016)

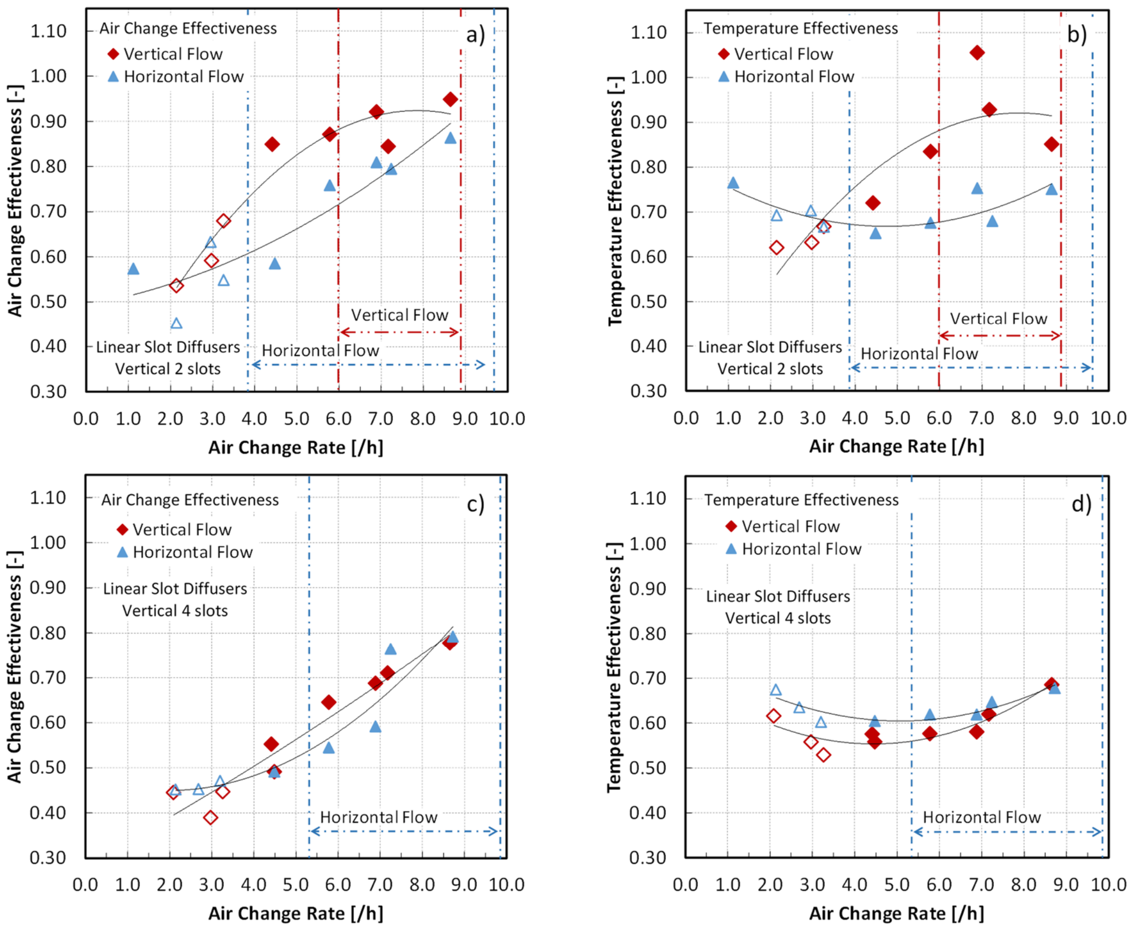
3.3.2 Improvement of air change effectiveness and temperature effectiveness

This section discusses improvements in E and ε_T due to the use of different supply diffuser adjustments, supply and exhaust air temperature differences, and exhaust locations.

3.3.2.1 Diffuser adjustments

Figure 3-9 compares vertical and horizontal flow adjustments of linear slot diffusers under ΔT = -5 °C. E and $\mathcal{E}_T$ results with the horizontal flow adjustments are extracted from Chapter 2. Figure 3-9 also shows the recommended ranges with dashed lines. The ranges indicate the air change rate that was converted from different $T_{0.25}/L$.

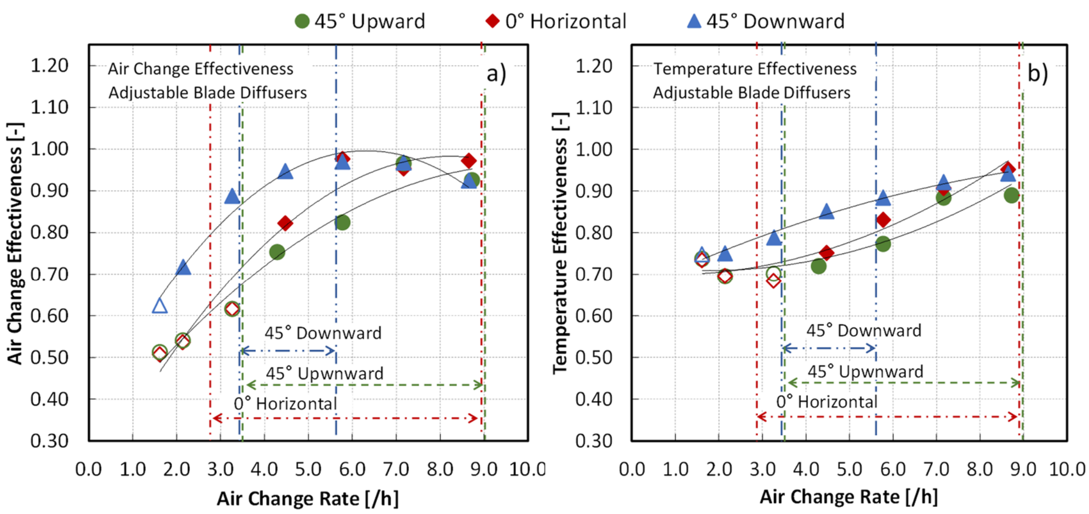
With 2-slot diffusers in Figure 3-2a and 3-2b, the recommended range was smaller with vertical flow than with horizontal flow. The maximum values of E and $\mathcal{E}_T$ were about 25 % and 30 % with the vertical flow, respectively. Adjusting air flow directions under the heating mode may increase both E and $\mathcal{E}_T$. However, careful considerations must be made as directing air flow to occupied spaces may also decrease the ADPI. Proper adjustment of diffusers and control of air flow rate under heating conditions are necessary to improve E and $\mathcal{E}_T$.



* mark with fill: T_{SA} higher than 8 °C above $\langle T \rangle_0$, mark without fill: T_{SA} higher than 8 °C or larger above $\langle T \rangle_0$

Figure 3-9 Linear slot diffusers with vertical and horizontal flow. a) $T_{0.25}/L$ vs. air change effectiveness with 2 slots: cases B.1–8 and from Chapter 2, b) $T_{0.25}/L$ vs. temperature effectiveness with 2 slots: cases B.1–8 and from Chapter 2, c) $T_{0.25}/L$ vs. air change effectiveness with 4 slots: cases B.9–17 and from Chapter 2, d) $T_{0.25}/L$ vs. temperature effectiveness with 4 slots: cases B.9–17 and from Chapter 2

Figure 3-10 compares *E* and $\mathcal{E}_T$ of different angle adjustments for adjustable blade diffusers under ΔT =-5 °C. The dashed lines display the recommended ranges with air flow rates. The recommended range was smaller with 45° downward than with 0° horizontal and 45° upward. The 45° downward adjustment was able to increase *E* and $\mathcal{E}_T$ about 30% and 15% (maximum values) compared with the 0° horizontal condition. Similar to the adjustment of the linear slot diffuser, downward blades can direct air flow to the occupied space to improve *E* and $\mathcal{E}_T$ under heating conditions. However, air flow rate needs to be properly controlled to avoid significant decreases in the ADPI.



* mark with fill: T_{SA} higher than 8 °C above $\langle T \rangle_0$, mark without fill: T_{SA} higher than 8 °C or larger above $\langle T \rangle_0$

Figure 3-10 Adjustable blade grills with different deflector adjustments (cases B.30–36, cases B.37–43, and cases B.44–50). a) Air change rate vs. air change effectiveness, b) air change rate rate vs. temperature effectiveness

3.3.2.2 Room supply air temperature differences

Figure 3-11 compares ΔT = -2 °C and ΔT = -5 °C under the same heating load for linear slot diffusers with vertical flow. The figure examines how ΔT effects ventilation performance. A lower ΔT may significantly increase E and $\mathcal{E}_T$. Compared to the cases with ΔT = -5 °C, cases with ΔT = -2 °C exhibited an increase in E of about 75 % on average (100 % at the maximum) and also an increase in $\mathcal{E}_T$ of about 45 % on average (65 % at the maximum), respectively. This analysis provides data that HVAC designers can use to determine the optimal design ΔT. A lower ΔT may require less supply air (ventilation rate) to satisfy the required ventilation rate in the occupied zone and less heating energy as it improves both E and $\mathcal{E}_T$. On the other hand, a lower ΔT requires a higher air flow rate to remove the same heating load, which requires more fan energy. For example, 2.5 times more air is required with ΔT = -5 °C to remove the same heating load than with ΔT = -2 °C, if the distributed air is perfectly mixed. The improvements in E and $\mathcal{E}_T$ with lower ΔT need to considered with this trade-off before designers decide on a room supply air difference.

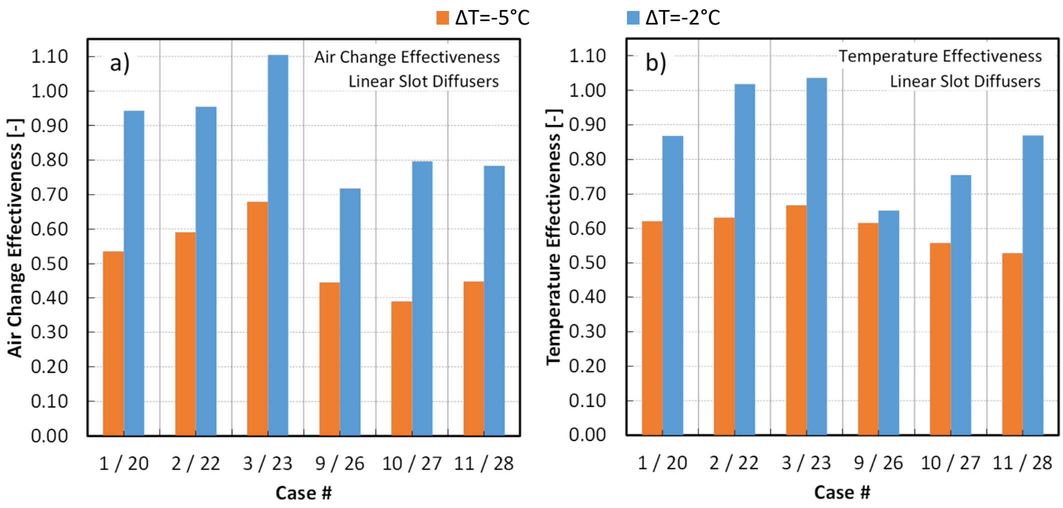


Figure 3-11 Linear slot diffusers with vertical flow under the same heating load (ΔT = -5 °C and ΔT = -2 °C). a) air change effectiveness; b) temperature effectiveness

3.3.2.3 Return air inlet (exhaust) locations

As shown in Figure3-8, an appropriate exhaust location may significantly increase E and $\mathcal{E}_T$. Locating the exhaust near the floor significantly increases E and $\mathcal{E}_T$ by at most 70% with both EX Fw and EX Fd than with EX Cn. However, locating the exhaust near the floor may require more duct work or room space for the HVAC system as a typical HVAC system is installed in the ceiling plenum space. In addition, the results show that an inappropriate exhaust location may decrease E and $\mathcal{E}_T$. With Ex Cw, E and $\mathcal{E}_T$ decrease by at most about 30 % than with the EX Cn location. A designer also needs to consider proper exhaust locations, especially when the high side wall supply strategy is utilized.

3.4 Conclusions

Experimental measurements of ADPI, E, and $\mathcal{E}_T$ in a full-scale test room were described in this chapter. Combined with previous studies (Liu et al, 2015, 2016, 2017, Amai et al, 2016), the results provide supportive data for optimal diffuser selection in mixing ventilation, with an emphasis on the improvement of E and $\mathcal{E}_T$ in heating applications. Proper application of each tested strategy, diffuser adjustment, lower ΔT, and exhaust location may significantly improve E and $\mathcal{E}_T$. This study shows that proper adjustment of the diffuser, a lower ΔT, and a different exhaust location may result in minimum, average, and maximum improvements in E of about 25–30 %, 75%, and 70%, respectively. However, the designer should consider other aspects of those strategies, such as a narrower range of the recommended $T_{0.25}/L$ with vertical flow and trade offs of fan power with lower ΔT.

Nomenclature

ADPI	Air Diffusion Performance Index
EDT	Effective Draft Temperature
$T_{0.25}$	Terminal Velocity Value of 0.25 m/s
L	Characteristic Length
E	Air Change Effectiveness, Arithmetic Mean of E_i in Occupied Zone
E_i	Local Air Change Effectiveness
A_i	Age of Air at a Location i
C_0	Initial Concentration of Tracer Gas at the time $t = 0$
$C(t)$	Decayed Concentration Measured at Time t
τ_n	Nominal Time Constant
ε_T	Temperature Effectiveness
T_{SA}	Supply Air Temperature
T_{EA}	Exhaust Air Temperature
T_i	Temperature at a Location i
$<T>_0$	Average Temperature in Occupied Space
ΔT	Exhaust Air and Supply Air Difference, T_{EA}-T_{SA}

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Chapter 4

Assessing the Measuring Procedure for Ventilation Effectiveness

4.1 Introduction

American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 62.1 (2010) specifies the minimum ventilation rates in buildings with different purposes. This ventilation rate is increased or decreased to take into account the impact of ventilation effectiveness. ASHRAE Standard 129 (2002) specifies how to modify the minimum ventilation rate by ventilation effectiveness. On the other hand, Society of Heating, Air-Conditioning and Sanitary Engineers of Japan (SHASE) Standard -102 (2011) specifies the methods that should be used to calculate the required ventilation rate when ventilation effectiveness is considered. SHASE also specifies the method that should be used to measure ventilation effectiveness in Standards 115 (2010) and 116 (2011). However, the Japanese building code only regulates the minimum ventilation rate per occupants. Ventilation effectiveness does not take into account the building code. Both ASHRAE and SHASE introduce ventilation effectiveness indices based on the age of air concept based on tracer gas measurements. The air change effectiveness is defined in the ASHRAE standard 129. The air change effectiveness is 1.0 when the air from the diffuser is perfectly mixed in the space. SHASE standards (2010, 2011) mention the standardized concentration in the occupied zone, which is inverse of the air change effectiveness. Furthermore, standard procedure for measuring air change effectiveness is defined in ASHRAE Standard 129 (2002). The standard requires air change effectiveness measurements to be conducted at 25% of the workstations or at least 10 locations in the test space. On the other hand, the SHASE standard 115 (2010) requires measurement at a minimum of three points or three repetitive measurements with one point in the target space. The measuring point should be near the center of each span (between columns) or each 10 m by 10 m grid. The ASHRAE Standard 129 (2002) requirement is more stringent than the one in SHASE standard 115(2010) as it requires measurements to be conducted at ten locations at least while SHASE requires only three measurements. Measuring ventilation effectiveness in the field while meeting the requirements of ASHRAE Standard 129 (2002) demands extensive measuring equipment or repetitive measurements. As a consequence, air change effectiveness is rarely measured in the field. As ventilation effectiveness may have significant impact on both indoor air quality and building energy performance, it is important to know how to properly measure it. Furthermore, practitioners would benefit if there were alternate methods for predicting ventilation effectiveness in the field. Therefore, the objective of this chapter is to assess a procedure of evaluating air change effectiveness of mixing ventilation with various conditions through intensive laboratory experiments. Variances of the local air change effectiveness in an occupied space and the correlation of temperature distribution in the space and the ventilation effectiveness were analyzed.

4.2 Methodology

In this chapter, the same experiments described in Chapters 2 and 3 were used. The results were anlysed in terms of the variance and coorelaiton of air change and temperature effectiveness. Figure 4-1 shows the chamber geometry and sensor positions, and Figure 4-2 shows the five different tested diffusers; linear slot diffuser (Price: model SDS75), round ceiling diffuser (Metalaire: model 3000-1), louvered face diffuser with no lip (Metalaire: model 5000-1), perforated diffuser directional pattern (4way) (Titus: model PCS) and adjustable blade diffusers (Model 51DV, frame size: 150 mm × 600 mm, Nailor HVAC, Inc.). Please refer to Chapters 2 and 3 for more details. The experiments utilzed CO_2 in the tracer gas decay test. The vertical temperature distribution was simultaneously measured at one location. Detailed horizontal temperature distributions were measured in some cases and the results are given in Chapter 3. The results revealed that horizontal temperature variance was minimal in the measured cases.

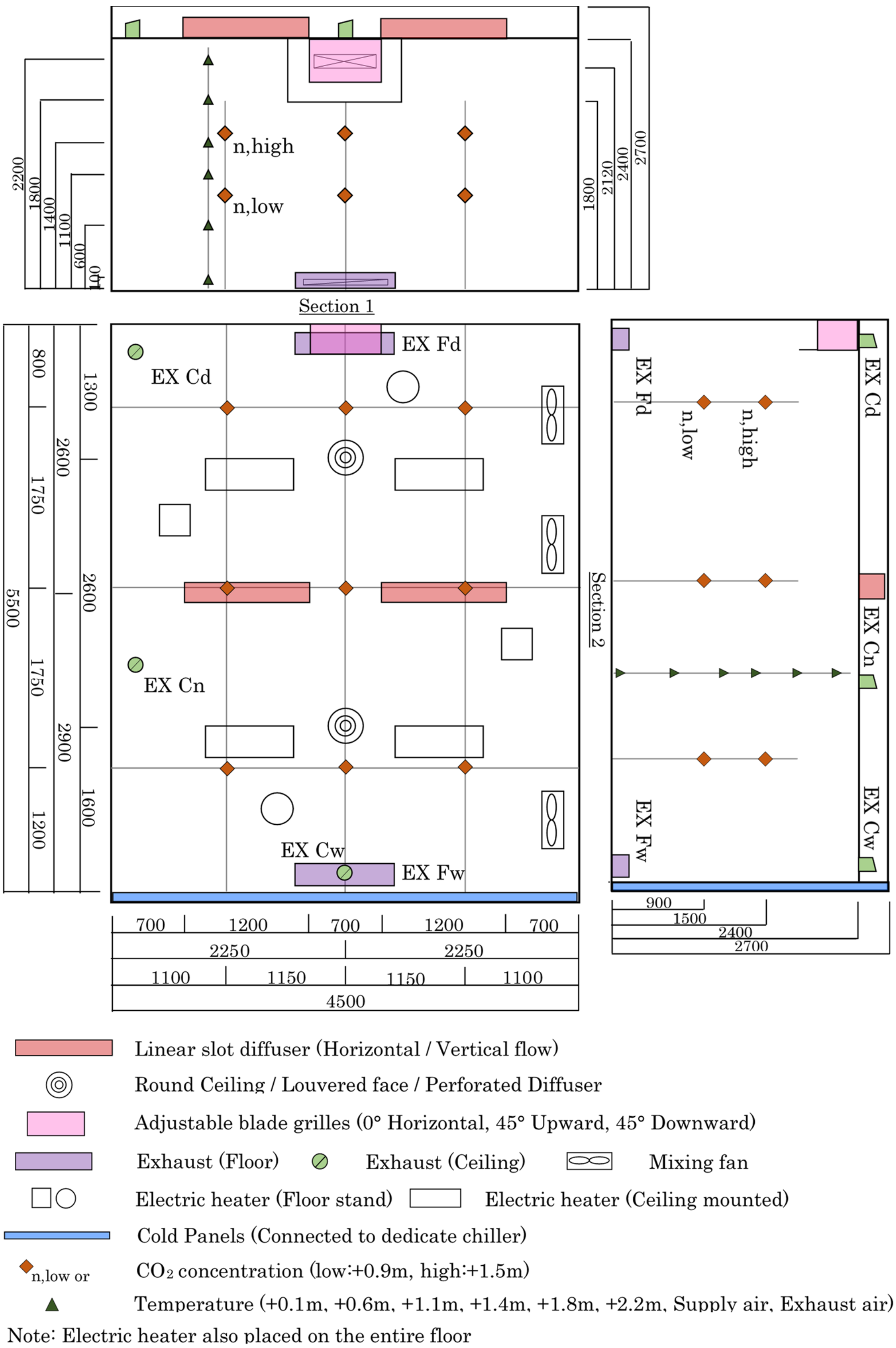


Figure 4-1 Experimental setup of a test room

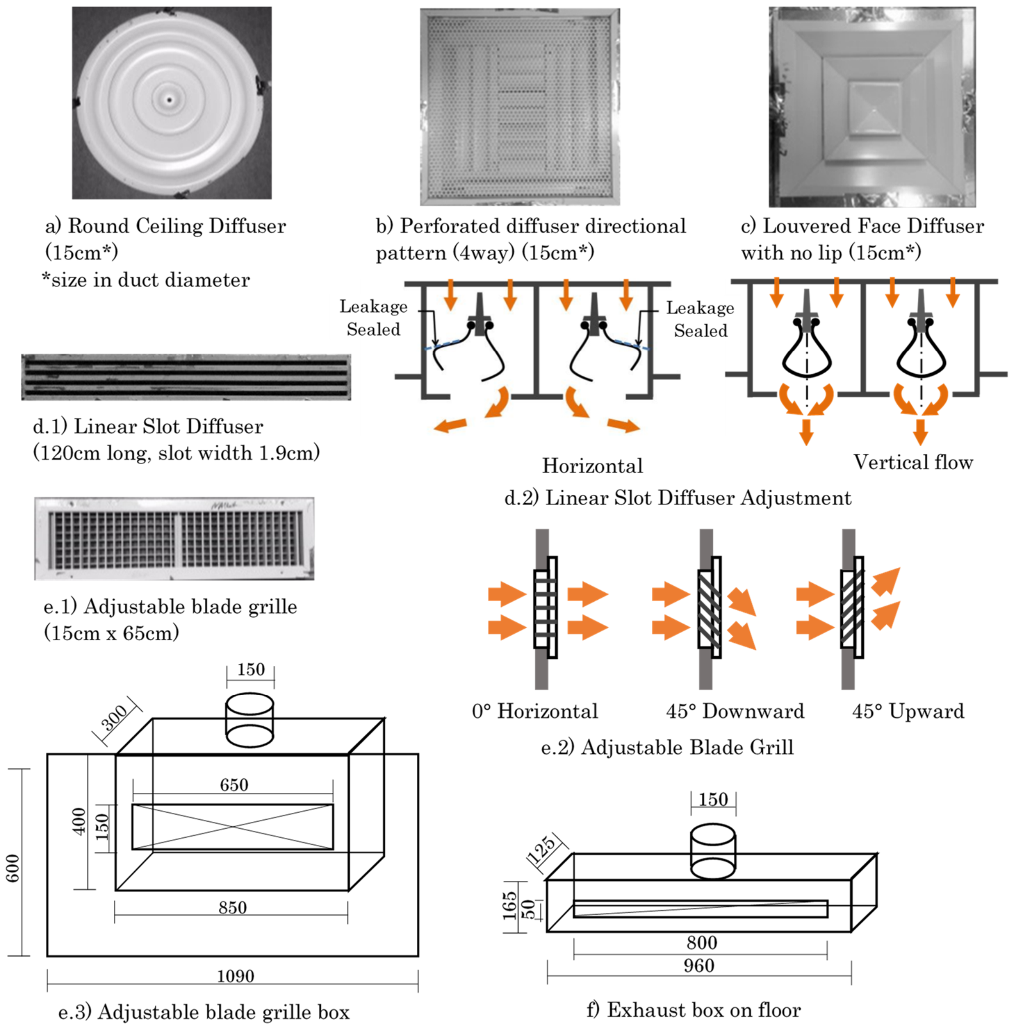


Figure 4-2 Tested diffusers

Table 4-1 and Table 4-2 shows the experimental conditions. As explained in previous sections, the experiments are the same as those in Chapters 2 and 3. However, case numbers were reorganized to make the analysis more comprehensive. The experiments were conducted for heating, cooling, and perfect mix conditions with five different diffusers, diffuser adjustments, exhaust locations, air flow rates, and internal loads. Overall, 179 experiments were conducted: 143 cases for heating, 26 cases for cooling, 10 cases for perfect mixing. A total of 7 sets of experiments were repeated twice to evaluate the uncertainty in the experiments. For the perfect mix test (cases 170-179), the mixing fans were operated during the decay measurement. Under heating cases, the cooled panels mimicked the heating load, and the supply air temperature was set to achieve the target ΔT (-2°C, -5°C and -8°C). In addition, the cooled panels were controlled to maintain a surface temperature of 32 °C for cases with ΔT = -2 °C, 18.5 °C for cases with ΔT = -5 °C, 15.5 °C for cases with ΔT = -2 °C. The temperature of cooling panels was designed to prevent condensation during experiments. In cooling cases, electric heaters mimicked the cooling load and the output of the heater was determined from the air change rate of experimental conditions and the target ΔT (8 °C). Then, supply air temperature was adjusted to ensure ΔT (8 °C) was achieved. The measured supply/exhaust air temperature of heating cases were 38 °C/36 °C max. and 36 °C/33 °C min. for cases with ΔT = -2 °C, 38 °C/34 °C max. and 30 °C/25 °C min. for cases with ΔT = -5 °C, and 41 °C /35 °C max. and 39 °C/29 °C min. for cases with ΔT = -8 °C, respectively. The measured supply/exhaust temperature of the cooling cases were 14 °C/22 °C max. and 12 °C /20 °C min.

According to ASHRAE Standard 129 (2002), perfect mixing is defined as a theoretical airflow distribution in which the concentration of all constituents in the air, and the age of air, are spatially uniform. In this study, perfect mixing was assumed to occur when the air change effectiveness at all measured point was 1.0 ± 0.08 based of the uncertainty assumed in the ASHRAE Standard 129 (2002). It was confirmed that perfect mixing was achieved by using the three mixing fans as all the measured E_i of cases 170–179 were in the defined range.

Table 4-1 Experimental matrix (heating)

Case #	Supply Diffusers		Air change rate [h^{-1}]	ΔT** [°C]	Exhaust (EX) Location
	Type*	Adjustment			
1–9	LS (2 slots)	Horizontal	1.1, 2.1, 3.0, 3.3, 4.5, 5.8, 6.9, 7.2, 8.6	-5	EX Cn
10–15	LS (2 slots)	Horizontal	2.1, 3.3, 5.7, 5.8, 7.2, 8.6	-2	EX Cn
16–24	LS (4 slots)	Horizontal	**2.1, 2.1**, 2.7, 3.2, 4.5, 5.8, 6.9, 7.2, 8.7	-5	EX Cn
25–30	LS (4 slots)	Horizontal	2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-2	EX Cn
31–37	RC		2.1, 3.3, 4.5, 5.8, 7.2, 8.6, 8.6	-8	EX Cn
38–43	RC		2.1, 3.8, 4.6, 5.8, 7.2, 8.6	-5	EX Cn
44–50	RC		2.1, 2.2, 3.3, 4.4, 5.7, 7.2, 8.6	-2	EX Cn
51–56	LF/no lip		3,0, 3.9, 4.2, **6.3, 6.3**, 9.4	-5	EX Cn
57–62	PF (4-way)		2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cn
63–72	LS (2 slots)	Vertical	**2.1, 2.1,** 3.0, 3.3, **4.4, 4.4**, 5.8, 6.9 ,7.2, 8.6	-5	EX Cn
73–78	LS (2 slots)	Vertical	2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-2	EX Cn
79–88	LS (4 slots)	Vertical	**2.1, 2.1**, 3.0, 3.3, 4.4, 4.5, 5.8, 6.9, 7.2, 8.6	-5	EX Cn
89-94	LS (4 slots)	Vertical	2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-2	EX Cn
95–101	ABG	0° Horizontal	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cn
102–108	ABG	0° Horizontal	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cw
109–115	ABG	0° Horizontal	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cd
116–122	ABG	0° Horizontal	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Fw
123–129	ABG	0° Horizontal	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Fd
130–136	ABG	45° Upward	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cn
137–143	ABG	45° Downward	1.6, 2.1, 3.3, 4.5, 5.8, 7.2, 8.6	-5	EX Cn

*LS: Linear slot, RS: Round ceiling, LF: Louvered face, PD: Perforated diffusers, ABG: Adjustable blade grill
** ΔT: Exhaust and supply air temperature difference
*** **Underlined cases**: repeated experiments

Table 4-2 Experimental matrix (cooling and perfect mix)

Case #	Supply Diffusers		Air change rate [h^{-1}]	ΔT** [°C]	Exhaust (EX) Location
	Type*	Adjustment			
Cooling					
144–148	LS (2 slots)	Horizontal	2.1, 3.3, 4.5, 5.8, 8.6	8	EX Cn
149–153	LS (4 slots)	Horizontal	2.1, 3.3, 4.5, 5.8, 8.6	8	EX Cn
154–159	RC		2.3, 3.3, 4.4, 5.5, 8.3, 8.6	8	EX Cn
160–164	LF/no lip		2.2, 3.3, 4.4, 5.8, 8.6	8	EX Cn
165–169	PD (4-way)		2.1, 3.8, 4.6, 5.8, 8.6	8	EX Cn
Perfect Mix					
170–179	RC		1.1, 2.3, **3.3, 3.3**, 4.5, 5.7, 5.8, **7.2, 7.2**, 8.6	-5	EX Cn

*LS: Linear slot, RS: Round ceiling, LF: Louvered face, PD: Perforated diffusers, ABG: Adjustable blade grill
** ΔT: Exhaust and supply air temperature difference
*** **Underlines cases**: repeated experiments

4.3 Results

This section examines the vertical, horizontal and overall variances of air change effectiveness in the test space according to the experimental settings. Section 4.3.2 analyses vertical variance and section 4.3.3 analyses horizontal variance. In addition, the correlation between the air change effectiveness and the temperature effectiveness is introduced in section 4.3.4.

4.3.1 Vertical variances in the test space

To evaluate the vertical variances of the air change effectiveness, differences in air change effectiveness between the high measuring plane and low measuring plane (Figure 4-1) were examined. The vertical variance of the air change effectiveness as a percentage at the point i, B_i, is defined as

$$B_i = ABS\left(1 - \frac{E_{i,high}}{E_{i,low}}\right) \times 100 \qquad [\%] \qquad (4\text{-}1)$$

where $E_{i,high}$ and $E_{i,low}$ are the local air change effectiveness values of the high measuring plane and low measuring plane at the same horizontal measuring point, respectively. Figure 4-3 shows the vertical variances of local ventilation effectiveness as percentiles. All B_i according to the experimental settings were rearranged in ascending order. The 5th, 25th, 50th, 75th and 95th percentile values are shown in the figure. The 75th percentile values for all cases were less than 16 %. The vertical variances for cases 10–15 and cases 25–30 (2-slot and 4-slot linear slot diffusers with a horizontal flow under ΔT = -2°C) were slightly higher than those of the other conditions with regards to the median, 75th, and 95th percentile values. Overall, all the heating conditions (cases 1–143), cooling conditions (cases144–169), and experiments (cases 1–179) had similar variances in each percentile. For all experiments (cases1-179), the variances were 19% in the 95th percentile, 8.5% in the 75th percentile, 4.5% in the median percentile, and 2.5% in the 25th percentile. Most of the variances observed this study were close to or less than the uncertainty in the measurement discussed in the previous section.

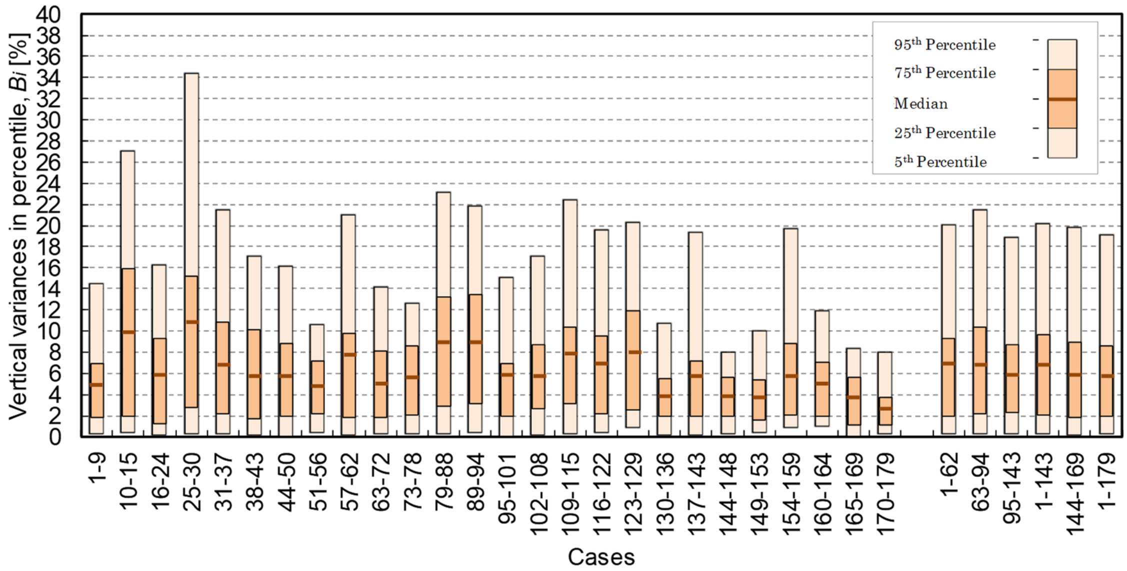
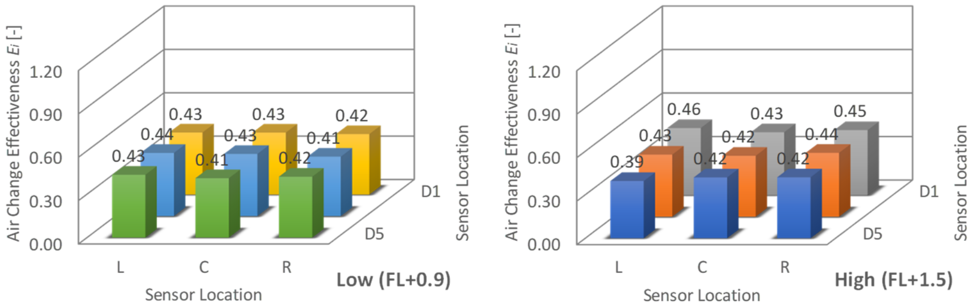


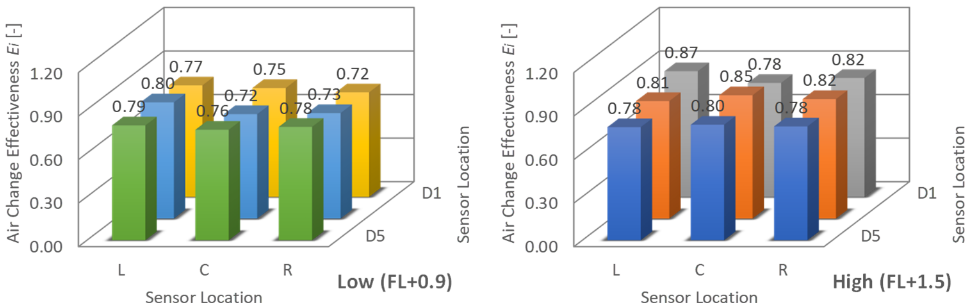
Figure 4-3. Vertical variances in percentile

4.3.2 Horizontal and overall variances in the test space

Figures 4-4,4-5, and 4-6 show the local air change effectiveness distribution in typical cases. Figure 4-4 shows the distribution in case 38 and case 42, figure 4-5 shows the distribution in case 95 and case 99, and figure 4-6 shows the distribution in case 154 and case 157. The left side graphs show the distribution in the low plane and the right side graphs show that in the high plane. For both cases, the difference between the maximum and minimum local air change effectiveness within measurement plane was quite small. For example, for case 38, the minimum local air change effectiveness within the low plane was 0.41, the maximum was 0.44, and the average was 0.42. For case 101, the minimum local air change effectiveness within the low plane was 0.9, the maximum was 1.09, and the average was 0.97. The variances need to be evaluated by indices that relatively show the distribution of different average air change effectiveness values.

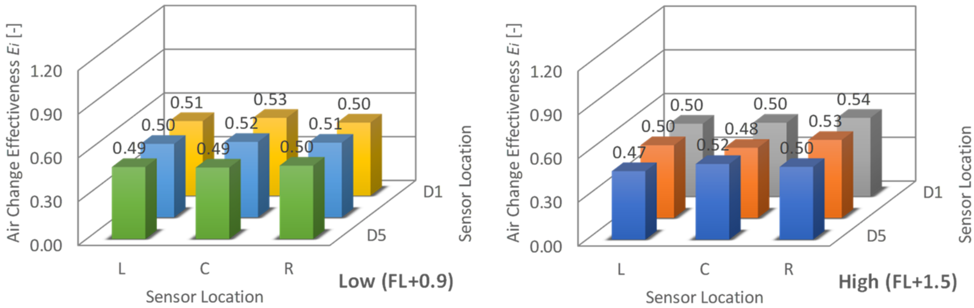


Case 38 (Heating) RC, ACH: 2.1, *ΔT:-5*, EX Location: EX Cn

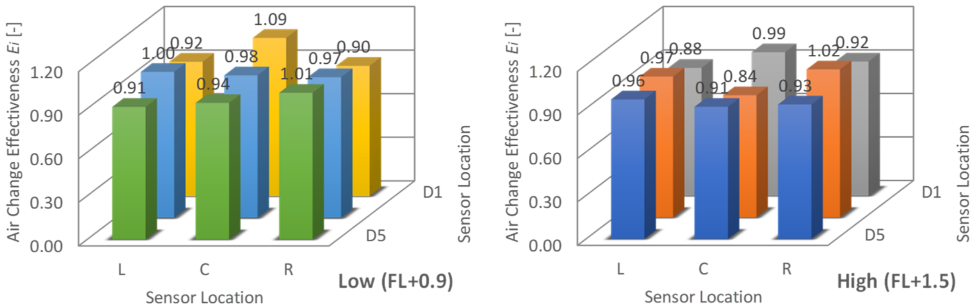


Case 42 (Heating) RC, ACH: 7.2, *ΔT:-5*, EX Location: EX Cn

Figure 4-4 Local air change effectiveness distribution (cases 38 and 42)

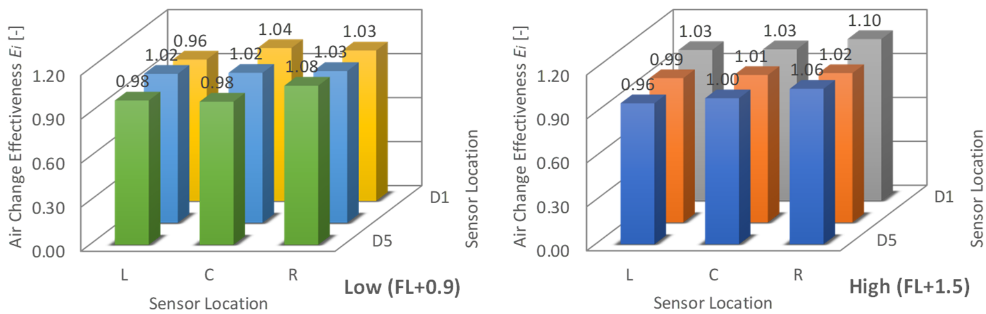


Case 95 (Heating) RC, ACH: 1.6, *ΔT:-5*, EX Location: EX Cn

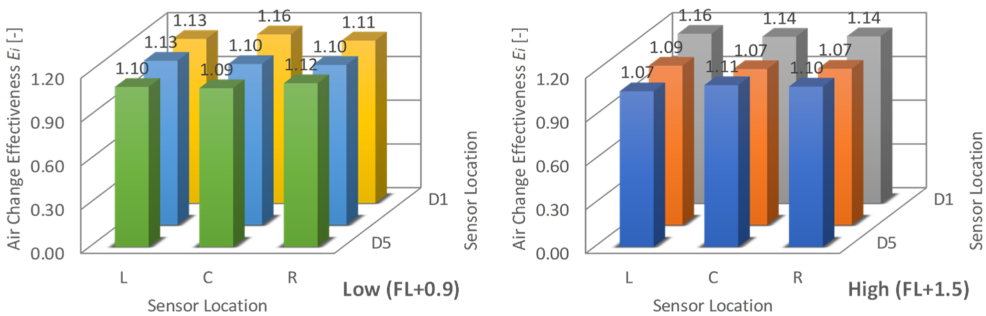


Case 99 (Heating) RC, ACH: 5.8, *ΔT:-5*, EX Location: EX Cn

Figure 4-5 Local air change effectiveness distribution (cases 95 and 99)



Case 154 (Cooling) RC, ACH: 2.3, *ΔT:8*, EX Location: EX Cn



Case 157 (Cooling) RC, ACH: 5.5, *ΔT:8*, EX Location: EX Cn

Figure 4-6 Local air change effectiveness distribution (cases 154 and 157)

To evaluate the horizontal and overall variance of air change effectiveness under different experimental conditions, the standard deviation σ_j was converted to a percentage by the equation that follows. The standard deviation of air change effectiveness expressed as a percentage for the low measuring plane, high measuring plane, or overall test space, C_j, is defined as

$$C_j = ABS\left(1 - \frac{\sigma_j}{E_j}\right) \times 100 \qquad [\%] \qquad (4\text{-}2)$$

where σ_j is the standard deviation of air change effectiveness for the high measuring plane, low measuring plane, and overall test space, and E_j is E_{low}, E_{high}, or E. Figure 4-4 shows horizontal and overall standard deviations of air change effectiveness as percentages. Cases 10–15 and cases 25–30 (2 slot and 4 slot linear slot diffuser with horizontal flow under ΔT = -2 °C) had slightly higher $C_{j,Max}$ in both low and high horizontal planes and overall. Cases170–179 (perfect mix) had the lowest variances. The variances in cases170–179 were less than 3% on average. The difference between variances in the high measuring plane and in the low measuring plane was small. Cases 1–143 (Overall heating conditions) had slightly higher overall maximum and average variances than cases144–169 (overall cooling conditions). The average variances of cases 1–179 (all experiments) were 4% in the low plane 5% in the high plane and overall test space. Similar to the vertical variances, most of the variances were close to or less than the uncertainty in the measurement.

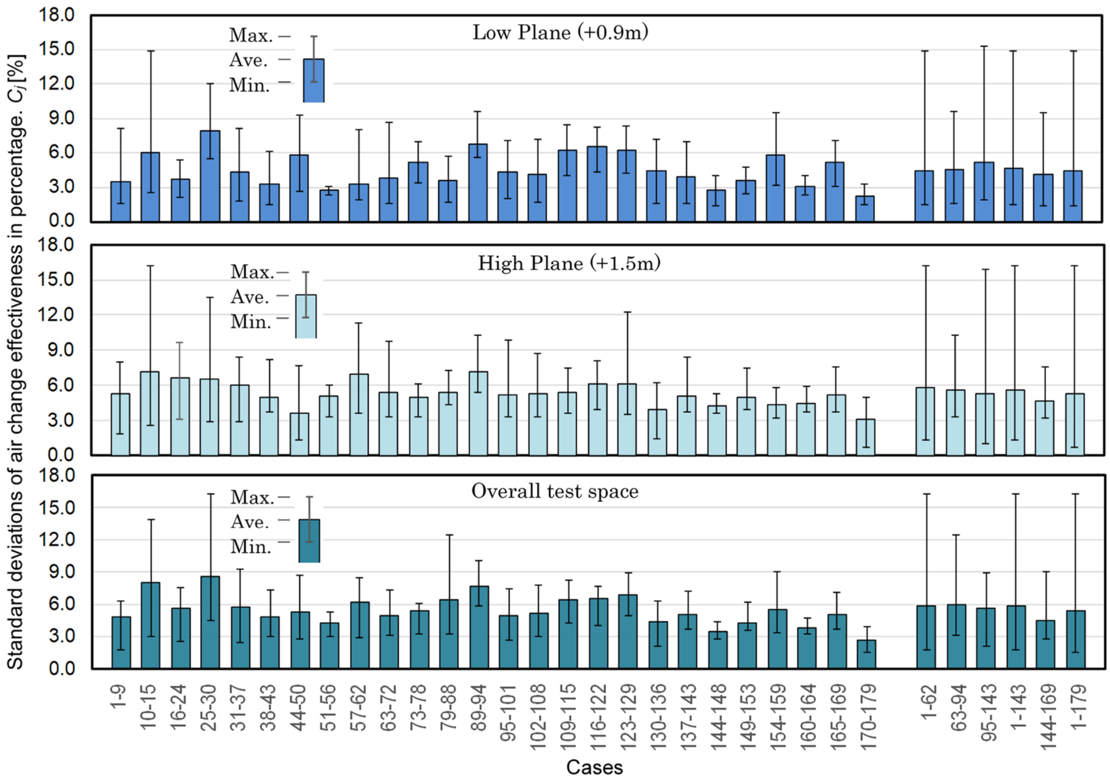


Figure 4-7 Horizontal and overall standard deviations in percentage

4.3.3 Correlation of air change effectiveness and temperature effectiveness

Temperature effectiveness (Etheridge et al., 1996), also defined as ventilation effectiveness for heat removal (Awbi et al., 1993), was implemented to evaluate the temperature gradient in the test space. Temperature effectiveness ($\mathcal{E}_T$) represents the effectiveness of energy utilization supplied into the occupied zone (Etheridge et al., 1996) whereas the air change effectiveness represents the effectiveness of contaminant removal from the occupied zone. This index was introduced detail in Chapter 1.

Figure 4-5 shows the correlation of $\mathcal{E}_T$ and E. The plot shows the temperature effectiveness along the x-axis and the overall air change effectiveness for each case. Plots with dark orange indicate the heating regime with vertical flow, light orange indicate heating with vertical flow, light blue indicate cooling with horizontal flow, and light green show perfect mix results. Second order polynomial curve fitting was applied to generate the profile of $\mathcal{E}_T$ and E for all the cases. As mentioned in previous chapters, $\mathcal{E}_T$ and E are around 1.0 for perfect mix cases and around 1.0 or above for cooling cases. Similar distribution were observed between heating with horizontal flow and vertical flow. From the definition, $\mathcal{E}_T$ is a dimensionless number that evaluates the temperature gradient in the test space whether it is under cooling or heating regimes. Both the denominator and numerator of $\mathcal{E}_T$ are negative under cooling and positive under heating. E can be also utilized to both heating and cooling. Thus, the correlation of $\mathcal{E}_T$ and E can be observed through analyzing the plots of both cooling and heating.

E was close to1.0 when $\mathcal{E}_T$ was also nearly 1.0, and E decreased as $\mathcal{E}_T$ decreased. $\mathcal{E}_T$ is slightly higher than E. As the regression curve shows, E was about 0.5 when $\mathcal{E}_T$ was 0.6. The R_2 value was 0.8. It was considered that E is correlated to temperature stratification of the test space, and $\mathcal{E}_T$ may be an appropriate index for showing this correlation. In addition, significantly low E was found with low $\mathcal{E}_T$. Detailed results and analyses of this low E and supply air diffusers return inlet locations can be found in Chapters 2 and 3.

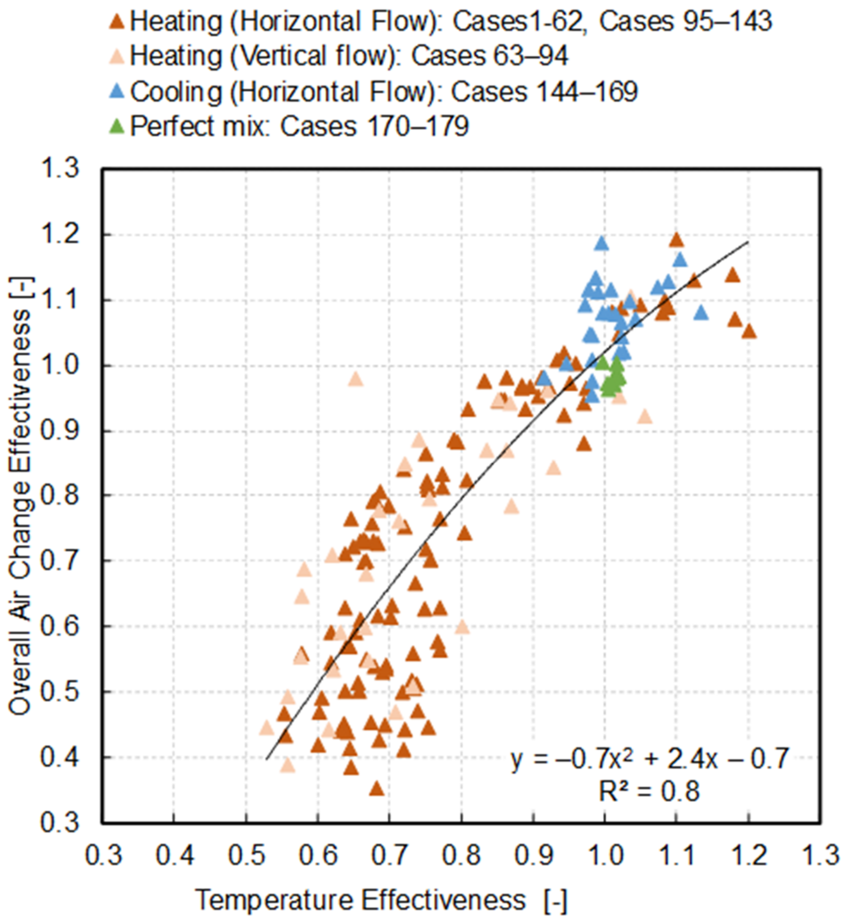


Figure 4-8 Correlation of temperature and air change effectiveness

4.4 Discussion

This section discusses the results of the experiments. The first section discusses the results of local air change effectiveness variances and measurement procedures in utilized standards. The second section discusses correlations of air change effectiveness and temperature effectiveness.

4.4.1 Variances in local air change effectiveness in the test space

ASHRAE Standard 129 (2002) stipulates that E_i should be measured at a minimum of 10 work stations. However, as the results revealed, the variances in E_i were minimal in mixing ventilation. Most of the differences found in the test space were close to the expected uncertainty of the experiment. In addition, previous research revealed the location of the workstation (partition height; 1.9 m, ceiling height; 2.9 m) had no significant effects on the air distribution patterns and the influences of workstation layout on the ventilation efficiency were minimal (Shaw et al. 1993). Lee's study (2004) found that the effects of internal partition were low when the partition height was 60% of the ceiling height and was significantly high when the partition height was 80% of the ceiling height. It is implied that a more conventional evaluation will be possible with a reduced number of measuring points with mixing ventilation when partitions in the space are low enough not to obstruct the air flow pattern in the targeted space.

4.4.2 Correlation between air change effectiveness and temperature effectiveness

A considerable amount of effort is required to conduct tracer gas tests in the field. Taking temperature measurements requires much less effort than conducting tracer gas tests, and many of the building control/monitor systems already measure temperature. The correlation found in this research may aid in the interpretation of the overall air change effectiveness of a space for which conducting the tracer gas test is not practical. However, careful considerations must be made with HVAC systems, especially with regards to the source of heating/cooling in the space. Krajcik et al. (2012) measured air change efficiency and temperature effectiveness in a test chamber with various combinations of radiant floor heating and mixing ventilation with ACH values of 0.5 and 1.0. A correlation between temperature effectiveness and air change efficiency was not observed as an internal heating source might affect the temperature of the occupied zone.

4.5 Conclusions

Extensive experiments of mixing ventilation were conducted in a test chamber to evaluate whether more practical evaluation of ventilation effectiveness would be possible by using fewer measuring points. The results revealed that the vertical, horizontal and overall variances of the local air change effectiveness were minimal. The overall variance of air change effectiveness in the occupied space of a room with ceiling diffusers was less than 16% in most of the cases, which is slightly larger than the experiments' uncertainty. Furthermore, the newly developed correlation of thermal effectiveness and air change effectiveness is considered to be useful as an alternative method to interpret air change efficiency.

Nomenclature

ABG	Adjustable blade grill
ACH	Air change rate per hour
B_i	Vertical variances of air change effectiveness
C_j	The standard deviation of air change effectiveness in percentage
E	Air change effectiveness, arithmetic mean of E_i in occupied zone
E_i	Local air change effectiveness
$E_{i,high}$, $E_{i,low}$	Local air change effectiveness of the high measuring plane and low measuring plane at the same horizontal measuring point
E_{low}	Air change effectiveness, arithmetic mean in low plane
E_{high}	Air change effectiveness, arithmetic mean in high plane
EX	Exhaust
LS	Linear slot diffusers
LF	Louvered face diffusers
PD	Perforated diffusers
RS	Round ceiling diffusers
T_{SA}	Supply air temperature
T_{EA}	Exhaust air temperature
T_i	Temperature at a location i
$<T>_0$	Average temperature in occupied space
ΔT	Exhaust air and supply air difference, T_{EA}-T_{SA}
ε_T	Temperature effectiveness
σ_j	Standard deviation of air change effectiveness

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Chapter 5

Conclusive Summary

Among various types of ventilation, the most known and used ventilation method is mixing ventilation. Current practice related to the air distribution design and diffuser selection is relaying on only ADPI which considers only temperature uniformity and draft, and this does not always result with the proper diffuser selection. The more comprehensive design process for diffuser selection and positioning that considers both thermal comfort and ventilation effectiveness at the same time is needed. On the other hand, although ventilation effectiveness has significant impacts on both indoor air quality and building energy performance, different standard specifies different procedure of measuring ventilation effectiveness. Some standard specifies very stringent procedure, it is important to know how to properly measure it.

The objectives of this research are as follows: 1) Define the operation range for selecting diffusers with an acceptable ADPI and air change/temperature effectiveness. 2) Provide design/operation options that improve the ventilation effectiveness under the heating regime. 3) Assess a procedure for evaluating *E* in mixing ventilation by analyzing the variances in local *E*. The authors performed experimental measurements in a full-scale test room. Carbon dioxide (CO_2) tracer gas decay tests were conducted to measure the age of the air at multiple locations in the test room simultaneously with various types of ceiling diffusers/pattern adjustments at different airflow rates and internal loads.

In Chapter 1, “Introduction” the objectives of the research were formulated with description of background and review of relevant research. The chapter first introduces current ventilation methods, indices regarding ventilation effectiveness and diffuser selection methods. Then, it reviews previous research on ventilation effectiveness in mixing ventilation and ADPI.

Chapter 2: "Experimental Study on Air Change Effectiveness in Mixing Ventilation" – This chapter describes the CO_2 tracer gas decay tests conducted in a full-scale test room with various types of diffusers at different airflow rates and internal loads to evaluate *E* and temperature effectiveness (ε_T). These experiments were conducted in both heating and cooling regimes. Although each diffuser had a unique shape, all diffusers tested showed similar results. The ranges of *E* and ε_T were examined within the recommended range of $T_{0.25}/L$ related to the ADPI; $T_{0.25}/L$ achieved an ADPI of more than 80%. Under the heating regime, the ranges of *E* and ε_T were from 0.56 to 0.87 and from 0.58 to 0.75, respectively, which were within the recommended ranges. A significant decrease in *E* was found when $T_{0.25}/L$ was small. Both *E* and ε_T increased as the supply and exhaust air temperature difference (ΔT) reached close to an isothermal flow. Under the cooling regime, the ranges of *E* and ε_T were from 0.98 to 1.12 and from 0.92 to 1.11, respectively, which were within the recommended ranges. Relatively good mixing occurred under the cooling regime. The range of $T_{0.25}/L$ capable of achieving good mixing under heating conditions was significantly smaller than that under cooling conditions. Thus, along with diffuser selection, the airflow rate and supplied air temperature should be carefully decided for all-air heating and cooling to achieve good mixing and thermal comfort.

Chapter 3: "Improving Ventilation Effectiveness under Heating Mode" – This chapter describes experimental measurements of the ADPI, *E*, and ε_T conducted in the same full-scale test room as described in Chapter 2. The experiments examined simple strategies to overcome the challenges of poor ventilation effectiveness under the heating regime while maintaining an acceptable ADPI. The strategies were focused on a deflector adjustment, lower supply and exhaust air temperature difference (ΔT), and exhaust locations. In addition, the experiments corrected data in the form of ADPI for diffusers with a vertical flow. Results show that a proper application of each tested strategy significantly improved E and $\mathcal{E}_T$. The proper adjustment of the deflector may improve E by a maximum of 30%. A lower ΔT (ΔT+3 °C) and a different valid exhaust location may improve E by approximately 70% on average. These results also show that an improper exhaust location may decrease E and $\mathcal{E}_T$. Moreover, design engineers also need to examine diverse strategies such as a smaller range of the recommended $T_{0.25}/L$ with a vertical flow and an increase in fan power to remove heating load with a lower ΔT.

Chapter 4: "Assessing Measuring Procedure for Ventilation Effectiveness" – This chapter examines vertical, horizontal, and overall variances in the value of local E to determine whether a more practical evaluation of the ventilation effectiveness is possible using fewer measuring points than those addressed in the standards. This chapter also analyzes the correlation between E and $\mathcal{E}_T$ as an alternative method to interpret the E. Results show that the vertical, horizontal, and overall variances in the value of local E are minimal. The overall variance in the value of local E in an occupied space was found to be less than 16% in most cases, which is slightly larger than the uncertainty of the experiment. Furthermore, the newly developed correlation between E and $\mathcal{E}_T$ is considered to be useful as an alternative method for interpreting E.

The presented research provides comprehensive diffuser selection and simple evaluation methods from planning to operation. A proper operation range for the selection of diffusers in mixing ventilation with a good ADPI, E, and $\mathcal{E}_T$ for both cooling and heating regimes is provided. Simple strategies to overcome common challenges under the heating regime are shown with their operation ranges. The provided data and strategies will be valuable for engineers/designers to plan optimized HVAC systems. This research also assessed a simple evaluation method of E that would help to evaluate whether the planned design was properly achieved.

Limitations / Applications

This research focuses on mixed ventilation, the most commonly used ventilation method. Other ventilation methods such as displacement ventilation and piston ventilation are not covered in this research. The experiments were conducted with Group A and Group E diffusers, following ASHRAE Handbook - Fundamental (2009) classification. In other words, the diffusers in Groups B, C, and D were not evaluated in this research. As illustrated in Table-1-1 in Chapter 1, Group A and Group E diffusers, which are mounted in or near the ceiling, discharge air horizontally or vertically. Group B, C, and D diffusers are mounted in or near the floor. To create displacement flow or remove perimeter load, diffusers in Groups B, C, and D may be installed on window sills.

As experiments were conducted in an experimental chamber, the geometry of the tested space was limited. Diffusers were tested with smooth and flat ceilings with a fixed ceiling height of 2.4 m. In addition, the experiments were not conducted with ceilings with complicated shapes, such as soffit ceiling or other decollated ceilings. The application of diffusers to different ceiling heights was not verified in this research. Same as ADPI, the data obtained from this research may most usable with ceiling height between 2.4m and 3m. Further analysis with different room dimensions utilizing CFD or field measurement may help to validate this suggestion. Furthermore, the suggestion may be applicable to air change effectiveness from 0.5 to 1.2 as the measurements were done with that range. It is also worth noting that designers need to consider the load characteristics of the heating load as the experiments mimicked only loads due to a cold window surface. If the space is expected to have larger loads due to heat transfer from the floor, wall, or ceiling, which may cause temperature gradients and variations in the occupied space, several considerations should be made.

With the aforementioned limitations, this research still covers the majority of applications of mixing ventilation with Group A and Group E diffusers, which are most commonly applied in HVAC systems. The findings of this research may be applied to offices, guest rooms in hotel, patients room healthcare facilities, and so on. One may properly select and locate the diffusers and verify their application with data provided by this research. In addition to properly select and locate diffusers, the findings of this research can be used to examine the optimization of an entire HVAC system.

For example, assume an office building with an area of 500 m^2 uses mixed ventilation due to linier slot diffusers with horizontal flow. Furthermore, assume diffusers are selected and located at $T_{0.25}/L$ = 0.25, which is in the recommended range of both cooling and heating regimes. Then, the HVAC design engineer needs to examine the supply air temperature difference during heating. If the number of occupants is 0.2 person/m^2 and the required outside air is 25 m^3/h/person, the total outside air volume is 2,500 m^3/h. When the system is designed with ΔT = -2 °C, from the results of this study, E is 0.9 and the actual outside air volume intake is 2800 m^3/h. When the system is designed with ΔT = -5 °C, the actual outside air volume intake is 3,600 m^3/h and E is 0.7. If it is assumed that the outside air temperature is 2 °C and the design set temperature is 22 °C, the heating capacity required to process the outside air is 18.3 kW and 23.6 kW, respectively.

Secondly, assume that the heating load for the building is 30 W/ m^2 and the heating load of the target room is 20 kW. When this load is processed with ΔT = -2 °C, as ε_T is 0.85 from this research, the actual load is 23.6 kW with an airflow rate 35,700 m^3/ h. When processed with ΔT = -5 °C, as ε_T is 0.65, the actual load is 30.8 kW with an airflow rate 18,700 m^3/h. Therefore, the total required air volume and treated heat quantity are 38,500 m^3/h · 41.9 kW with ΔT = -2 °C and 22,300 m^3/h · 54.5 kW with ΔT = -5 °C. An HVAC designer is able to evaluate energy consumption of ΔT = -2 °C and ΔT = -5 °C by taking into consideration the fan efficiency and the efficiency of the heat source system. In the general, the amount of outside air volume and heat load are often calculated by assuming the presence of perfect mixing. It is possible to design and examine more aspects of the system according to the actual situation based on the findings of this research.

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Figure 3-4: Linear slot diffusers with vertical flow under ΔT = -5 °C. a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 2-slot diffusers: cases a.1–6 and cases B.1–8, b) $T_{0.25}/L$ vs. temperature effectiveness with 2-slot diffusers: cases a.1–6 and cases B.1–8, c) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 4-slot diffusers: Cases A.7–12 and cases B.9–17, d) $T_{0.25}/L$ vs.

temperature effectiveness with 4-slot diffusers: cases a.7–12 and cases b.9–17

Figure 3-5: Linear slot diffusers with vertical flow under ΔT = -2 °C. a) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 2-slot diffusers: Cases A.13–18 and cases B.18–23, b) $T_{0.25}/L$ vs. temperature effectiveness with 2-slot diffusers: Cases A.13–18 and cases B.18–23, c) $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis) with 4-slot diffusers: Cases A.24–29 and cases B.24–29, d) $T_{0.25}/L$ vs. temperature effectiveness with 4-slot diffusers: Cases A.24–29 and cases B.24–29

Figure 3-6: Velocity and temperature fields in the ADPI measurements (cases A.1, A.3, A.5, and A.12)

Figure 3-7: Adjustable blade grill with different blade angles (cases B.30–36, cases B.37–43 and cases B.44–50, and ADPI from a previous study (Liu et al., 2016). a), c) and e): $T_{0.25}/L$ vs. air change effectiveness (left side y-axis) and ADPI (right side y-axis), b), d) and f): $T_{0.25}/L$ vs. temperature effectiveness

Figure 3-8: Adjustable blade grill (0° horizontal adjustment) with different exhaust locations (cases B.30–36, cases B.51–57, cases B.58–63, cases B.64–71, and cases B.72–78). a) $T_{0.25}/L$ vs. air change effectiveness, b) $T_{0.25}/L$ vs. temperature effectiveness

Figure 3-9: Linear slot diffusers with vertical and horizontal flow. a) $T_{0.25}/L$ vs. air change effectiveness with 2 slots: cases B.1–8 and from Chapter 2, b) $T_{0.25}/L$ vs. temperature effectiveness with 2 slots: cases B.1–8 and from

Chapter 2, c) $T_{0.25}/L$ vs. air change effectiveness with 4 slots: cases B.9–17 and from Chapter 2, d) $T_{0.25}/L$ vs. temperature effectiveness with 4 slots: cases B.9–17 and from Chapter 2.

Figure 3-10: Adjustable blade grills with different deflector adjustments (cases B.30–36, cases B.37–43, and cases B.44–50). a) Air change rate vs. air change effectiveness, b) air change rate rate vs. temperature effectiveness

Figure 3-11: Linear slot diffusers with vertical flow under the same heating load (ΔT = -5 °C and ΔT = -2 °C). a) air change effectiveness; b) temperature effectiveness

Figure 4-1: Experimental setup of a test room

Figure 4-2: Tested diffusers

Figure 4-3: Vertical variances in percentile

Figure 4-4: Local air change effectiveness distribution (cases 38 and 42)

Figure 4-5: Local air change effectiveness distribution (cases 95 and 99)

Figure 4-6: Local air change effectiveness distribution (cases 154 and 157)

Figure 4-7: Horizontal and overall standard deviations in percentage

Figure 4-8: Correlation of temperature and air change effectiveness

早稲田大学　博士（工学）　学位申請　研究業績書

(List of research achievements for application of doctorate (Dr. of Engineering), Waseda University)

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