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Published in:
Building and Environment

DOI:
10.1016/j.buildenv.2021.108641

Published: 01/02/2022

Document Version
Peer-reviewed accepted author manuscript, also known as Final accepted manuscript or Post-print

Please cite the original version:
Wu, Y., Mustakallio, P., Kosonen, R., Kaukola, T., Chen, J., Liu, H., & Li, B. (2022). Experimental study of five different VAV air terminal devices under variable heat gain conditions in simulated office and meeting rooms. *Building and Environment*, *209*, Article 108641. https://doi.org/10.1016/j.buildenv.2021.108641

VSI: AQV_RoomVent2020

Experimental study of five different VAV air terminal devices under variable heat gain conditions in simulated office and meeting rooms

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Abstract

Demand control of the air distribution systems is crucial to the satisfactory thermal environment under variable heat gain conditions in an energy efficient manner. In this study, the airflow patterns, ventilation performance and local thermal comfort have been experimentally investigated in mock-up meeting and office rooms with a variable air volume (VAV) system. The Active Diffuser, which had a constant inlet air velocity, was compared with some commonly used diffusers: Rectangular; Swirl-Radial; Swirl-Compact and Multi-Nozzle Diffusers. In the test, there were four different heat gain levels: high heat gain (46 W/m^2) in a meeting room setting; high (46 W/m^2), medium (24 W/m^2) and low (11 W/m^2) heat gain in an office room setting, respectively. The corresponding specific supply airflow rates in the high, medium and low heat gain cases were 4.3 L/s m^2, 2.1 L/s m^2, 1 L/s m^2, respectively. Visualized figures showed the effect of VAV on airflow patterns were quite different among different diffusers. The indexes including air velocity, ventilation effectiveness, effective draft temperature, draught rating and so on were therefore analysed. By using the Active Diffuser, the airflow distribution and most of these indexes were more uniform and more stable under the VAV. Although the Rectangular, Swirl-Compact, Swirl-Radial and Multi-Nozzle Diffusers might perform better in one or two cases regarding some indexes, their airflow patterns and performances of indexes were unstable under the VAV. Thus, the Active Diffuser was recommended for the VAV system for the application where heat gains vary significantly.

Keywords: variable air volume; diffuser; air distribution; ventilation performance; local thermal comfort; draught.

1. Introduction

The building systems generally take account for about 30% of the final energy consumption along with 30% of the global CO_2 emissions [1], in which the ventilation systems consume a major part of energy especially in non-residential buildings [2]. Except for the consideration of energy consumption, the ventilation system is responsible for indoor climates [3], and to guarantee healthy and comfortable indoor air quality and thermal comfort [4, 5]. A good air distribution method of the ventilation system that could create a uniform indoor thermal environment [6] is essential to the thermal comfort of occupants which can significantly affect working productivity and physical well-being of occupants [7, 8].

To deal with the varied indoor heat gain conditions in an energy-efficient way, the variable air volume (VAV) air-conditioning systems are widely in use today [9]. Ventilation airflow rates are typically controlled based on occupancy, indoor air quality and/or room air temperature in office rooms with VAV systems [10]. The target of the system operation is to maintain a good thermal environment for occupants and stable ventilation air distribution in varying conditions [11]. However, the airflow patterns of air distribution with variable air volume (VAV) modes can be varied from the designed condition [12, 13], which might cause energy wastes and local thermal discomforts such as temperature gradient [14] or draught [15]. Thus, it is important to guarantee the performance of air distribution systems with VAV modes under different heat gain levels.

A recent novel solution to better fit with the VAV system is the active supply diffuser, where the performance is based on the constant static pressure in the ductwork [16]. The Active Diffuser could be used to maintain the constant air throw with an adapting geometry, i.e. controllable slot openings according to the airflow rates [17]. It is designed to guarantee the similar thrown pattern with all air flow rates supplied [18]. It is noticed that the Coanda effect works well when the air velocity is high enough with cold supply air. In the certain point, the buoyance is stronger and the jet separates from the ceiling and drops down to the occupied zone, which increases risk of draught. Unlike typical VAV air terminal devices, in which the inlet air velocity changes as the airflow rate varies due to the constant inlet supply area, the adaptive geometry in the Active Diffuser can maintain a constant inlet air velocity to ensure a draught-free performance for VAV systems [19]. However, the comprehensive assessment of the Active Diffuser for VAV systems is insufficient. Therefore, one aim of this study was to evaluate the advantages and disadvantages of an Active Diffuser for VAV systems.

The mixing ventilation (MV) could perform well under the designed condition [20]. To meet the varied cooling requirements, there are many methods to improve local thermal comfort and energy efficiency proposed by different studies [21], such as integrated mixing ventilation with personalized ventilation [22, 23], or with ceiling radiant panels [24] and so on. Compared to other air distribution methods, such as displacement ventilation [25] and stratum ventilation [26, 27], local thermal discomfort due to draught was less significant in the occupancy zone if the MV systems were used appropriately. But some recent studies [28, 29] reported draught risk under MV which might be caused by varied throw patterns of diffusers under lower heat gain conditions than in the peak cooling load conditions However, the effects of varied airflow rates and different types of diffusers on the results of local thermal comforts were not clearly identified in the previous studies. Challenging situations for VAV system operation with ceiling diffusers can happen with small ventilation airflow rate [30] when cool air jets may detach from the ceiling and drop down to the occupied zone and further it may cause draught [31]. Also with a big airflow rate and interaction with jets and thermal plumes may cause discomfort for occupants [32]. Therefore, it is essential to compare the local thermal comfort of different types of diffusers under varied heat gains.

In this study, the airflow patterns, air distribution and local thermal discomfort have been experimentally investigated in mock-up meeting and office rooms with a variable air volume (VAV) system. Different types of diffusers were compared with each other: Active; Rectangular; Swirl-Radial; Swirl-Compact and Multi-Nozzle Diffusers. In the study, there were four different heat gain cases (11- 46 W/m^2) with corresponding typical varied specific supply airflow rates (1 - 4.3 L/s m^2) in the commercial buildings. The novelty of this research was to analyze the performance of air distribution with typical varied airflow rates under different heat gains levels with different types of ceiling diffusers. Thus, the knowledge on the performance of different diffusers under VAV systems could be presented, and the operation ranges of different diffusers in VAV system under variable heat gain conditions could be recommended.

2. Methodology

2.1 Thermal chamber for variable air volume (VAV) system analysis

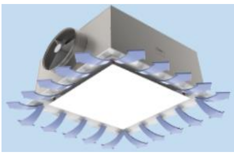
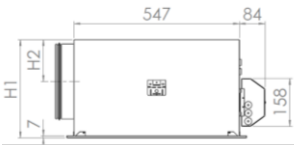
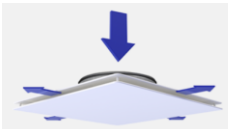
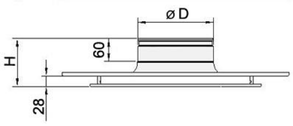
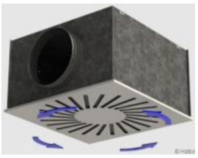
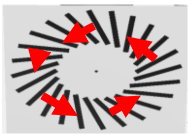
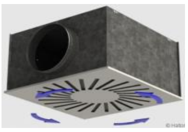
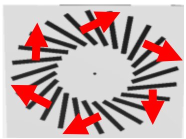
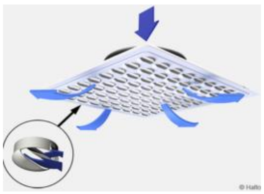
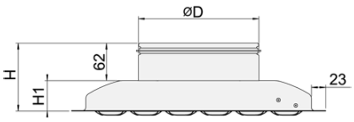
The experimental tests were conducted in a thermal chamber located at the Kausala factory of the Oy Halton Group Ltd in Finland. The chamber is a full-scale test room setup that could be used to simulate typical office or meeting rooms. The dimensions of the chamber were 6.1 m x 4.4 m x 2.7 m (L x W x H), with a floor area of 26.6 m^2. The original chamber length was 6.5 m and it was reduced with light wall built with Mdf board. The suspended ceiling, the floor and the walls except an observation window side were thermally insulated. The climate chamber is located inside other chamber where the surrounding air temperature was constant 25°C during the experiment.

The full-scale test room was shown in Figure 1.Two similar diffusers of the VAV system were installed close to the central line of the room (x=1.6 and 4.4 m, y=2.2 m). The air was exhausted through a round ceiling valve with a diameter of 0.3 m close to the observation window and at the middle of the length.

2.2 Studied diffusers

Five different types of diffusers were tested and compared in this study, as listed in Table 1, including product name, figure and dimensioning of diffusers [33]. There was one Active Diffuser (JDS – A) with constant inlet velocity by adjusting supply slot according to supply airflow rates (constant static pressure 35 Pa in the ductwork). The inlet velocities of the other four diffusers were varied according with airflow rates. The JDA – R was Rectangular Diffuser, with the height of inlet 28 mm and wide 300 mm. Two models of Swirl Diffusers with wide of 495 mm, namely compact (JTH – C) and radial (JTH – R) were studied. The JSC – M was a Multi-Nozzle Diffuser with wide of 420 mm.

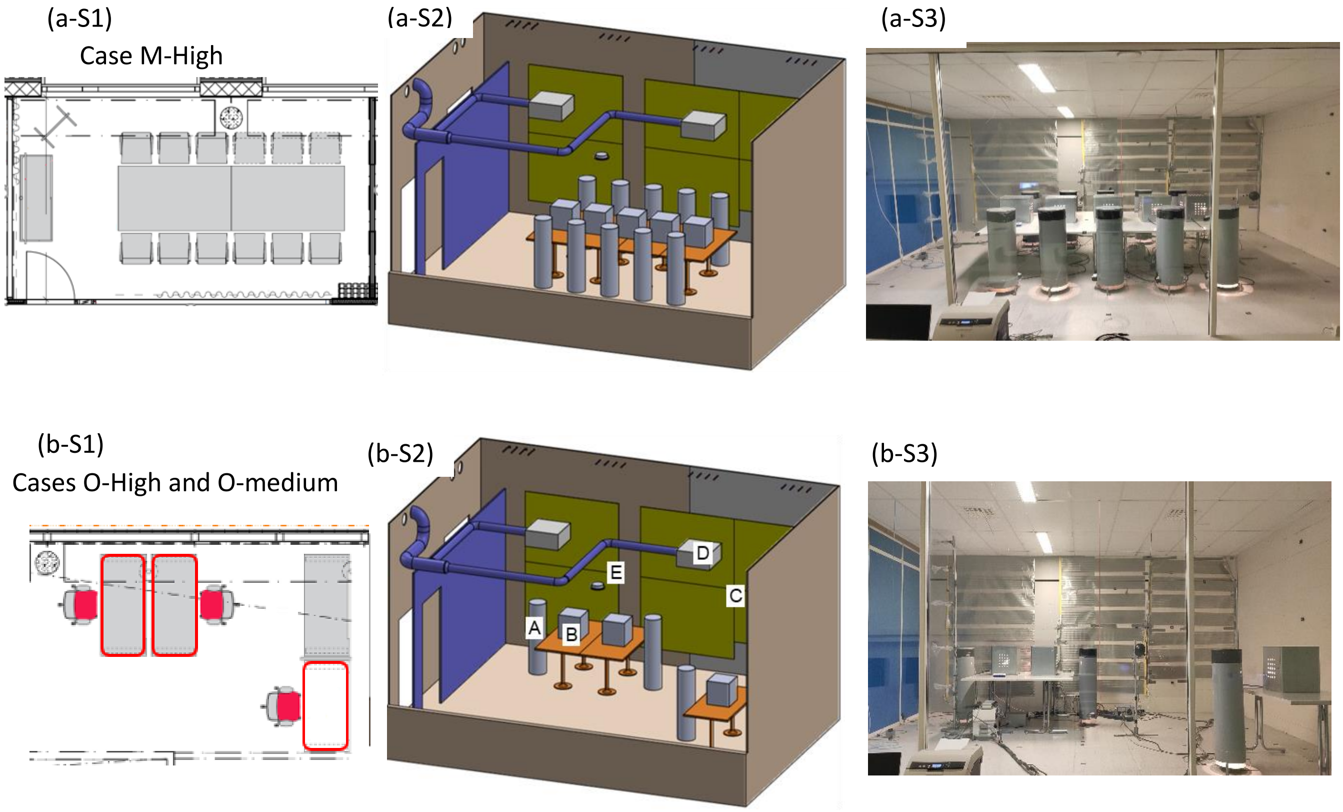
Table 1. Product information of the five diffusers.

No.	Product name	Product illustration	Dimensions (mm)	Notes
Active	JDS - active ceiling diffuser		 D=160; H1=276; H=114;	Constant initial supply air velocity with different airflow rates
Rectangular	JDA - rectangular ceiling diffusers		 H=120; D=125; W=300	
Swirl-Compact	JTH - swirl ceiling diffuser (Compact)		 D=160; W=495	Inward radial round supply pattern with adjust strips in slots
Swirl-Radial	JTH - swirl ceiling diffuser (Radial)		 D=160; W=495	Outward radial round supply pattern with adjust strips in slots
Multi-Nozzle	JSC - multi-nozzle ceiling diffuser		 H=112; H1=50; D=160; W=420	

Note: W- wide; D – diameter; H -height

2.3 Studied cases

The effect of the varied air flow rates of diffusers on the air distribution and local thermal comfort was studied in the office room and meeting room setup, as **Figure 1** shows. The studied cases as shown in **Figure 1-S3** mimic the real building floor as shown in **Figure 1-S1**. The cylindric dummy (height of 1.2 m and diameter of 0.3 m) was used to represent a seated occupant. Light bulbs located inside the manikin were used to simulate the heat of the human body [34]. The simulated computer (0.4×0.4×0.4 m) was also an iron sheet with light bulbs in it. The heat gain of the simulated occupant and computer were adjustable.



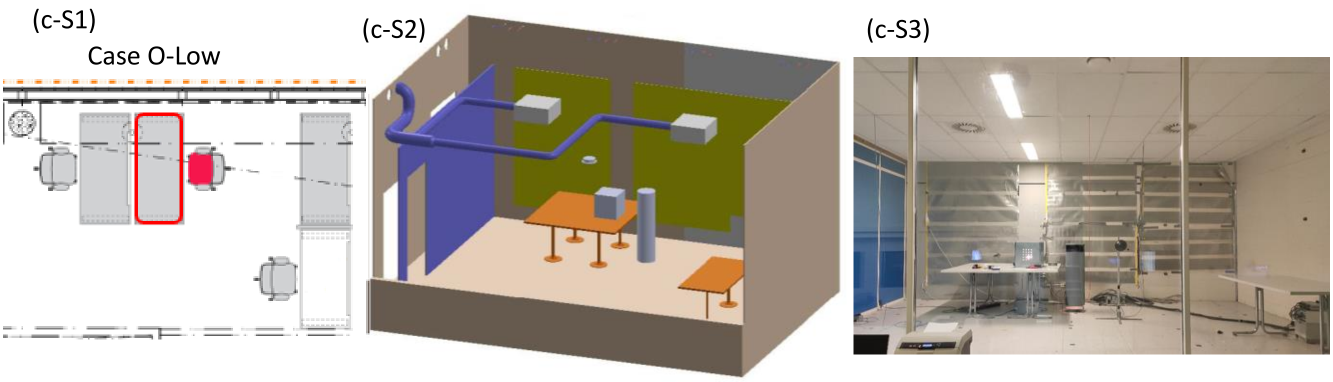


Figure 1. The appearance of room setting in cases: (a) M-High; (b) O-High and O-Medium; (c) O-Low. (From left to right are: S1) building floor layout in real buildings; S2) full-scale test setups and S3) photos of the experimental layout. Heat gains presented in office room setup (b-S2): A) occupants, B) computers and C) simulated warm windows. Also supply air ductwork and D) diffusers, and E) exhaust air valve.)

Table 2. Room components of the cases with different heat gains

Case No.	Room types	Appearance of room setting	Occupant		Computer		Warm windows
			No.	Size	No.	Size	Yes/No
M-High	Meeting	Figure 1a	10	H=1.2m; D=0.3 m	5	0.4×0.4×0.4 m	No
O-High	Office	Figure 1b	3		3		Yes
O-Medium	Office	Figure 1b	3		3		No
O-Low	Office	Figure 1c	1		1		No

The average room air temperatures were all set at 25°C [35] in different cases. Four different heat load conditions with different air volumes were studied. Detailed information on each case was listed in Tables 2 and 3. In the meeting room, there was a table (3.2×1.6 m) in the middle of the room with five computers (5×75 W) on it and 10 simulated occupants (10×75 W) along with it. The heat load of a computer is 75 W in meeting room setting, because it might be just used for a short time during a meeting. The total heating gain (included lighting 112 W) of the meeting room (M-High) was 1237 W (or 46 W/m^2) in this study. In-office room, conditions with three different heat gains were studied, i.e. the high heat gain case (O-High), the medium heat gain case (O-Medium) and the low heat gain case (O-Low). The total heating gain of the office in the case of O-High was the same as M-High 1237 W (46 W/m^2), but the heat gains were located non-uniformly in the room space. Electric heating foils installed in the wall with the power of 600 W were used to simulate the solar heat gain from windows in the O-High case. Besides that, three tables (1.6×0.8 m) with three computers (3×100 W) and three manikins (3×75 W) were used in the O-High case. In the medium heat gain case (O-Medium) of the office setting with the total heat gain of 637 W (or 24 W/m^2), there were also three computers (3×100 W) and three manikins (3×75 W) but there was no solar heat gain. In the low heat gain case (O-Low) of the office setting with the total heat gain of 287 W (or 11 W/m^2), there was only one computer (100 W) and one manikin (1×75 W) on and along with the table close to the middle of the room. The heating load of lighting was 112 W in all cases.

The supply airflow rates for high, medium and low heat load cases were 115 L/s (4.3 L/(s·m^2) or ACH=5.7), 59 L/s (2.2 L/(s·m^2) or ACH=2.9), 27 L/s (1.0 L/(s·m^2) or ACH=1.3), respectively. The studied airflow rate range was based on the recommended ventilation airflow rates in EN 16798-1 standards Category II for office room and conference room with very low and low building material emissions (1.1 – 4.2 l/(s·m^2) floor). Cooling need requiring larger supply airflow rates than 4.3 L/(s·m^2) in modern offices was not studied. If the cooling demand is higher in offices it is assumed to cover with radiant panels in real buildings [36, 37].

Table 3. Room inputs of the cases with different heat gains

Case No.	Heating load (W)					(W/m^2)	Supply airflow rate		ACH
	Occupant	Computer	Solar	lighting	Total	per m^2	L/s	L/(s·m^2)	h^{-1}
M-High	10×75	5×75	0	112	1237	46	115	4.3	5.7
O-High	3×75	3×100	600	112	1237	46	115	4.3	5.7
O-Medium	3×75	3×100	0	112	637	24	59	2.2	2.9
O-Low	1×75	1×100	0	112	287	11	27	1.0	1.3

Note: Temperature of supply air was constant 16.0°C; Target room air temperature was 25.0°C; ACH-Air exchange rate.

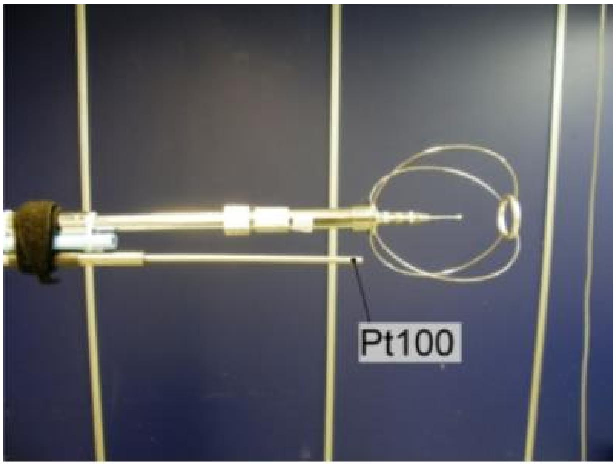
2.4 Measurement instruments and test points

Airflow of supply and exhaust air was measured with orifice plate with differential pressure transmitter Furness Controls FCO33 (accuracy < 5 % of reading). Both airflow rates could be controlled by adjusting fan speed (“rotating speed”). The air pressure difference in the duct before the diffuser inlet was measured with a Mikor TT470S micro-manometer with accuracy of ±0.6 Pa (range 0–199.9 Pa). The room globe (operative) temperature was measured, using PT100 sensor inside black bulb with an accuracy of ±0.2 °C, in a location close to the middle of the room (0.8 m above the floor), but away from the airflow path, as the reference point. More equipment characteristics was listed in Table 4.

Table 4. Ranges and precision of instruments in climate chamber study

Brand/model	Equipment	Variables	Range	Accuracy
Sensor/PT100	Balk bulb	Globe temperature	−50 to 100 °C	±0.2 °C
Furness Controls/FCO33	Orifice plate	Airflow rate	/	5 % of reading
Mikor/TT470S	Micro-manometer	Air pressure	0–199.9 Pa	±0.6 Pa
Sensor/PT100	Temperature sensors	Air temperature	−50 to 100 °C	±0.2 °C
Sensor/HT412	Air velocity probes	Air velocity	0.05 to 1.00 m/s	±0.02 m/s

The parameters at the test sample points were measured using a multichannel low Velocity Thermal Anemometer (Measurement system HT-400, manufactured by “Sensor”). Air velocity and turbulent intensity were measured using the Omni-directional velocity probes (HT412). The accuracy of air velocity is ±0.02 m/s for the measurement range of 0.05 to 1.00 m/s. The air temperatures of test points were measured using PT100 sensors positioned close to each velocity probe as Figure 2 shows. The systems were interfaced a computer for the data record. The highest data recording frequency is more than 5 times / s. But in this study, readings are three minutes average values. And the data were waited to be stable with at least 5 mins interval between two adjacent tests.



a b

Figure 2.(a) The air velocity probe (HT412) and air temperature sensor (Pt100); (b) the height of sensors

Six HT412 anemometers were attached to a vertical pole at heights of 0.1 m, 0.6 m, 1.1 m, 1.7 m, 2.2 m and 2.6 m [38] as Figure 2b shows, represent the height levels of different body parts and the upper space of the room, respectively. There were altogether forty-five sampling lines in total, in which twenty-five sampling lines were in the occupied zone (at least 0.5 m away from the wall), as **Figure 3** shows. Overall, there were 270 test sample points in the room and 100 test sample points in the occupancy zone which was defined in ASHRAE standard 55 [38] as the space with the height from 0.1 m to 1.8 m and at least 0.5 m away from the walls. In case of M-High, the test points in the middle of the large table were unable to measure and, thus not presented. It is noticed that the closest horizontal distance of some test points to the heat sources of dummy was about 0.1-0.2 m (i.e. e5 in office setting and c5, d5 and e5 in meeting setting). Considering the effect of the convective boundary layer flow of the manikin [39, 40], the abnormal data above 1.1 m of these test points was excluded from the data analysis of the local thermal comfort in this study.

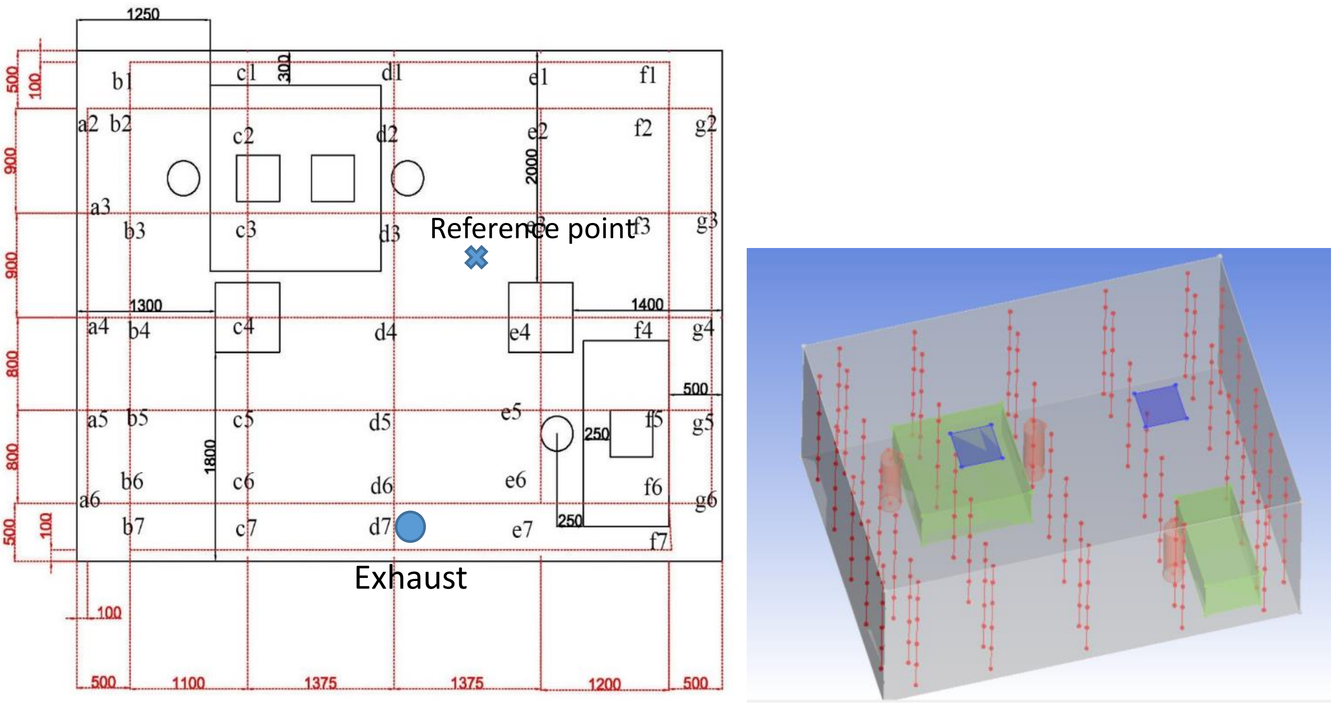


Figure 3. The locations of sampling points (unit: mm)

2.5 Experimental procedure

First, to make sure that the thermal environment was in the designed condition for each test session, the physical parameters (such as the supply airflow rate and room air temperature) for the next day test where set in the previous evening. Before each test, it was controlled that the thermal parameters were in the steady-state conditions for at last two hours.

Second, the airflow patterns of the five ceiling diffusers were visualized by using smoke (Look Solutions regular fog), as Figure 4 shows. The smoke was injected into the duct of supply air by a fog generator system (Martin Magnum 850, UK). As a result, the supply airflow could be visualized from the inlet of diffusers and guarantee that the overall performance was as confirmed before the test

Finally, the values of room air temperature and air velocity at different locations and height levels were measured and recorded. The data along one sampling line at six different heights were measured simultaneously and then moved to the next sampling line. The measuring period was 3 minutes and a five-minute interval was applied to ensure the stabilization of sensor readings [38].

2.6 Evaluation indices

The PMV [41] is useful to evaluate the thermal comfort conditions as comprehensive indexes which including six physical parameters. Besides that, the following indices are considered in this study.

2.6.1 Ventilation effectiveness for cooling

The ventilation effectiveness is the index used to characterize the ability of a ventilation system to exchange the air [42]. The ventilation effectiveness for cooling was calculated using formula as follows:

$$\epsilon_c = \frac{T_e - T_S}{T_a - T_S}, \tag{1}$$

where, T_e is the exhaust air temperature, in °C; T_s is the supply air temperature, which is constant 16°C in this study; T_a is the average room air temperature of occupancy zone, in °C; $T_s{}'$ is the exhaust air temperature, e, in °C. A higher value of ϵ_c means the heat gain in the occupancy zone was removed more effectively by the ventilation system

2.6.2 Effective draft temperature (EDT)

To evaluate the uniform of indoor thermal environment, the effective draft temperature (EDT, °C) was defined as follows [43]:

$$EDT = (T_x - T_a) - 8(v_x - v_a) \tag{2}$$

where T_x = local air temperature, °C; Ta = average room temperature, °C; v_x = local air velocity, m/s; v_a = average room air velocity, m/s. Lower values of EDT mean the indoor thermal environment was more uniform.

2.6.3 Draught rating (DR)

Based on Reference [44], draught rate (DR) was calculated according to ISO 7730 [45] by using the following equation to predict the percentage of persons dissatisfied by draught.:

$$DR = (34 - T_X)(v_x - 0.05)^{0.62}(0.37 v_x TI_x + 3.14) \tag{3}$$

where T_x is the local air temperature, °C; v_x is the local air velocity, m/s; TI_x is the local turbulence intensity, %.

TI is determined using the equation as follows:

$$TI = \frac{v'}{v} \times 100\% \tag{4}$$

where *v'* is the standard deviation of velocity fluctuation.

The Air Diffusion Performance Index (ADPI) is determined as follows [38]:

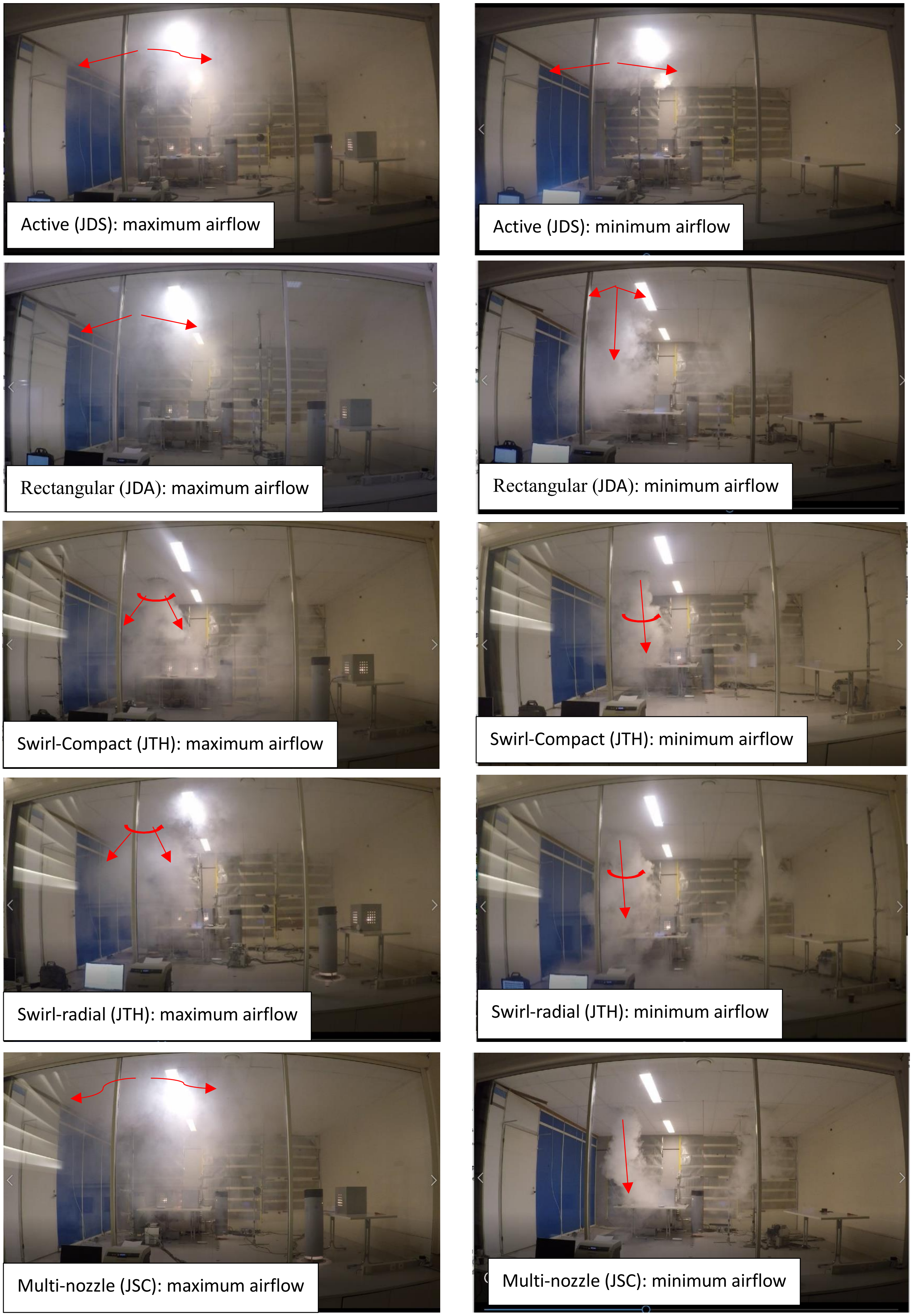
$$ADPI = n/N \times 100\% \tag{5}$$

According to Equation (5), the ADPI (%) is defined as the ratio of comfort points (n) to the total measuring points in space. The comfort points are defined by the criteria that the DR <10% or 20% for categories of thermal environment A or B, respectively [45], and the local air speed (V_x) is lower than 0.35 m/s.

3. Results and discussion

3.1 Smoke visualization and static air pressure before diffusers

In order to visualize the airflow pattern of different diffusers under variable airflow rate, the smoke tests were performed in this study, as **Figure 4** shows. The flow patterns from the supply inlet were captured to reflect the airflow characteristics using active and other commonly used diffusers. Under maximum (115 L/s) airflow rate, the airflow from the supply inlet was released horizontally along with the ceiling by using the Active (JDS) and Rectangular (JDA) Diffusers (the smokes could be clearly seen through the lighting on the ceiling). As a function of time, the airflow then dispersed and went down along the walls. While the air jet from the inlet by using Swirl-Compact Diffuser was more directly delivered downward to the head level and mixed with the room air in the occupancy zone, which had a higher draught risk and could be treated as a reference of worse case. However, under minimum (27 L/s) airflow rate, the airflow patterns by using the active and commonly used diffusers were quite different. For the Active Diffuser (JDS), the airflow from the supply inlet was still moved horizontally along with the ceiling, due to the constant initial velocity of the jet (see Table 3). For the Rectangular Diffusers (JDA), the airflow pattern from the supply inlet was turned to downdraft instead of horizontally dispersed. The reason for that is low momentum flux of the jet due to the constant airflow outlet area in the diffuser, and the buoyance affecting the throw pattern (see Table 6). The performance of the other diffusers studied including the Swirl-Radial, Swirl-Compact and Multi-Nozzle Diffusers was similar. To improve the clarity of visualized airflow patterns in this study, we have also provided the 5 second videos (Gif) with all cases of smoke visualization in the supplementary material "Appendix A. Supplementary data1. Table S1: The visualized airflow patterns (the smoke spread out at about 5th -10th s) from different supply diffusers".



Active (JDS): maximum airflow

Active (JDS): minimum airflow

Rectangular (JDA): maximum airflow

Rectangular (JDA): minimum airflow

Swirl-Compact (JTH): maximum airflow

Swirl-Compact (JTH): minimum airflow

Swirl-radial (JTH): maximum airflow

Swirl-radial (JTH): minimum airflow

Multi-nozzle (JSC): maximum airflow

Multi-nozzle (JSC): minimum airflow

Figure 4. The visualized airflow patterns (about 10th s after the smoke started to spread) under maximum (115 L/s) and minimum (27 L/s) airflow rate (Case O-High and O-low) from different supply diffusers

Table 5 shows the static duct air pressure (Pa) before the diffusers using different diffusers during the tests. The static air pressures from the Active Diffuser (JDS) were constant within 34.8-36.5 Pa in all cases, while the air pressures from the Rectangular Diffuser were higher in the cases with high airflow rate (36.0 pa) and much lower in the cases O-Low (2.0 pa), when the control damper of the plenum box of VAV terminal unit is used for air flow rate control

Table 5. Static air pressure (Pa) of the duct using different diffusers during the tests.

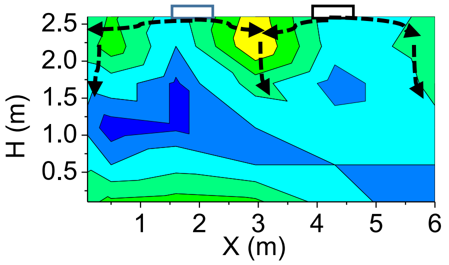
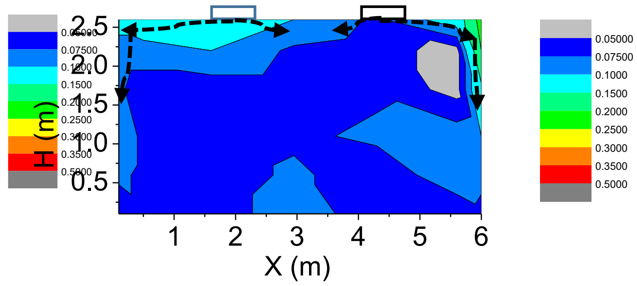
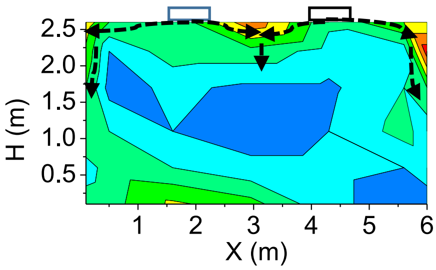
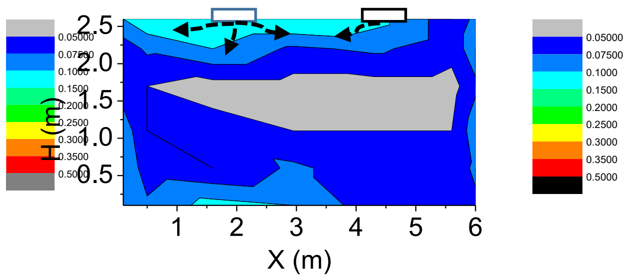
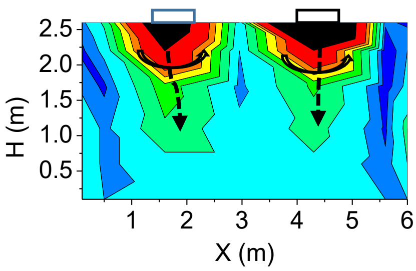
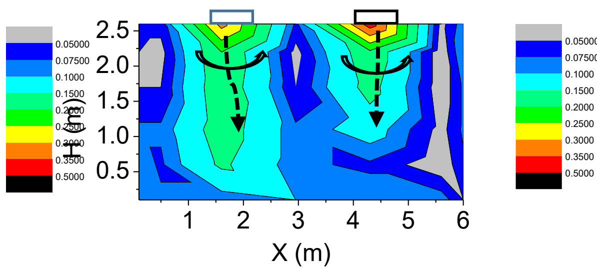
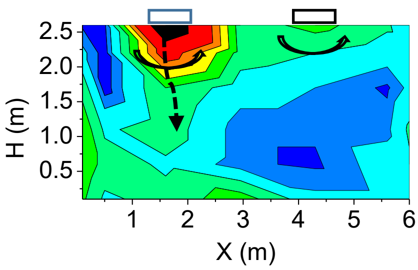
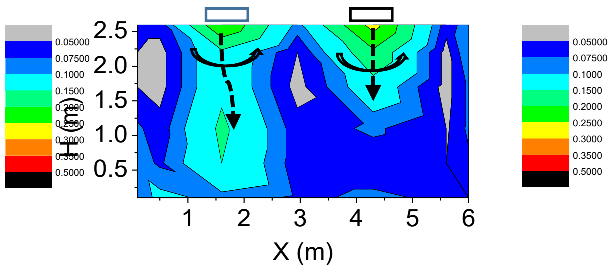
Diffusers	M-High	O-High	O-Medium	O-Low
Active	36.5	36.0	34.8	35.0
Rectangular	36.0	36.0	9.6	2.0
Swirl-Compact	7.0	6.3	1.8	0.5
Swirl-Radial	7.0	7.0	1.8	0.5
Multi-Nozzle	28.7	28.7	8.0	1.5

3.2 Air velocity distribution

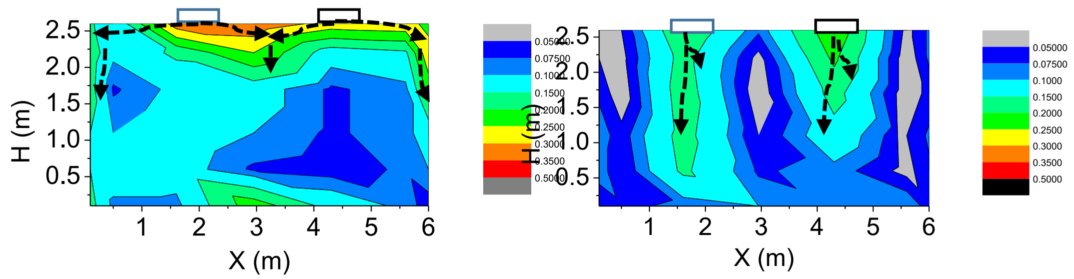
Based on the test data from the experiments, the air velocities measured below the central line of the two diffusers were visualized with different colors in the plane by using the function of plot-contour-color fill in Origin 9.0, as listed in **Table 6**. For the Active and Rectangular Diffusers, the inlet airflow was supplied horizontally along with the ceiling and went down along the walls in all cases with different heat gains and airflow rates, as the same as the smoke tests indicated in **Figure 4** . When the airflow rates were 1.0 L/(s·m^2) in cases O-Low, the air velocities were lower than 0.20 m/s in the most areas of the room including the ceiling. When the airflow rates were 4.3 L/(s·m^2) in cases O-High, the air velocities were obviously higher (about 0.20 m/s) when the airflow went along in the middle of the central line, the walls and the floor than other areas. Generally, the air velocities were more uniform using the Active Diffuser than the Rectangular Diffuser, due to the supply air velocities from the Active Diffuser were constant with all air flow rates. While the supply air velocities from the Rectangular Diffuser were much higher in the cases with high airflow rate and much lower in the cases with lower airflow rate.

The airflow from Multi-Nozzle showed different patterns with different airflow rates: In cases with higher airflow rates, the airflow patterns were similar with the Active and Rectangular Diffusers, i.e. more airflow dispersed along with the ceiling due to higher air pressure (28.7 Pa); In case O-Low, the air pressure from the diffuser was so slow (1.5 Pa) that the airflow was directly dropping down due to buoyancy effect of the inlet air velocity. The air pressure from the Swirl-Compact and Swirl-Radial Diffusers was much lower than other diffusers, thus the airflow of them was more likely to be directly delivered to the occupancy zone. The air velocities at the upper area of the room were much higher than the occupancy zone and that of other diffusers, which were increased up to more than 0.50 m/s in the cases with higher airflow rates from Swirl-Compact and Swirl-Radial Diffusers. Compared to Swirl-Compact, the supply airflow from the Swirl-Radial dispersed with a wider range and slightly lower air velocities in the head level. These visualizations using the test data of air velocities in **Table 6** also agreed with the smoke tests in **Figure 4**. To these who might concern more cases and planes, we also have provided the information with all cases in the supplementary material in "Appendix A. Supplementary Data 2. Table S2: Contour plot based on the measured air velocities in the plane below the central line of the two diffusers" and "Table S3: Contour plot based on the measured air velocities in the plane at the height of 1.1 m."

Table 6. Contour plot based on the measured air velocities in the plane below the central line of the two diffusers

	O-High (max. airflow rate)	O-Low (min. airflow rate)
Active		
Rectangular		
Swirl-Compact		
Swirl-Radial		

Multi-Nozzle



Note: **Plane of the test point (Z axle: height (H))** **Legend: colorful bar of air velocity (m/s)**

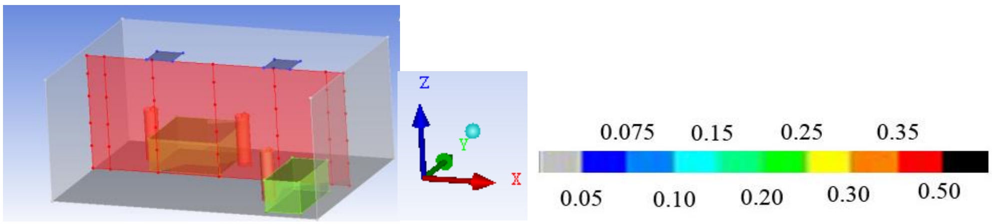


Figure 5 shows cumulative distributions (%) of air velocity (0.01 m/s bin) in the occupancy zone using different diffusers. The air velocities of sampling points were mainly distributed in the range of 0.05 -0.20 m/s in high heat gain cases and 0.01 -0.15 m/s in lower heat gain cases for most diffusers, respectively. By using the Rectangular and Swirl-Compact (i.e. Swirl-C) Diffusers, there were more numbers of the sampling points with air velocities higher than 0.20 m/s in the case M-High, which were mainly located in the areas close to corners and the areas along the table (and down to the floor), respectively. It is also noticed that, the air velocity distributions of Rectangular and Swirl-Compact Diffusers were highly affected by the different heat gain cases and less concentrated to the average values in high heat gain cases. While, the Swirl-radial Diffuser seem better than the Swirl-Compact Diffuser. The performance of Multi-nozzle Diffuser seems similar with the Active Diffuser and better than other common-used diffusers in high heat gain cases. Generally, the distribution of the air velocities using the Active Diffuser was more stable among different heat gain cases and concentrated to the average values (i.e. more uniform) than most other diffusers in high heat gain cases.

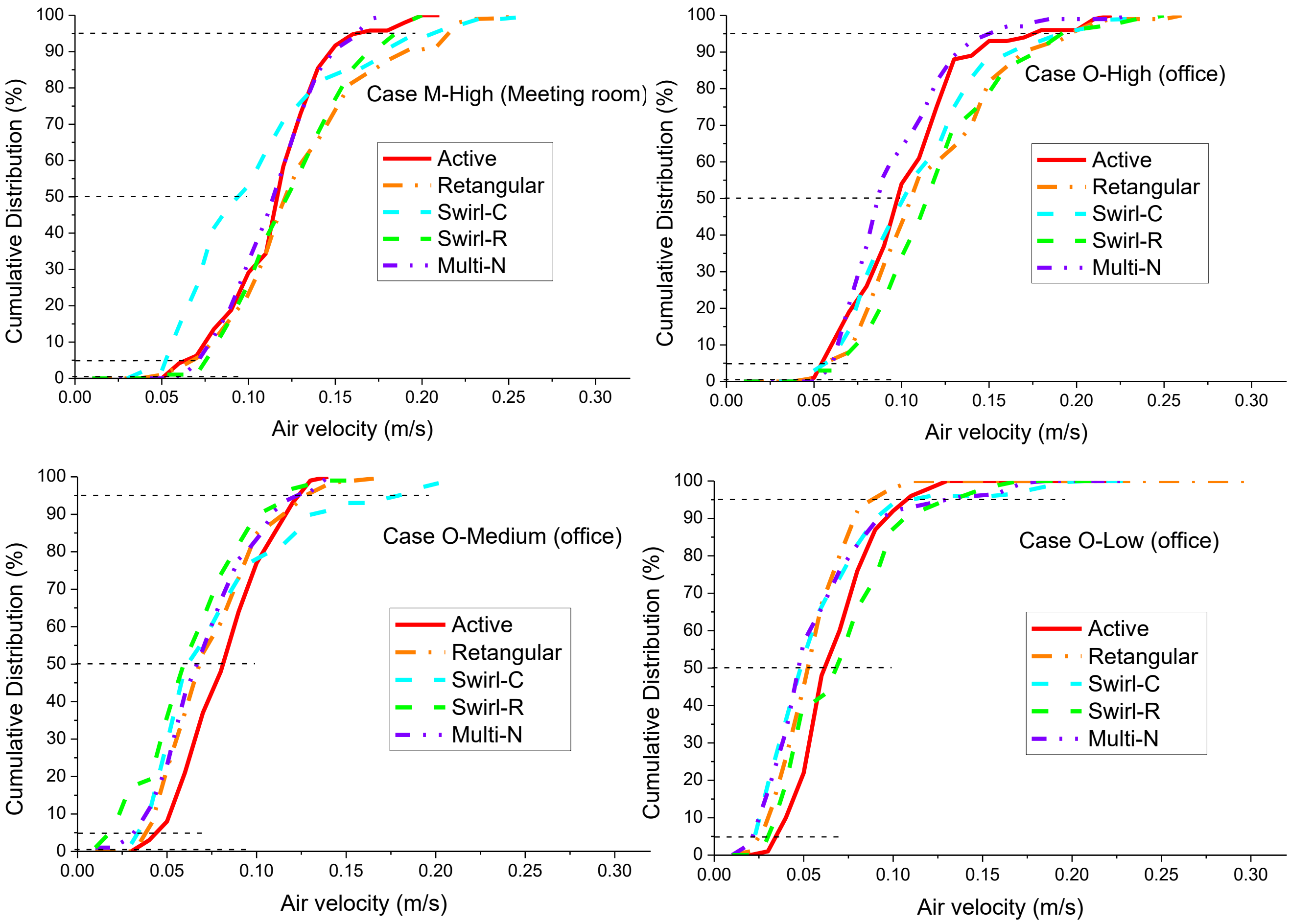


Figure 5. The cumulative distributions (%) of air velocity (0.01 m/s bin) in occupancy zone.

3.3 Air temperature and ventilation effectiveness for cooling

The ranges of air temperature in the occupancy zone using different diffusers were shown in **Figure 6**. Generally, the air temperatures in all cases ranged from 24°C to 25°C. In the case M-High, the air temperatures were more non-uniform using the Swirl-Radial Diffuser (i.e. Swirl-R) than other diffusers. In the case O-High, the air temperature was lowest using the Swirl-Compact Diffuser than other diffusers, due to the inlet airflow from the Swirl-Compact was directly delivered to the occupancy zone. In the cased O-Medium and O-Low, the air temperatures were lower using the Swirl-Radial and Multi-Nozzle Diffusers (i.e. Multi-N) than other diffusers, for there was relatively more cooling air directly dropped down due to low inlet air pressure from these diffusers.

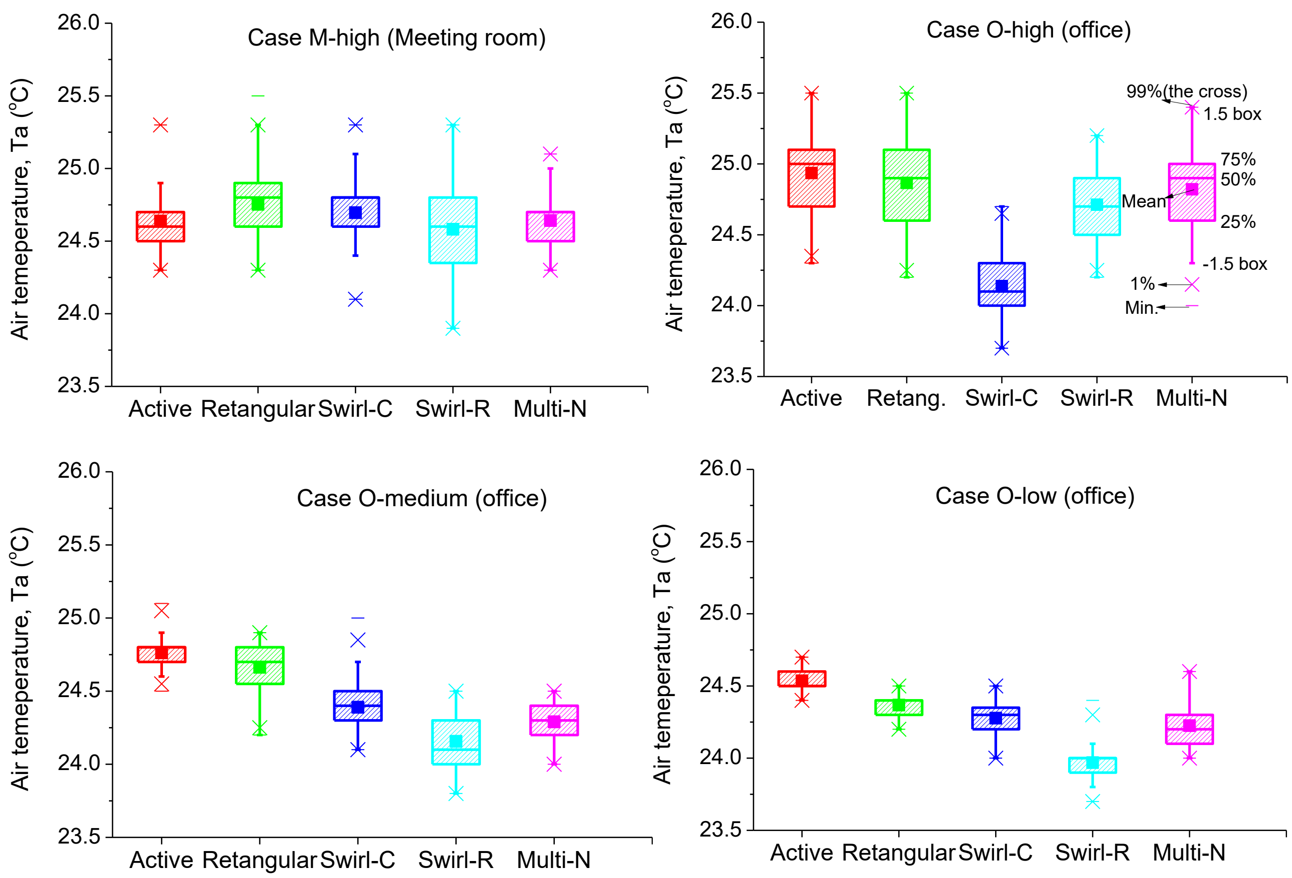


Figure 6. The ranges of air temperature in occupancy zone.

Figure 7 shows the ventilation effectiveness for cooling of four heat gain cases using different diffusers, which was calculated according to Equ. (1). By the results of T test (matched pair), the ventilation effectiveness of Rectangular (P =0.017) and Active (P =0.022) Diffusers were significant lower than the Swirl-Compact. But the ventilation effectiveness between any other two diffusers were not significantly different. The reason could be that under the experimental setting in this study, the cooling air from Swirl-Compact Diffuser was delivered directly to the occupancy zone. The diffusers with highest to lowest Standard Deviations of ventilation effectiveness for cooling were: Swirl-Radial, Multi- Nozzle; Rectangular; Swirl-Compact and Active Diffusers. That means the ventilation effectiveness of the Active Diffuser was more stable under varied conditions than other diffusers due to its constant inlet air velocity with varied airflow rates, which could benefit for the accurate control of the thermal environment using a VAV system.

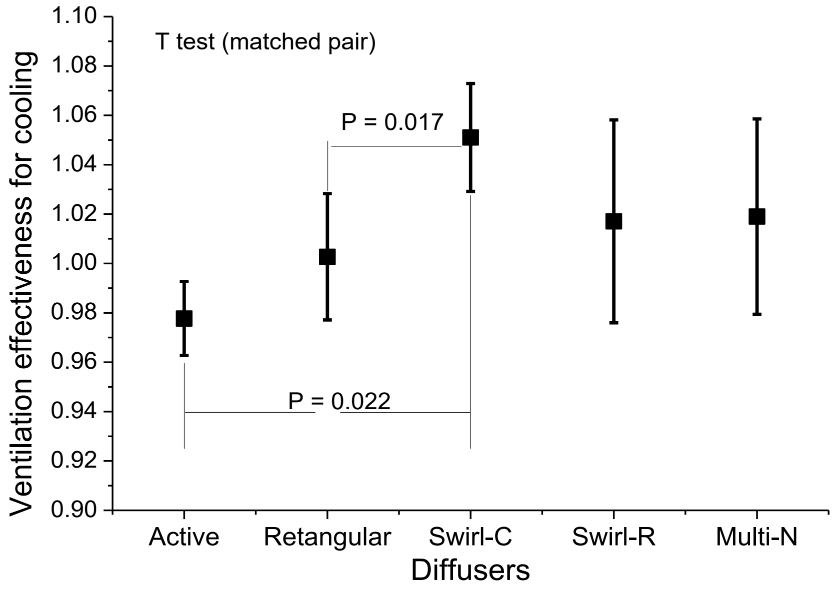


Figure 7. The ventilation effectiveness for cooling of four heat gain cases (mean±standard deviation)

3.4 Overall and local thermal comfort

3.4.1 Overall thermal conditions

Because the PMV is not suitable for evaluating local thermal comfort, it only served as a comprehensive physical index reflex heat stress but not indicate the actual thermal comfort of occupant in this study. The air velocities, air temperatures and radiant temperature are the value of all measured points in occupancy zone. The air relative humidity is set to be 50%, the activity level is 1.1 met and clothing level is 0.6 clo [38]. From the Table 7, we can see the mean PMV values are all within comfort range (-0.5,0.5). Thus, the overall thermal comfort was satisfied in all cases.

Table 7. PMV values of the sampling points in occupancy zone in different cases

Diffusers	M-High	O-High	O-Medium	O -Low
Active	-0.12±0.10	0.01±0.15	0.00±0.06	-0.05±0.04
Rectangular	-0.12±0.14	-0.04±0.18	0.02±0.09	-0.09±0.03
Swirl-Compact	-0.07±0.15	-0.25±0.14	0.12±0.11	-0.13±0.06
Swirl-Radial	-0.13±0.14	-0.08±0.16	-0.18±0.09	-0.23±0.06
Multi-Nozzle	-0.11±0.10	-0.00±0.13	-0.14±0.07	-0.15±0.07

Note: mean ± standard deviation

3.4.2 Effective draft temperature (EDT)

Figure 8 shows the ranges of EDT in the occupancy zone using different diffusers, which were calculated according to Equ. (2) in Section 2.6.2. A smaller deviation of EDT means the thermal environment of the occupancy zone was more uniform. Generally, the deviation/range of EDTs using the Active Diffuser was the smallest in all cases. The deviation/range of EDTs using the Rectangular Diffuser was generally larger with a high airflow rate in the cases O-High and M-High than other diffusers, due to its higher air velocities in the occupancy zone (**Figure 5**). The performances of Swirl-Radial and Swirl-Compact Diffusers were generally in the middle rank. The performance of the Multi-Nozzle Diffuser was better than the Swirl Diffusers in the cases M-High and O-Medium, but worst among all diffusers with low airflow rates in case O-Low due to the more significantly dropping cold air below the Multi-Nozzle Diffuser (**Table 6**).

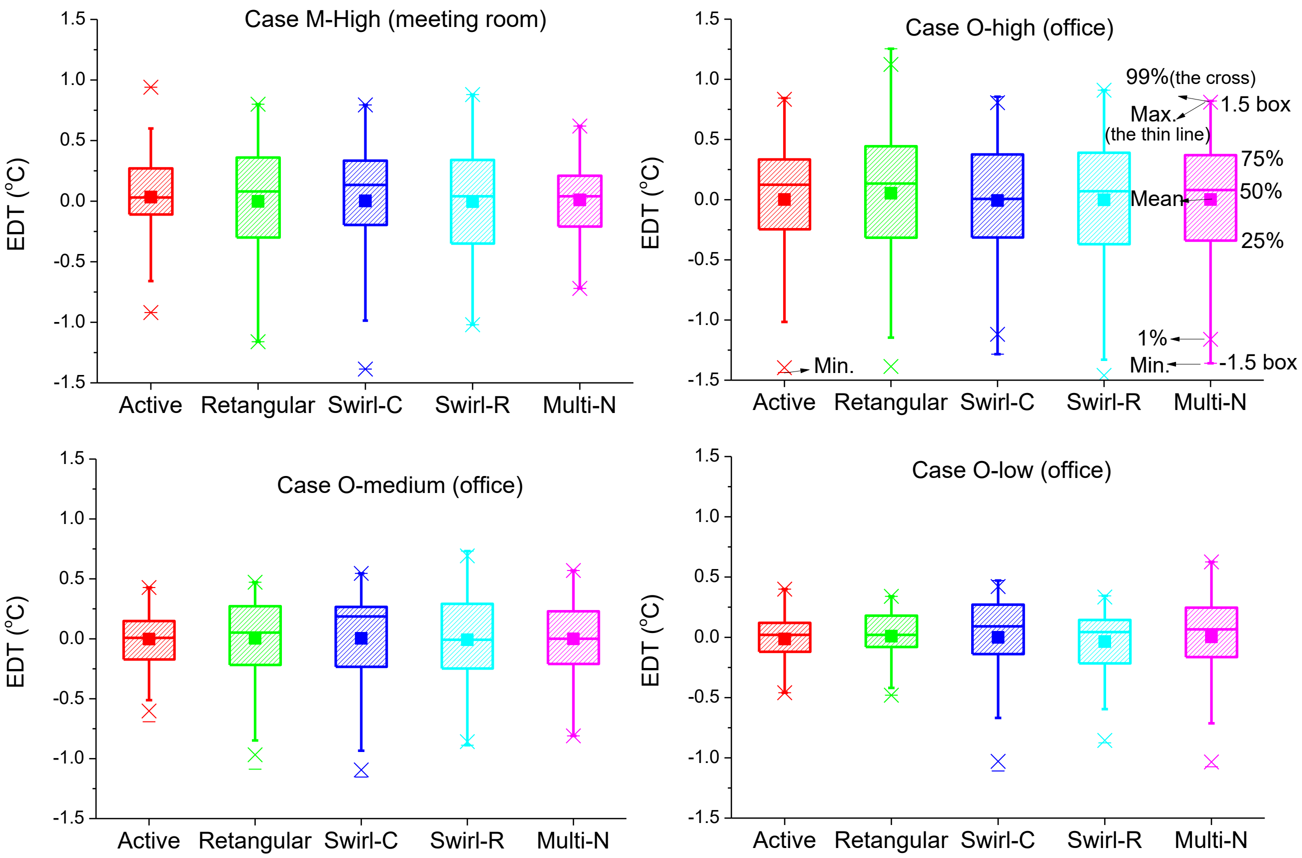


Figure 8. The effective draft temperature (EDT) in occupancy zone.

3.4.3 Draught rating (DR)

Draught discomfort usually occurs in locations where exits high local air velocities or/and undesired low local air temperature[38]. Thus, the draught rating (DR) and the percentage of local comfort sampling points (i.e. Air Diffusion Performance Index, ADPI) in the occupancy zone were calculated according to the Equations (3) - (5) in Section 2.6.3. **Figure 9** shows the cumulative distributions (%) of draught rating (DR) bins (1%) in the occupancy zone and the ADPI in the occupancy zone. The ADPI indicates the percentage of sampling points which are within the local thermal comfort criteria, that were DR <10% for categories of thermal environment A or DR <20% for categories of thermal environment B, according to ASHRAE standard 55 [38] and ISO 7730 [45]. Taking categories of thermal environment B for example, mostly all diffusers meet the requirement of DR <20%, expect very few sampling points of Swirl-compact and Swirl-radial Diffusers in the high heat gain cases. Thus, the criteria of DR <10% for categories of thermal environment A was used in this study for comparison purpose. Generally, as it could be expected, the ADPI was decreased with the increase of airflow rates, the values of ADPI were all higher than 95% in case O-Low but lowest to 69% in cases O-High and M-High. Using the Active and Multi-Nozzle Diffusers, the ADPI was generally higher than other diffusers in all cases, with the values of ADPI higher than 80% in meeting room case (M-High) and higher than 90% in office room cases with different airflow rates. The ADPI using the Rectangular Diffuser was generally the lowest in the cases with high airflow rate (M-High and O-High), in which there was only about 70% of area was within the acceptable draught rate range. The ADPI of the Swirl-Compact Diffuser was higher in case M-High and lower in O-High than the Swirl-Radial Diffuser, the main reason was the air temperature below the Swirl-Compact Diffuser was colder (**Figure 6**) than other diffusers due to its airflow pattern (**Table 6**).

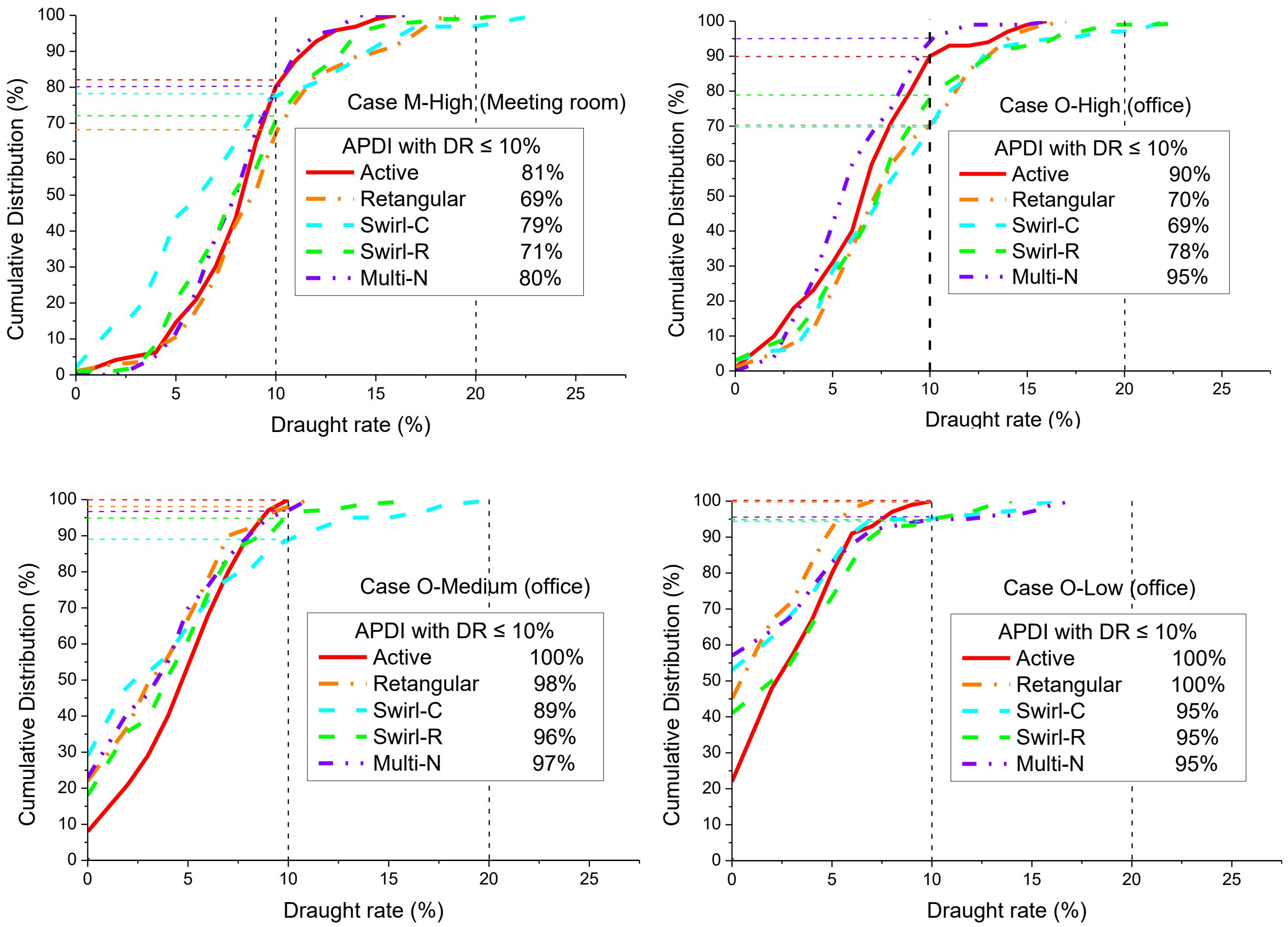


Figure 9. The cumulative distributions (%) of draught rating (DR) bins (1%) in occupancy zone

4. Summary and limitations of this study

The uniformities of the indexes including air velocity (AV), effective draft temperature (EDT) and draught rating (DR) in each heat gain case were evaluated by the standard deviations of their values in each heat gain cases. The rank from lowest to highest standard deviation when using different diffusers was marked with star symbol from five stars to one star, as Table 8 lists. It is noticed that Active Diffuser got three to five stars with these indexes in all cases. That means the room thermal environments created by Active Diffuser were not only more uniform but also more stable regarding the indexes of AV, EDT and DR than other diffusers in VAV conditions. It is also noticed that, the performance of Swirl-Compact Diffusers was the worst. While, the Swirl-radial Diffuser seem better than the Swirl-Compact Diffuser. The performances of Rectangular and Multi-Nozzle Diffusers seem similar with the Active Diffuser and better than other common-used diffusers in low and high heat gain cases, respectively. But the performances of Rectangular Diffuser were worse in high heat gain cases and the performances of Multi-Nozzle Diffuser were worse in low heat gain case, respectively.

Table 8. The summary of the uniformities of the indexes in each heat gain case when using different diffusers

Diffusers	M-High			O-High			O-Medium			O-Low		
	AV	EDT	DR	AV	EDT	DR	AV	EDT	DR	AV	EDT	DR
Active	****	****	****	****	***	****	*****	*****	*****	*****	****	****
Rectangular	**	**	**	*	*	***	**	***	****	****	*****	*****
Swirl-Compact	*	*	*	***	****	*	*	*	*	**	**	**
Swirl-Radial	***	***	***	**	**	**	****	**	**	***	***	***
Multi-Nozzle	*****	*****	*****	*****	*****	*****	***	****	***	*	*	*

Note: AV-air velocity; EDT-effective draft temperature; DR – draught rating; "*****" is the best and "*" is the worst performance.

The stabilizations of other two indexes including ventilation effectiveness (VE) and Air Diffusion Performance Index (APDI, DR≤10%) with VAV conditions were also evaluated by the standard deviations of their mean value in different heat gain cases. The rank from lowest to highest standard deviation when using different diffusers was marked with star symbol from five stars to one star, as Table 9 lists. The Active Diffuser also performed better than other diffusers regarding the stabilizations of VE and APDI (DR≤10%). The stabilizations of other common-used diffusers were worse regarding to either VE or APDI, which less than three stars.

Table 9. The summary of the stabilizations with VAV conditions of different diffusers

Diffusers	VE	APDI
Active	*****	****
Rectangular	***	*
Swirl-Compact	****	**
Swirl-Radial	*	***
Multi-Nozzle	**	*****

Note: VE-ventilation effectiveness; APDI - Air Diffusion Performance Index with DR≤10%; "*****" is the best performance.

In this study, the size of Swirl Diffusers was the same as Active and Muti-nozzle Diffusers (their diameter is all 160 mm). However, the static pressure was much lower for Swirl Diffusers (0.5-7.0 Pa) than others (1.5-36.0 Pa) in all tested cases as Table 5 shows. This may mean that the supply air velocity of Swirl Diffusers was very low and therefore buoyancy forces prevailed in the air flow. It may also suggest that the size of the swirl diffuser was not properly selected or the air volume flow in this study was lower than the minimum recommended flow rate for this type and size of the diffuser. Thus, the performance of the swirl diffuser might be better with a smaller size.

The cylindrical dummy and cube were used as heat sources for this study, because the main purpose was focused on the different performance of diffusers to the thermal environments of the whole room in a VAV system, the microenvironments of the heat sources were not focused. In a study by Zukowska, D. et al [40], it was found that thermal plume of a cylindrical dummy generated much more concentrated plumes compared to a siting person. And the effect of body movement [46] was also not considered in this study. Similarly, it seems that the proportion between convection and radiation of computer may not be properly reproduced by a cube. What's more, the outside weather was fluctuant [47].These factors might have some effects to the results of this study. These need to be studied in the further.

5. Conclusions

In this study, the airflow patterns, air distribution and local thermal discomfort of five different diffusers have been experimentally investigated in mock-up meeting and office rooms under a variable air volume (VAV) system. The main findings are summarized as follows:

(1) The airflow patterns were quite different among different diffusers, which could be seen from the visualizations of smoke paths and air velocity distributions. The cooling air was dispersed along with

the ceiling, walls and part of the floor using the Active and Rectangular Diffusers under maximum heat gain cases. Under varied supply air flow rates, the airflow distribution was more stable under different heat gain cases and the thermal environment was more uniform by using the Active Diffuser. But the air distribution was much varied using the commonly used diffusers. There is more cooling air was dispersed along with the designed directions in high heat gain cases but when the supply air velocity was low under the minimum airflow rate (27L/s or 1.0L/(s·m^2)) cases, the buoyancy forces prevailed in the air flow dropping the cooling air directly down. The static pressure was much lower for Swirl Diffusers (0.5-7.0 Pa) than others (1.5-36.0 Pa), which may suggest the air volume flow in this study was lower than the minimum recommended flow rate for this size of the Swirl-Compact and Swirl-Radial Diffusers.

(2) Under varied supply air volumes, the thermal environment was more uniform regarding the indexes including air velocity (AV), effective draft temperature (EDT), draught rating (DR)and also more stable regarding these three indexes and other two indexes including ventilation effectiveness (VE) and Air Diffusion Performance Index (APDI, DR≤10%) using the Active Diffuser with constant inlet air velocity. Compared to other common-used diffusers, the performance of Active and Multi-Nozzle Diffusers on local thermal comfort was much better. However, the performances of Multi-Nozzle Diffuser were only good in high and medium heat gain cases but worse in low heat gain cases.

(3) The performance of Swirl-Compact Diffusers was the worst with the size selected in this study, and its performance could be better with a smaller size. While, the Swirl-radial Diffuser seems better than the Swirl-Compact Diffuser in most cases and with most indexes. The performances of Rectangular Diffusers were better in low heat gain case and worse in high heat gain cases. By using the Rectangular and Swirl-Compact (i.e. Swirl-C) Diffusers, there were more numbers of the sampling points with air

velocities higher than 0.20 m/s in the case M-High, which were mainly located in the areas close to corners and the areas along the table (to the floor), respectively.

Thus, the Active Diffuser with constant inlet air velocity is recommended for the VAV system if used appropriately.

Acknowledgement

The study was supported by Business Finland in SmartOtaniemi research project (8199/31/2018), and the 111 Project of China (Grant No. B13041). The author, Yuxin Wu, would like to thank the Chinese Scholarship Council (No. 201806050244) for their sponsorship of a research visiting study at Aalto University in Finland.

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Appendix A. Supplementary data

The following is the supplementary data to this article: