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Original research article

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Experimental comparison of thermal conditions in office rooms: diffuse ceiling ventilation, chilled beam system and chilled ceiling combined with mixing ventilation

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Abstract

Thermal environments generated by diffuse ceiling ventilation (DCV), chilled beam (CB) and chilled ceiling with mixing ventilation (CCMV) systems were compared at different heat gains. Experiments were carried out in two test chambers, in which a double office layout was investigated near heated windows. The heat gain strength was increased from a normal level of 37-40 W/floor-m^2 to a peak load level of 57-64 W/floor-m^2 to observe effects on indoor air thermal conditions and draft discomfort. The target indoor air temperature was 26±0.5°C in the occupied zone. The results show that the air temperature, air speed and draft risk levels were reasonable at the normal load and increased slightly at the peak load conditions. The CCMV system provided the smallest share of locations, in which the draft risk level was higher than 10 %. None of the studied systems achieved the category A defined by EN ISO 7730. However, all the systems fulfilled the demands of category B, except for CB which felt to category C at the peak load conditions by exceeding the draft risk limit in one location. Thermal conditions were at a same level with the studied systems. However, CCMV may reduce slightly the draft discomfort.

Keywords: office, air distribution, convective system, radiant system, draft

Introduction

Indoor thermal conditions and comfort are important factors of human well-being. For instances, ASHRAE 55, EN ISO 7730 and EN 15251 standards recommend the acceptable parameters on thermal comfort (ASHRAE 2017; CEN 2005; CEN 2007). Furthermore, earlier studies have shown that occupants have ranked thermal comfort to be more important factor than the visual comfort, acoustic comfort or even a good indoor air quality (Frontczak and Wargocki 2011). Dissatisfaction with one factor does not necessarily leads to a dissatisfaction with overall comfort (Humphreys 2005). For instances, Arens et al. (2006) have proposed that an overall thermal sensation follows the warmest sensation at head and the coldest sensation of hands and feet in the warm and the cool environments, respectively. Therefore, it is important to explore thermal discomfort locally. However, occupants are usually more sensitive to a local thermal discomfort when the whole body is cooler than neutral and less sensitive to a local thermal discomfort when the whole body is warmer than neutral (ASHRAE 2017). The thermal discomfort can be assessed e.g. by conducting a local draft risk model, which predicts a local discomfort due to draft as a percentage of people dissatisfied (Fanger 1970; Fanger et al. 1988; CEN 2005; ASHRAE 2017). Earlier, thermal comfort has been investigated in the large web-based survey in the USA, Canada and Europe (Zagreus et al. 2004; Frontczak et al. 2012). The authors found that an average office building occupant is usually rather satisfied with a workstation environment. Regardless of that, thermal discomfort has been a common complaint indoors (Melikov et al. 2005; Kosonen et al. 2011; Taleghani et al. 2013).

Thermal discomfort can have many implications on occupants. Melikov and Kaczmarczyk (2012) showed how the indoor air movement improves an acceptability on perceived air quality and freshness of indoor air. The authors found that air movement reduces a negative impact of increased indoor air temperature, relative humidity and pollution level. However, a draft risk may increase when the additional air is supplied to the occupants (Melikov et al., 2002; Melikov et al., 2007a). In a subsequent study, Melikov (2016) emphasized that the advanced air distribution methods can improve thermal comfort and energy efficiency. Fang et al. (2004) proposed that the perception of air freshness and acceptability can be improved when inhaled air temperature and humidity decrease. Wargocki and Wyon (2007) found that reducing room air temperature improves the performance of schoolwork and Lan et al. (2011) stated that the health symptoms and performance get worse when the occupants feel thermally warm. Kosonen and Tan (2004) explained that the productivity loss is significant at the higher indoor air temperature compared to the neutral thermal perception. Those studies greatly justify an importance of discovering the thermal conditions and draft risk in office environments.

Ventilation is an important factor on thermal and healthy indoor environment (Müller et al. 2013; Yang et al., 2019). Generally, the main challenge in air distribution is to provide occupants with clean air for breathing and maintain their thermal comfort. The ventilation also dilutes indoor air and extracts contaminants (Mundt et al. 2004; Sandberg et al. 2019). Seppänen et al. (1999) stated that ventilation rates below 10 L/s per person can be associated with a statistically significant decline in health or perceived air quality outcomes. Altogether, ventilation and indoor climate have been shown to affect significantly on learning and human performance (Wargocki et al., 2002; Wargocki and Wyon, 2007; Haverinen-Shaughnessy et al. 2011; Maula et al., 2017; Carrer et al., 2018). Therefore, ventilation and convective cooling methods are necessary for indoor environments.

Diffuse ceiling ventilation is a convective cooling method, in which an even supply of perforated suspended ceiling penetrates supply air down to the occupied zone instead of local supply openings (Nielsen 2011; Zhang et al. 2014; Zhang et al. 2016; Zhang et al. 2017). It may improve thermal comfort and reduce the draft risk (Jacobs et al. 2008; Jacobs and Knoll 2009; Fan et al. 2013). However, the heat gain strength, the heat gain distribution and the room height may affect the ventilation performance (Nielsen et al. 2015; Nielsen et al. 2017). The findings also indicate that the highest cooling capacity can be reached while the perforated panels cover the entire ceiling and the heat sources are distributed evenly (Rahnama et al. 2019). In addition, the diffuse ceiling ventilation has been found to perform well in the transient conditions (Kristensen et al. 2017). Furthermore, a recent review concluded that the diffuse ceiling ventilation may handle high heat gains without increasing the risk on thermal discomfort (Wu et al. 2018b). The authors emphasized that the system can have an energy saving potential due to a low-pressure drop through the diffuse ceiling.

Thermal radiation asymmetry affects the thermal comfort (Fanger et al. 1985). Consequently, a ventilation rate can be reduced by using a radiant system at high heat gain levels. In this way, the indoor air temperature can also be kept at a higher level than without a radiant system (Le Dréau and Heiselberg 2014). However, a local radiant cooling on a face may improve an acceptability on perceived air quality, but not as much as a convective cooling by raising the air velocity (Melikov et al. 2013b). Furthermore, the combined air and water system may require more adjusted design and control strategy compared to all air system (Saber et al. 2016). An important matter is also to keep the water inlet temperature higher than the surrounding indoor air dew point to prevent a condensation on cold surfaces (Kosonen and Tan 2005; Rhee and Kim 2015). Generally, a radiant cooling system usually promotes the thermally uniform environment and the reduced draft risk (Murakami et al. 2015). However, higher indoor air temperature drifts may provide higher dissatisfaction with radiant systems (Kolarik et al. 2015).

Chilled ceiling is a radiant system which can be combined with an air distribution system to provide fresh air for the occupants (Mustakallio et al. 2017b). In those systems, the additional cooling or heating is usually generated by a water circulation in the system. The chilled ceiling may offer a promising system that improves the ventilation performance and energy efficiency (Schiavon et al. 2012; Schiavon et al. 2015; Wu et al. 2019). Furthermore, a draft risk can also be reduced by conducting a chilled ceiling system, because a small air movement with the radiant cooling system can improve the comfortable sensation (Kitagawa et al. 1999; Corgnati et al. 2009). The main characteristics of chilled ceiling are the cooling capacity, the portion of convection and radiation as well as the dynamic response on changing thermal environment (Novoselac and Srebric 2002). However, the distribution of heat sources and the surface temperatures may affect considerably the panel performance (Fonseca Diaz and Cuevas 2011). Moreover, the chilled ceiling may dampen thermal plumes at a cold surface (Chen et al. 2013).

Active chilled beam system, in turn, is a conventional system comprised of the supplied airflow and a water circulation in the cooling coil. The chilled beam systems are energy-efficient solutions for the indoor environments that require individual zone control and involve moderate humidity sources (Woollett and Rimmer 2015; Nam and Zhai 2019). The chilled beam operates by induction so that the supplied fresh air is mixed with the entering room air in the system (Mustakallio et al. 2017a). Therefore, the chilled-beam may generate larger airflow movement than the chilled-ceiling system. The airflow pattern is not stable and the area of draught risk may variate in time and due to changes in room heat sources (Koskela et al. 2010). In addition, the increased draft risk have been found in the areas where the supply air jets turn down to the occupied zone (Koskela et al. 2012). The active chilled beams may provide better thermal comfort with the symmetrical heat sources than with the asymmetrical ones (Wu et al. 2018a).

In earlier studies, no significant differences have been found between the chilled-beam, radiant-panel and chilled ceiling systems on thermal environment (Kosonen et al. 2011; Duszyk et al. 2011; Bolashikov et al. 2014). Minor differences were found between a convective system and combined convective and radiant systems in office environments (Mustakallio et al. 2016; Mustakallio et al. 2017a). However, earlier studies have found significant effects of heat gain strength and distribution on indoor airflows (Kosonen et al. 2010; Koskela et al. 2010; Lipczynska et al. 2015; Nielsen et al., 2017). Furthermore, a recent study concludes that the radiant systems may provide equal or better thermal comfort compared to all-air convective systems (Karmann et al. 2017), which is in agreement with an another study where the radiant ceiling achieved the most uniform comfort conditions (Le Dréau and Heiselberg 2014). In addition, a radiant cooling system can provide the equivalent thermal comfort at a higher air temperature and decrease the draft risk by operating with smaller airflow than the conventional systems (Rhee et al. 2017). Altogether, Mustakallio et al. (2016) stated that the convective and radiant cooling systems provided only slight differences in office thermal environments and a major factor on indoor airflows was a heat gain distribution. However, less attention has been paid to a local draft risk in the occupied zone and the descriptive statistics of the air temperature, air speed and draft risk.

In this study, the diffuse ceiling ventilation (DCV), the chilled beam system (CB) and the chilled ceiling combined with mixing ventilation (CCMV) were compared. The motivation was to provide new knowledge on the convective and radiant cooling systems, the airflow characteristics and the local draft discomfort in office environment. The draft is defined as an unwanted local cooling of the body caused by air movement (ASHRAE 2017). The novelty of this study lies in comparing those advanced and topical ventilation and air conditioning methods while reducing draft discomfort and maintaining thermal conditions. New knowledge are necessary in the future while developing the intelligent office environments, in which the user centric environment and the advanced systems interact on the changes in thermal environment. The objective was to investigate the effects of heat gain strength on the air temperature, air speed and draft risk in the occupied zone. The study will contribute to the building sciences, indoor airflows, heat gains and air distribution in thermal environments. In the future, the radiant and ventilative cooling methods could play a central role when responding temporal and spatial changes in indoor climate.

Methods

Test chambers

Experiments were carried out in two test chambers under stable thermal conditions. The results of these measurements have been reported earlier by Mustakallio et al. (2016), Mustakallio et al. (2017a) and Lestinen et al. (2018b). This study adds knowledge on differences of thermal conditions and draft discomfort with the studied systems. The CCMV and CB systems were studied in the test chamber 1 with the internal dimensions of 4.1 m (length), 4.2 m (width) and 2.9 m (height) (Figure 1a). The DCV system was studied in the test chamber 2 with the internal dimensions of 5.5 m length, 3.8 m width and 3.2 m height (Figure 1b). A double office layout was located at the perimeter area 0.6 m away from the heated window panels in both test chambers.

The experimental setups were designed according to recommendations by the EN 15251 standard (CEN 2007) in which the target level for indoor environment was category II. Heat gains from sedentary occupants, computers, lighting and solar thermal radiation were modelled. Zukowska et al. (2012) showed that a thermal manikin and a seated dummy may simulate human thermal plumes properly. The target indoor air temperature was 26±0.5°C and the measurement locations were divided equally (Figure 1).

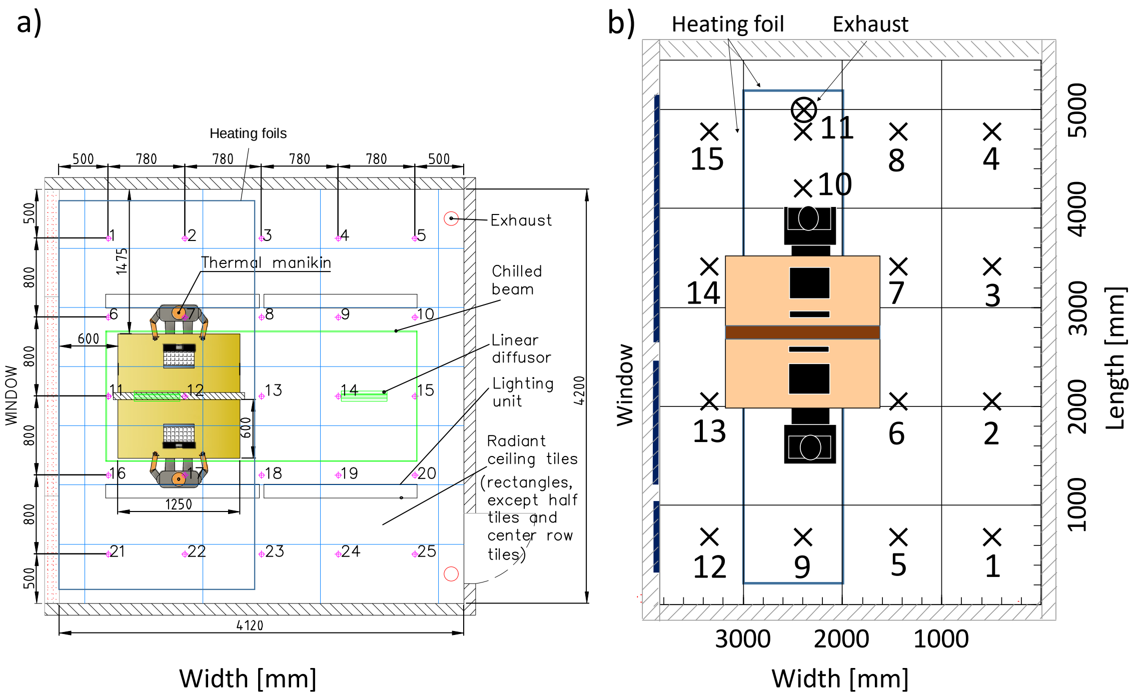

Figure 1. Top-view of test chambers and the measurement locations: a) Test chamber 1 with CB and CCMV systems. The measured locations are marked as 1-25. b) Test chamber 2 with DCV system. The measured locations are marked as 1-15.

Control systems of indoor climate

The CCMV and the CB systems are air-water systems where the majority of the cooling power is covered with water side. The DCV system, in turn, has a ventilative cooling mechanism that controls a thermal environment by adjusting ventilation rate and supply air temperature.

In the CCMV system, the cooling panels were installed into the suspended ceiling at the height of 2.9 m (Figure 2a). The cooling panels covered 77% (13 m^2) of the total ceiling area. The top of suspended ceiling was not insulated. Two linear diffusers of 472 mm x 20 mm (LxH) were installed to distribute supplied air in two directions to the test chamber.

In the CB system, an exposed chilled beam of a total length of 3.0 m and a coil length of 2.1 m was installed at the height of 2.5 m (Figure 2b). In the CB system, the supplied outdoor air is mixed with induced indoor air that is flowing through the cooling coil before the mixed supply air is delivered into the room. The solution has a small space requirements for ducting and a good flexibility in office environments. (True et al. 2007). Also in this method, the supplied mixed airflows were directed in two directions.

In the DCV system, a uniformly perforated suspended ceiling was installed to bring supply air down to the occupied zone (Figure 2c). The perforation rate was 0.50±0.02% of the ceiling area and the nozzles had a diameter of 14 mm each. The nozzle rows closest to the walls were sealed.

In the test chamber 1, the indoor air was extracted from near the corridor wall at the height of 2.9 m by installing one exhaust valve at both corner (Figure 1a). In the test chamber 2, the indoor air was extracted from the suspended ceiling at the height of 3.2 m near the location 11 (Figure 1b).

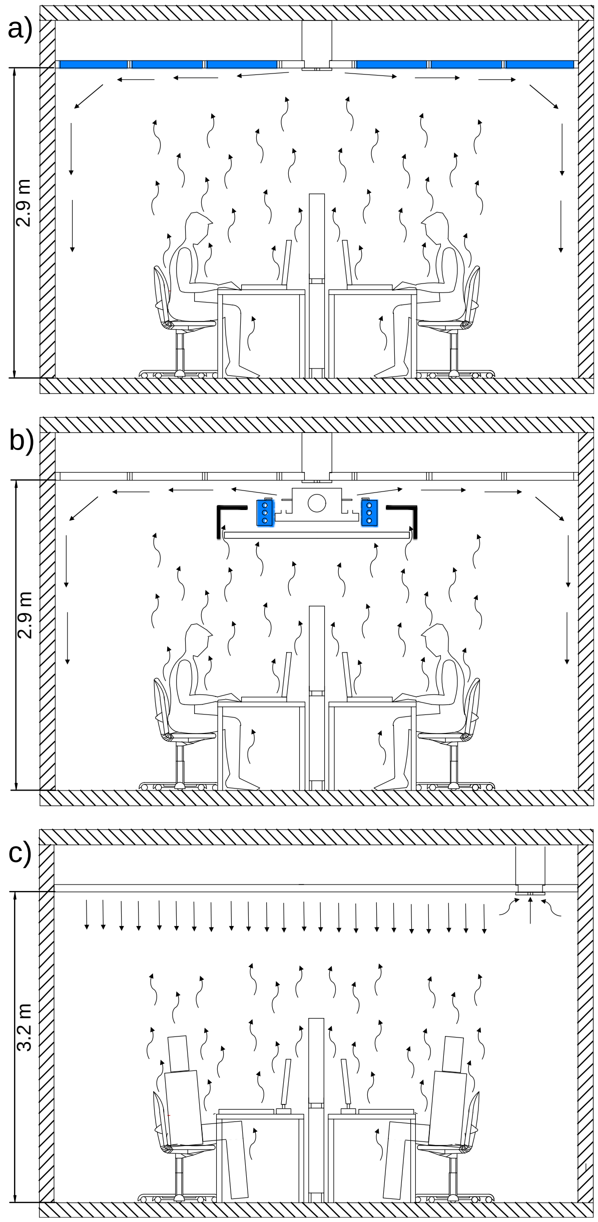

Figure 2. Schematic principle of indoor airflows: a) The chilled ceiling with mixing ventilation system (CCMV). b) The chilled beam system (CB). c) The diffuse ceiling ventilation system (DCV).

Indoor airflow measurement

Low-velocity thermal anemometers with omnidirectional velocity sensor were installed at the heights of 0.1 m, 0.6 m, 1.1 m, and 1.7 m according to recommendation of the standard EN ISO 7726 (CEN 2001).

Measurement period in each measurement point was 5 min for the CB and CCMV systems, and 1 h for the DCV system. Accuracy of the measurement instruments are shown in Table 1.

Table 1. Measuring instruments.

Meter-type	Variable	Accuracy
Low velocity thermal anemometer with omnidirectional velocity sensor 1 (test chamber 1)	Air temperature Air speed (at height 0.1 m-1.7 m)	Air speed: ±0.02 m/s ±1% of reading Air temperature: ±0.2°C STDerr < 10 % up to 1.5 Hz
Low velocity thermal anemometer with omnidirectional velocity sensor 2 (test chamber 2)	Air temperature Air speed (at height 0.1 m-1.1 m)	Air speed: ±0.01 m/s ±2.5% of reading Air temperature: ±0.15°C STDerr < 10 % up to 2 Hz
Low velocity thermal anemometer with omnidirectional velocity sensor 3 (test chamber 2)	Air temperature Air speed (at height 1.4 m-2.9 m)	Air speed: ±0.02 m/s ±1.5% of reading Air temperature: ±0.2°C STDerr < 10 % up to 1.5 Hz
Tinytag data logger	Air temperature	±0.5°C (0…+45°C)
Infrared camera	Surface temperature	±2°C or ±2% of reading
Infrared thermometer	Surface temperature	±1°C or ±1% of reading
Measuring instrument	Pressure difference	±0.3% of reading, lowest ±0.3Pa
Regulation and measuring device	Airflow rate	±5%

The measured physical quantities were the air temperature and the air speed in the occupied zone and the pressure difference at the measuring devices of supplied air and exhaust air for estimating the airflow rate. The draft risk model defined by EN ISO 7730 standard (CEN, 2005) was used. The draft risk is expressed as

$$DR = (34 - t_{a,l})(\bar{u}_{a,l} - 0.05)^{0.62}(0.37 \cdot \bar{u}_{a,l} \cdot Tu + 3.14), \quad (1)$$

where $t_{a,l}$ [°C] is the local air temperature, $\bar{u}_{a,l}$ [m/s] is the local mean air velocity, and Tu [%] is the local turbulence intensity. The model is suitable for the people at light activity, e.g. for a sedentary person. However, the model may overestimate the predicted draft risk at the level of arms and feet. The measurement uncertainty is around 5% percentage points (p.p.) for the draft risk (Melikov et al. 2007b).

Test cases

The test case parameters are shown in Table 2. With the CB and CCMV systems, the supply air temperature was kept at 16±0.5°C. In the water cooling systems, the inlet water temperature was kept at 15±0.5°C and the return water temperature was 2-3°C warmer than the inlet water temperature. With the DCV system, the supply air temperature was kept at 17±0.5°C. One degree difference from 16°C was considered acceptable, because 1°C increase in supply air temperature has earlier been shown to have a small effect on the averaged airflow characteristics (Lestinen et al. 2018a).

In the test chamber 1, the surface temperature of heated water panels simulated a solar heat gain from the windows. The direct solar heat gain was simulated by using 5 electrical heated foils of 2.0 m x 0.75 m (L x W) which were connected in parallel and placed on the floor next to the window wall (Figure 1). The heated window area was 6.3 m^2. In the test chamber 2, a heating foil of 5 x 1 m^2 (L x W) was used and the window area was 7.5 m^2.

Table 2. Test case factors for the usual heat gains and the peak heat gains.

Test chamber 1: CB, CCMV	Usual gains	Peak gains	Test chamber 2: DCV	Usual gains	Peak gains
Floor area, m^2	17.3	17.3	Floor area, m^2	21.0	21.0
Heat gain, W/floor-m^2	37	64	Heat gain, W/floor-m^2	40	57
2 x Thermal manikin, W	156	156	2 x Test dummy, W	176	176
2 x computer, W	130	130	2 x computer, W	96	96
2 x monitor, W	0	0	2 x monitor, W	70	70
Lighting, W	160	160	Lighting, W	116	116
Solar-load at window, W ~	202	404	Solar-load at window*, W ~	381	317
Solar-load at floor, W	0	250	Solar-load at floor*, W	0	420
Total heat gain, W	648	1100	Total heat gain, W	839	1195
Supply airflow rate, l/s,m^2	1.5	1.5	Supply airflow rate, l/s,m^2	3.6	5.2
Supply air temperature,°C	16	16	Supply air temperature,°C	17	17
Room air temperature,°C	26	26	Exhaust air temperature,°C	26	26
Supply air cooling, W/floor-m^2	−18	−18	Supply air cooling, W/floor-m^2	−40	−57
Water cooling, W/floor-m^2	−19	−46	Water cooling, W/floor-m^2	0	0

* The solar load at window was adjusted lower at the peak gains, because the heating foil generated a constant power.

Results

Air temperature

The room air temperature results are summarized in Table *3*. The average air temperature was set to be 26°C in all systems. However, there were slight variations with the average room temperature. At the usual heat gain conditions, the range of local mean air temperatures was largest with the DCV and smallest with the CCMV system. However, the uncertainty of measurements was on the order of ±0.2°C. The standard deviation was at the same level for all the studied systems. At the peak heat gain conditions, the CCMV system had the largest range and the CB system had the smallest range. The standard deviation increased with the heat gain strength. The largest standard deviation was observed with the CCMV and smallest with the DCV at the peak heat gain level. The horizontal air temperature difference between the perimeter area and the corridor area was largest with the CCMV and smallest with the DCV at both the heat gain strengths. At the usual heat gain strength, the horizontal air temperature differences were between 0.5-0.7°C with the studied systems. At the peak heat gain strength, the corresponding differences were 0.8-1.1°C. The vertical air temperature differences were quite a similar level with those systems. The vertical differences were slightly larger with the CCMV and the CB systems than with the DCV system. However, the average differences were within the uncertainty in measurement. Consequently, there were only minor differences in the air temperature profiles between the studied systems.

Table 3. Descriptive statistics of local mean air temperatures in the occupied zone, where *n* denotes the number of locations. Uncertainty in measurement is ±0.2°C.

Double office layout	**Usual heat gain**			**Peak heat gain**		
At heights 0.1 m, 0.6 m, 1.1 m and 1.7 m	CCMV 37 W/m^2 (*n*=87)	CB 37 W/m^2 (*n*=88)	DCV 40 W/m^2 (*n*=60)	CCMV 64 W/m^2 (*n*=88)	CB 64 W/m^2 (*n*=87)	DCV 57 W/m^2 (*n*=60)
Average air temperature [°C]	26.0	25.8	25.6	26.1	25.8	25.9
Standard deviation of air temperature [°C]	0.3	0.3	0.3	0.6	0.5	0.4
Range of local mean air temperatures [°C]	1.3	1.4	1.7	2.7	1.8	2.1
Average of horizontal differences from window side to corridor side at height 1.1 m [°C]	0.7	0.6	0.5	1.1	1.0	0.8
Average of absolute differences between height 0.1 m and 1.7 m [°C]	0.3	0.4	0.2	0.4	0.4	0.3

Air speed

The mean air speed results are summarized in *Table 4*. The spatially averaged mean air speeds were quite a similar with all the studied systems. At the usual heat gain conditions, the local maximum of mean air speeds was 0.23 m/s with the CCMV, 0.27 m/s with the CB and 0.25 m/s with the DCV system. At the peak heat gain strength, the corresponding maximum air speeds were 0.24 m/s, 0.29 m/s and 0.30 m/s, respectively. Hence, the maximum mean air speed was slightly lower with the CCMV system than with those other studied systems. This is rather reasonable, because the indoor water cooling system usually reduces the required ventilation rate if a sufficient amount of clean air has been supplied to occupants. Moreover, although the chilled beam system includes a water cooling system, the total supply air movement increases due to induced secondary airflow, compared to the CCMV system. Obviously therefore, the maximum mean air speed was highest with the CB system at the usual heat gain level. However, at the peak heat gain conditions, the DCV system provided the highest maximum air speed level, but only slightly higher than with the CB system. This is also reasonable, because the DCV system operates without a water cooling system except a cooling coil of air handling unit. This means that the necessary cooling is controlled by the ventilation rate if the supply air temperature is kept constant.

Consequently, a smaller share of high air speed (>0.2 m/s) locations was obtained with the CCMV than with the other studied systems. At the usual heat gain conditions, a considerably larger share was observed with the CB system than with the other ones. However, at the peak heat gain conditions, the largest share was obtained with the DCV system. This shows clearly the effect of increased ventilation rate on air speed levels. In contrast, the DCV system provided also the largest share of low air speed (<0.1 m/s) locations at the usual heat gain conditions, together with the CCMV system. Furthermore, both the CB system and the DCV system produced the largest share of low velocity locations at the peak heat gain conditions. This means that the CCMV system provided the largest share of locations under the both heat gain strengths while the mean air speed level was from 0.1 m/s to 0.2 m/s. At this stage, the smallest share was obtained generally with the DCV system.

The spatial standard deviation ranged from 31 to 42% over the average air speed level, with the CCMV being the lowest and the DVC the highest particularly at the peak heat gain conditions. This indicates that the CCMV system created a slightly more stable airflow field than those other studied systems. Furthermore, the average air speed was lower on the window side than on the corridor side of the room. This indicates a circulating airflow pattern from heat sources to other side of room which was also observed with a marker smoke visualisation (Mustakallio et al., 2016; Lestinen et al., 2018b).

Table 4. The mean air speed in the occupied zone, where *u* denotes the air speed. Uncertainty in measurement is $\pm du = \pm(0.025 + 0.025\bar{u})$ (Melikov et al. 2007b).

Double office layout	**Usual heat gain**			**Peak heat gain**		
At heights 0.1 m, 0.6 m, 1.1 m and 1.7 m	CCMV 37 W/m^2 (*n*=87)	CB 37 W/m^2 (*n*=82)	DCV 40 W/m^2 (*n*=60)	CCMV 64 W/m^2 (*n*=87)	CB 64 W/m^2 (*n*=86)	DCV 57 W/m^2 (*n*=60)
Average air speed [m/s]	0.11	0.12	0.11	0.13	0.13	0.14
Standard deviation of mean air speeds [m/s]	0.04	0.05	0.05	0.04	0.05	0.06
Max. mean air speed [m/s]	0.23	0.27	0.25	0.24	0.29	0.30
Std. dev. / avg [-]	0.36	0.42	0.41	0.31	0.38	0.42
Avg. air speed in window side [m/s]	0.10	0.09	0.11	0.13	0.12	0.13
Avg. air speed in corridor side [m/s]	0.14	0.14	0.12	0.17	0.14	0.16
Difference from window to corridor side [m/s]	0.04	0.05	0.01	0.04	0.02	0.03
Share of locations $u < 0.1$ m/s [-]	0.48	0.41	0.50	0.25	0.30	0.30
Share of locations $0.1 \leq u \leq 0.2$ m/s [-]	0.48	0.48	0.45	0.67	0.60	0.57
Share of locations $u > 0.2$ m/s [-]	0.03	0.11	0.05	0.08	0.09	0.13

Draft risk

The draft risk results are summarized in *Table 5*. The average draft risk was below 10 % in the studied cases except with the CB system at the peak heat gain strength. Furthermore, the standard deviation was smallest with the CCMV, indicating more uniform conditions. Generally, the range of maximum draft risks was 13-18 % at the usual heat gain and 16-21 % at the peak heat gain conditions. The largest maximum of local draft risks was obtained with the CB system and the smallest one with the CCMV system. However, the maximum draft risk was at the same level with both the CB and the DCV systems. Consequently, the results indicate that a classification for thermal environment is the category B defined by EN ISO 7730 standard (CEN, 2005) for the studied systems apart from the CB system at the peak heat gain conditions which falls into the category C.

Although the average draft risk was at a same level with the studied systems, a share of locations, in which the draft risk was lower than 10 %, was the largest with the CCMV system and the smallest with the CB system. However, those shares were more near each other while the heat gain was increased at the peak heat gain level. Generally, the draft risk was smaller in the window side than in the corridor side. This may be explained by the observed airflow patterns from the heat sources to the other side of the room. The results also imply that the radiant cooling may reduce the draft discomfort more efficiently than the convective cooling.

Table 5. The draft risk in the occupied zone. Uncertainty in measurement ±5% (p.p.) (Melikov et al. 2007b).

Double office layout	**Usual heat gain**			**Peak heat gain**		
At heights 0.1 m, 0.6 m, 1.1 m and 1.7 m	CCMV 37 W/m^2 (*n*=87)	CB 37 W/m^2 (*n*=82)	DVC 40 W/m^2 (*n*=60)	CCMV 64 W/m^2 (*n*=87)	CB 64 W/m^2 (*n*=86)	DVC 57 W/m^2 (*n*=60)
Average draft risk [%]	6	8	7	8	10	9
Standard deviation of draft risk [%]	3	5	4	3	5	5
Max. draft risk [%]	13	18	18	16	21	19
Std. dev. / avg [-]	0.49	0.59	0.57	0.42	0.48	0.52
Avg. draft risk in window side [%]	5	4	6	7	7	8
Avg. draft risk in corridor side [%]	8	9	8	11	10	11
Difference from window to corridor side [%]	4	5	2	4	3	3
Share of locations DR < 10% [-]	0.94	0.67	0.85	0.68	0.58	0.65
Share of locations 10% ≤ DR ≤ 20% [-]	0.06	0.33	0.15	0.32	0.41	0.35
Share of locations DR > 20% [-]	0.00	0.00	0.00	0.00	0.01	0.00

Figure 3 shows a color map of the maximum draft risks in the occupied zone at the peak heat gain conditions from the height of 0.1 m up to 1.7 m at each measured location (Figure 1). The results show that the draft risk level was larger in the corridor side than in the window side, in which the heat sources existed. In addition, the results indicate that the CCMV system provided the lowest maximum draft risk levels, actually only slightly above 10% at the workstations. Furthermore, the CCMV system provided slightly larger maximum draft risks in the door corner compared to other regions of the room (Figure 3a). The CB system, in turn, produced the larger maximum rates in the upper-corner on the right (Figure 3b). In the DCV system, the maximum draft risk occurred at the both end-sides and in the middle of the room (Figure 3c). However, only the CB system provided over 20% maximum draft risk at the upper end-side of room (Figure 3b). This may imply that the limited zones in the room can have the category A conditions for thermal environment (EN ISO 7730) e.g. at the workstation area, but it is very difficult to achieve those conditions for the whole occupied zone.

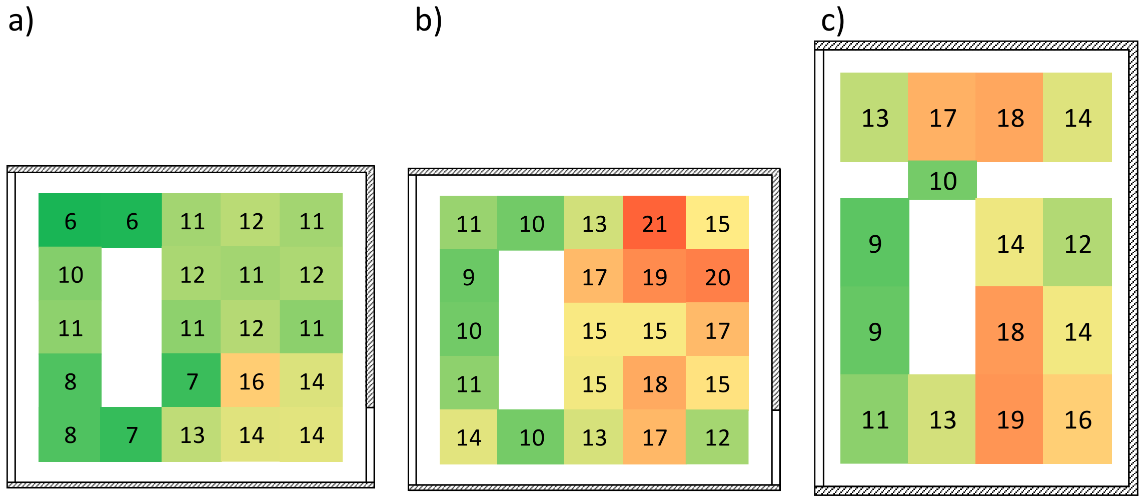

Figure 3. The maximum draft risk (EN ISO 7730) of the heights from 0.1 m to 1.7 m at each location. a) The CCMV system (Mustakallio et al. 2013), b) the CB system (Mustakallio et al. 2013) and c) the DCV system at the peak heat gain conditions.

Discussion

The study adds new knowledge on comparing topical cooling systems that were based on earlier studies by Mustakallio et al. (2016), Mustakallio et al. (2017a) and Lestinen et al. (2018b). New knowledge may inspire new studies on comparing the advanced cooling systems and continue scientific discussion on draft discomfort in the office environments. The measurements were carried out in two separate test chambers. Therefore, the most obvious limitation is the slight differences between those chambers. However, the relative locations of heat sources and cooling systems are rather similar that, in turn, has been shown to have a main effect on room airflow patterns (Kosonen et al. 2010; Nielsen 2011; Koskela et al. 2012). Furthermore, the height levels were similar in the experimental setups. The usual heat gain strength was 8% higher and the peak heat gain level was 11% lower with the DCV system than with those CCMV and CB systems. This means that the draft risk levels may slightly decrease at the usual heat gain conditions and increase slightly at the peak heat gain conditions by using the DCV system, if exactly similar heat gains are obtained (Lestinen et al. 2019). Therefore, the given results may even highlight slightly the use of radiant systems at the high heat gain levels while reducing the draft discomfort.

The results show that the studied systems provided rather similar performance between the perimeter area and the corridor side of the room. Furthermore, the vertical air temperature differences were quite similar. The buoyancy flows and air distribution flows had considerable effects on thermal environment. The convection flows drive the indoor airflows with increasing strength of heat sources. The commonly used workstation layout provided asymmetric heat gain distribution in the both test chambers. The heat sources produced a circulating airflow pattern from the perimeter area to the opposite corridor side. The buoyancy flows pushed also the supplied ventilation jets to the opposite side. This could be prevented by increasing the momentum of ventilation jets. However, this may increase the draft risk correspondingly. However, at 26°C air temperature levels, people may require air movement to improve satisfaction with thermal environment. In addition, thermal radiation affects a human sensation. Thus, the sensation is warmer near a heat source than further away from the source. In any case, there were no significant differences with the studied systems compared to uncertainty levels. The result is in agreement with findings of earlier studies (Mustakallio et al. 2016; Kosonen et al. 2011; Duszyk et al. 2011).

The further studies are recommended on advanced air distribution and cooling systems to improve knowledge and understanding of future user-centric indoor environments. This could highlight the open-plan offices where the interaction of convection flows can be complex due to numerous workstations, the indoor air may flow to the different facades and the thermal radiation can increase on ceiling due to larger view factor. In addition, the volume of indoor environment impact on the indoor airflows generated by ventilation jets and heat sources. The airflow elements may interact differently in large and small environments, because their relative locations may differ and the room height can be different (Lin et al. 2006; Lin et al. 2009; Koskela et al. 2010; Nielsen et al. 2015).

The buoyancy and air distribution flows can be reduced by decreasing the heat sources and by using the low-velocity systems. The possible system is also a perimeter chilled beam installation with a ceiling opening and a plenum cavity that will capture thermal plumes over the window (Woollett and Rimmer 2015). The ventilation rate can be decreased to a smaller level by using the radiant panels under the high heat gain conditions. Furthermore, an integration of radiant panel in chilled-beam design can increase the uniformity of thermal environment (Mustakallio et al. 2017a). In addition, the indoor air temperature can be kept at a higher level than without the radiant panels (Le Dréau and Heiselberg 2014). A recent study shows that the cooling capacity of radiant ceiling can be enhanced by conducting a distributed panel layout instead of a centralized layout (Shin et al. 2019). Moreover, a radiant ceiling cooling may compensate the presence of warm surfaces in the chamber, resulting similar indoor air and operative temperatures (Kazanci et al. 2018; Jia et al. 2018). In addition, the air velocity around the panel surface impact on the cooling performance (Tian et al. 2012).

The results indicate that all those studied methods are appropriate for the given heat gain levels. The draft risk levels were mainly below 20%, which means that the draft risk was acceptable for the thermal environment defined by ASHRAE Standard 55 (ASHRAE 2017). However, none of the studied systems achieved the category A defined by EN ISO 7730 (CEN 2005) even at the usual heat gain conditions. However, since the uncertainty of draft risk can be on the order of 5% p.p. (Melikov et al. 2007b), there is a corresponding uncertainty in the category, because the category boundaries are only 10% apart from each other. Furthermore, the gategory B can also be reasonable for the occupants, because a human thermal sensation is an individual matter. Altogether, the measured results reveal quite a low air speed and draft risk levels and the most of locations had the draft risk below 10% at the usual heat gain level.

The relative locations of supply air diffusers, radiant panels and heat sources, as well as their interaction with each other, could be enhanced by a good design and an investigation on thermal environment including indoor airflow measurements and assessment on local draft risk levels in stable and dynamic conditions. In this way, the A gategory indoor environments can be reached. In the future, the radiant systems are likely an increasing trend when reducing the draft risk levels in office environments, e.g. the diffuse ceiling ventilation and the radiant panels can be combined. Alternative choice could be a displacement ventilation system, in which the low-velocity supply airflow generates the horizontal air layers in the occupied zone and the panels can then be located above the workstations (Hodder et al. 1998; Skistad et al. 2002). The advanced systems can also be combined with the personalized ventilation systems to improve microenvironments around workstations, in which the individualized control and knowledge on airflow interactions are desired matters while improving healthiness, thermal comfort and reducing energy use (Melikov 2004; Lipczynska et al. 2015; Melikov 2016; Xu et al. 2018). However, there are large individual differences between people with regard to the thermal preferences. Thus, an individualized control for a personalized ventilation or a radiant panel systems is recommended (Jaakkola et al. 1991; Melikov 2004; Melikov et al. 2013a; Luo et al. 2018).

Conclusions

This study compares advanced cooling systems of indoor climate on indoor air thermal conditions and draft discomfort:

- None of the studied systems achieved the desired thermal environment of category A defined by EN ISO 7730. However, all the systems can fulfill the demands of category B except the CB system which falls to category C at the peak load conditions by exceeding the draft risk limit in one location.
- The studied systems provided mainly the acceptable draft risk level below 20% in thermal environment according to ASHRAE 55.
- The horizontal air temperature difference between the perimeter area and the corridor side was largest with the CCMV system and smallest with the DCV system at the both heat gain levels.
- The CCMV system provided the smallest share of locations, in which the draft risk level was higher than 10 %. However, the differences decreased between the studied systems while the heat gain was increased to the peak load level.
- The maximum of draft risks and the maximum of mean air speeds were smallest with the CCMV system in the both heat gain conditions.
- The study suggest to use radiant cooling systems combined with an air distribution sytem to decrease draft discomfort caused by free and forced convection flows.

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