ELSEVIER

Energy and Buildings 34 (2002) 497–509

ENERGY and BUILDINGS

www.elsevier.com/locate/enbuild

A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems

Atila Novoselac, Jelena Srebric*

222 Engineering Unit A, Department of Architectural Engineering, The Pennsylvania State University, University Park, PA 16802 1417, USA

Received 11 July 2001; accepted 2 November 2001

Abstract

This paper reviews the studies and design of cooled ceiling and displacement ventilation (CC/DV) systems in buildings. If properly designed, the combined CC/DV systems can provide better indoor air quality and thermal comfort level compared to the widely used variable air volume (VAV) mixing systems. The cooling load removed by DV is a key design parameter. A low DV load has a positive effect on thermal comfort due to a small vertical temperature gradient, yet also has a negative effect on indoor air quality due to the increased mixing of room air. The impact of the room height on the temperature and contaminant concentration profiles is negligible in the occupied zone. The CC/DV systems are more effective in removing active contaminants (as indicated by CO_2) than passive contaminants (e.g. VOCs). The condensation risk on the chilled ceiling panel is high because of the high humidity ratio in the region close to the panel. To prevent condensation on the panel, it is important to properly control the system for transient regimes, such as startup and shutdown periods, and to minimize infiltration of humid outdoor air. Whether a CC/DV system may or may not reduce energy consumption depends on the supply air temperature, outdoor airflow rate, and cooling load. Therefore, it is necessary to develop design guidelines for CC/DV systems for US buildings because the climate, building layout, and cooling load can be different from those studied elsewhere.

Keywords: Cooled ceiling; Displacement ventilation; Indoor air quality

1. Introduction

Indoor air quality and thermal comfort are the most important characteristics of an indoor environment. As people spend more time indoors, heating, ventilating and air-conditioning (HVAC) systems that provide high indoor air quality and thermal comfort become very important. High indoor air quality may be achieved with an HVAC system that provides a sufficient amount of fresh air to the occupied zone while effectively removing the contaminants. Also, HVAC systems should provide appropriate air temperature by removing heat from the occupied zone while avoiding drafts, large air temperature gradients, and large radiant asymmetry. Both thermal and indoor air quality requirements should be satisfied economically because these systems are the largest energy consumers in buildings. According to the Department of Energy [1], HVAC systems consume more than 40% of the total energy used by commercial buildings. Therefore, providing optimal HVAC system designs for different types of buildings, such as offices, educational institutions, health care or any other type of commercial buildings, is crucial for reduction of energy consumption. However, defining optimal HVAC system designs is challenging, and it was the main topic of many research studies such as those presented in this paper. One of the HVAC systems that attracted attention for its potential for energy savings while providing high indoor air quality and thermal comfort is a combined cooled ceiling and displacement ventilation (CC/DV) system.

The combined CC/DV system is based on the extensive development of CC/DV systems. The DV system provides high indoor air quality by supplying fresh air with low air velocity directly to the occupied zone [2]. This direct supply of cold air and characteristic temperature stratification in the occupied zone may cause thermal discomfort. Therefore, the temperature difference between the supply and room air should be relatively small, which limits DV cooling capacity. For higher cooling capacities, supply airflow rate is considerably increased, and that may turn the stratified flow into mixing ventilation [3]. Also, larger airflow rates require larger DV diffusers, ductwork and air handling units that result in higher energy consumption for fan operation and air handling [4].

An alternative increase of cooling capacity with DV is possible by removing heat with an additional cooling

* Corresponding author. Tel.: +1-814-863-204; fax: +1-814-863-4789.
E-mail address: jsrebric@engr.psu.edu (J. Srebric).

0378-7788/02/$ – see front matter
PII: S0378-7788(01)00134-7

Nomenclature

C_e contaminant concentration in the exhaust air (mg/m^3)
C_i local contaminant concentration (mg/m^3)
C_s contaminant concentration in the supply air (mg/m^3)
P total cooling load (kW)
Q supply airflow rate (m^3/s)
R portion of cooling load removed by CC
W humidity ratio (g/kg)
μ contamination degree

system, such as a CC system. The CC system removes heat from the heat sources directly by radiation and indirectly by convection. There are different types of CC, such as chilled beams, cooling slabs, cooling grids, but most widely used are water-cooled radiant panels built in dropped ceilings (Fig. 1). For radiant panels, cooled water flows through metal tubes connected with metal-sheet panels, removing heat collected by the panels. Because of fast dynamic response and capability to use plenum above the cooling panels for building service systems, the water-cooled radiant panels seem to be most suitable and popular for the combination with DV. Therefore, this paper will focus on that type of CC.

With combined CC/DV systems, ventilation and cooling tasks are separated. The CC panels remove part of sensible cooling loads, while DV system removes pollutants, latent cooling loads and another part of sensible cooling loads. The CC removes sensible cooling loads by convection and radiation with minimum possible disturbances to the stratified airflow (Fig. 1). Therefore, with proper design, it is possible to achieve high indoor air quality and thermal comfort in the occupied zone. Combined CC/DV systems are especially suitable for office buildings and other buildings with a large core zone and small moisture variation due to outdoor air infiltration. In these buildings, the influence of outdoor weather conditions is small, and, therefore, cooling is needed through the whole year to remove internal cooling load. For zones near the externals walls, an additional perimeter-heating system is usually used during the winter when heating is needed. It is possible to use the same ceiling panels for heating in these external zones, but perimeter heating is more suitable for the combination with DV because perimeter heating supports the stratified airflow pattern and prevents downdraft from external windows. Also, warm ceiling heating is limited with relatively low temperature of ceiling panels because people are more sensitive to radiation asymmetry caused by worm ceiling than by CC [5].

The studies of combined CC/DV systems use two main research approaches: experimental and numerical (computer) modeling. Experimental research was mostly done in environmental chambers [6–9], but some research results were based on on-site measurements [10]. The measured data are used to assess thermal comfort and indoor air quality. In addition, some studies included human subjects for direct validation of thermal comfort models [3,8]. The air quality is usually assessed with contaminant distributions, where contaminant sources are simulated with a tracer gas in an environmental chamber. Also, the gaseous contaminant dispersion within a space can be revealed with the smoke visualization. Results obtained by experimental research are considered to be reliable, but also they are very costly and time consuming. As the computer modeling improves its accuracy and speed over time, it is becoming more and more popular.

The relatively low cost of computer simulations is a main reason for their wide application in research of combined CC/DV systems. Another advantage over experimental

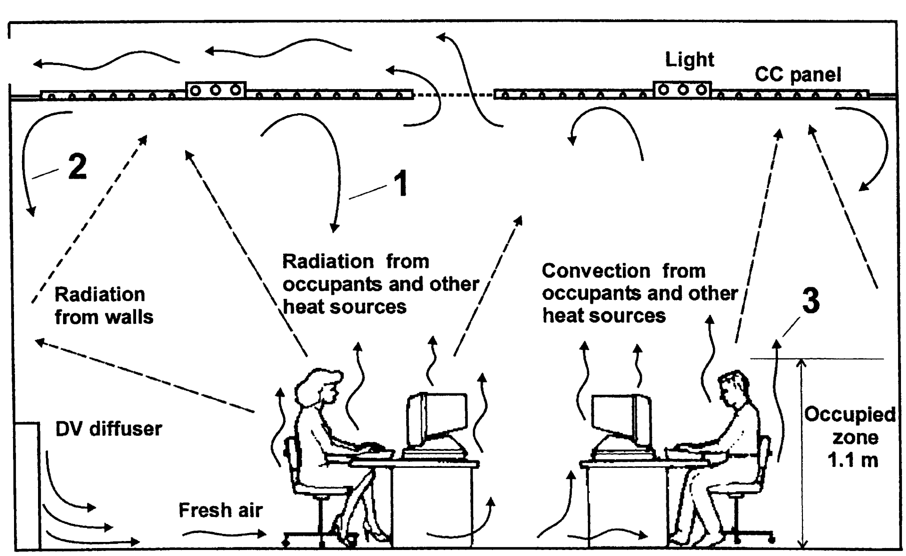


Fig. 1. The combined CC/DV system performance (1) downward convection below CC, (2) downward convection from cold walls, and (3) upward thermal plumes from heat sources.

approach is that computer simulations can easily change the boundary conditions to study different scenarios. The simulations include the calculations of airflow and temperature distributions by computational fluid dynamics (CFD) [11–15], heat transfer and energy consumption by dynamic simulations of building thermal performance [16,17] or the combination of both [18].

The simulation or experimental results can be grouped in the following three aspects:

1. thermal comfort: the distributions of predicted people dissatisfied (PPD) temperature, and velocity;
2. air quality: contaminant distribution and flow pattern; and
3. energy efficiency: energy consumption of CC/DV compared to that of other HVAC systems.

The objective of this paper is to provide a critical review of the CC/DV systems in the three aspects.

2. Thermal comfort parameters

The thermal comfort parameters are air temperatures, relative humidity, air velocities, and surface temperatures [19].

2.1. Air temperature distribution

For the combined CC/DV system, a vertical temperature gradient should exist because it indicates stratified airflow pattern and vertical stratification of contaminants. Consequently, an upward air motion around heat sources such as occupants provides fresh air from lower air layers directly to the breathing zone. On the other hand, the temperature gradient should be small for an acceptable thermal comfort. Therefore, design of CC/DV needs to achieve appropriate vertical temperature stratification [8,20].

The current design of CC/DV system is based on the vertical temperature gradient between the ankle and head levels. Table 1 presents vertical temperature gradient in the occupied zone (0.1–1.1 m above the floor) obtained from several different studies. The temperature gradient in the occupied zone varies from 0 to 2 °C/min. These differences are due to different experimental thermal and fluid flow conditions, such as cooling loads, ventilation rates, supply air temperatures and CC panel temperatures. Furthermore, distribution of supply diffusers, CC panels, and heat sources also influences the temperature gradient in the occupied zone.

Table 1
The vertical temperature gradient in the occupied zone (0.1–1.1 m)

References	Temperature gradient (°C/min)	Research approach
Niu and Kooi [12]	2	Simulations
Kruhne [25]	0	Experiments
Kulpmann [6]	1.5	Experiments
Fitzner [22]	0	Experiments
Alamdari [23]	1.2–1.7	Simulations and experiments
Behne [20]	0.4–1.2	Experiments

In general, vertical temperature gradients depend on the ratio of the cooling load removed by DV to the cooling load removed by CC. In cases where the cooling load removed by CC is considerably larger than that removed by DV, downward airflow motion from the CC panels is strong. Therefore, a stratified displacement airflow pattern is destroyed and the airflow becomes mixing, causing relatively uniform temperature and contaminant distribution. The uniform temperature distribution provides better thermal comfort, but not a better air quality. For a good indoor air quality, a temperature gradient is preferred, but the gradient should not cause thermal discomfort, especially near the floor. According to ANSI/ASHRAE standard 55-1992 [21], the upper limit for the vertical temperature gradient is 3 °C/min. Tan et al. [8] concluded from their experiments that this upper limit should be 2.5 °C/min.

2.2. Velocity

Velocity is another important thermal comfort parameter. Loveday et al. [3] showed that a low CC temperature, which increases the CC cooling capacity, could increase air velocities in the occupied zone. This velocity increase is due to the downward airflow motion caused by the negative buoyancy force from the CC. Nevertheless, this natural convection induced air velocities that are still low and usually do not cause draft. In fact, the highest air velocities in the occupied zone are usually just above the floor (~5 cm) [6], which is the jet region and approximately the position of the maximum jet velocity from the displacement diffuser.

For a cooling load of 62 W/m^2 and CC temperature in range from 21 to 14 °C, Loveday et al. [3] measured maximum air velocity of 0.11 m/s in the occupied zone. This air velocity is at the lower limit of the velocity range, 0.11–0.16 m/s, estimated for the same cooling load in another experimental study [7]. In both studies, velocities were far below velocity that can cause draft. According to Behne [7] and Fitzner [22], there is no risk from draft caused by CC if the total cooling load in a space is less than 100 W/m^2 (floor area).

For large spaces, it was noticed that the cold air form the ceiling could burst into the occupied zone in the areas without heat sources. However, this effect disappears when heat sources such as occupants move into the area because the thermal plumes displace the cold air and eliminate draft risk [6,23].

2.3. Mean radiant temperature and radiant asymmetry

Rooms with the combined CC/DV system usually have slightly lower or approximately the same mean radiant temperature as the room air temperature because the entire enclosure is cooled by radiation. On the other hand, with all air systems such as variable air volume (VAV) systems, mean radiant temperature is higher than the room air temperature. For perimeter rooms with conductive solar heat gains, a study [24] showed that CC/DV system has sufficient radiant capacity to bring down the mean radiant temperature close to the room air temperature. Authors investigated the influence of different building envelope insulation on mean radiant temperature. For a room without insulation and all air system they calculated 2 °C higher mean radiant temperature than the room air temperature. These results agree with Kruhne's [25] and Fitzner's [22] measured data that show approximately 2 °C lower mean radiant temperature with combined CC/DV systems than with all air systems. Consequently, based on ANSI/ASHRAE standard 55-1992 [20], room air temperature with CC/DV system could be approximately 2 °C higher to obtain the same thermal comfort as with the all air system.[1] Also, lower mean radiant temperature has positive effect on the thermal comfort because it provides effective radiant heat extraction from the occupied zone.

Differences in room enclosure temperatures can lead to thermal discomfort due to radiant asymmetry, even when the mean radiant temperature is within acceptable limits [5]. The temperature difference between the CC panels and floor increases with the decrease of the CC temperature, where the floor temperature is higher than the ceiling temperature. However, this temperature asymmetry has small effect on the thermal comfort for the CC/DV system. Kulpmann [6] measured the temperature asymmetry of 5.3 °C at 1.1 m above the floor, which was clearly below the highest acceptable value of 14 °C for CC surfaces [5]. Loveday et al. [3] confirmed this in their experiments with human subjects. They showed that the CC temperature of 22–12.5 °C causes radiant temperature asymmetry from 0 to 4 °C. Also, they noticed that this radiation asymmetry has almost no effect on the thermal comfort.

3. Indoor air quality parameters

The environmental parameter that indicates indoor air quality is contaminant concentration in the occupied zone, especially in the breathing zone. The air quality for a certain location in a room can be determined by the contamination degree [22,25] that is also called dimensionless concentration [6] or contaminant removal efficiency [20],

$$\mu = \frac{C_i - C_s}{C_e - C_s} \tag{1}$$

where C_i is the local contaminant concentration, C_s the contaminant concentration in the supply air and C_e the contaminant concentration in the exhaust air. The contamination degree depends on spatial distribution of the contaminant sources, but also on the airflow pattern within a room.

A typical value of contamination degree for the perfect mixing ventilation is 1.0 because the exhaust contaminant concentration tends to be equal to the local room concentration. However, DV provides better contaminant removal than the mixing ventilation due to the vertical contaminant stratification. Consequently, for the same supply flow rate, the displacement ventilation has a higher ventilation effectiveness than mixing ventilation. Hence, contamination degree is lower than 1.0 for DV system.

3.1. Influence of heat sources

The vertical contaminant stratification creates a high ventilation effectiveness for DV system without CC. This stratification is characterized by the following two zones in a room:

1. a lower zone with stratified airflow pattern and clean air;
2. an upper zone with mixed airflow pattern and polluted air.

Between the stratified and mixed zone is a transitional area usually called "stratified boundary". Well-designed CC/DV systems should have both zones characteristic for DV.

Three effects are important for the height of the stratified boundary and air quality in the occupied zone in a room with CC/DV (Fig. 1):

(1) downward convection below CC;
(2) downward convection from cold walls;
(3) upward thermal plumes from heat sources.

The downward convection below CC moves the stratified boundary with CC/DV system closer to the occupied zone compared to the height of the stratified boundary layer with only DV system. According to Loveday et al. [3], the stratified boundary appeared at height of about 2.0 m regardless of cooling load (from 25 to 52 W/m^2) in their experiments. However, when CC system was on, the boundary layer was suppressed to 1.5 m above the floor for cooling load of 62 W/m^2. Kulpmann [6] noticed that the stratified boundary might even have a higher pollutant concentration than that with the perfect mixing ventilation ($\mu > 1.0$).

The downward motion from CC might suppress the stratified boundary into the occupied zone, especially in room sections without heat sources. This causes unexpected reverse effects on air quality in this zone. However, when pollutant sources are associated with heat sources, pollutant concentrations in the occupied zone are similar to the concentrations with DV only [26]. Fitzner [22] analyzed the influence of heat source and upward convection on local air quality and found that sitting person might have remarkable improvement of the inhaled air due to buoyancy driven

[1] Operative temperature is the average value of mean radiant temperature and room air temperature for the same coefficient of convective and radiant heat transfer from human body, which is typical for office spaces (ASHRAE fundamentals 1997).

transport of fresh air from the layer near the floor to the person's nose (Fig. 1). Alamdari [23] also reported that, for cooling load of 60 W/m^2, upward convection was dominant in the vicinity of the occupants even with the strong downward air motion from the cooled panels. This study also showed that the downward convection near the sidewalls with low wall temperatures could cause a transport of pollutants from the upper mixed zone directly into the supply air layer with clean air.

3.2. Influence of contaminant source type

The pollutant sources can be active or passive. Active sources are those associated with heat sources, such as CO_2 from occupants. Passive sources are not linked with heat sources, such as volatile organic compounds released from building materials. The displacement ventilation can remove active sources more efficiently than the passive sources [22,27]. The reason is that the thermal plumes from the heat sources bring the active contaminants to the upper zone, and prevent the mixing of pollutants with the clean air in the lower zone. Passive contaminants, especially those released below the breathing zone, can mix with the clean air and decrease the indoor air quality in the lower zone.

4. Energy and capital costs—comparison with other systems

The thermal comfort and indoor air quality requirements are the same for any HVAC system. However, the annual energy consumption and capital costs can vary significantly for different HVAC systems, and represent crucial parameters for selection of HVAC systems.

4.1. Energy consumption

Almost all energy consumption analyses for the combined DV/CC systems were done by numerical simulations because it is too expensive and time consuming to perform hourly measurements of energy consumption. Several researchers [16–18,28], calculated the total annual energy consumption. This review expresses the energy consumption in terms of dimensionless unit for comparison of the CC/DV systems with other HVAC systems. The following energy price ratio is used in order to make different types of energy comparable [17,18], heating energy:cooling energy:electrical energy = 1:1:3.

Fig. 2 compares the annual energy consumption of the combined CC/DV systems with that of VAV systems from three different studies [17,18,28]. The energy consumption for the VAV systems was normalized to be 100%. All the three studies are for western European climate conditions. During the heating period perimeter heating is used to offset heating load from the building external enclosure. Heat recovery and free cooling were used in both the VAV and combined CC/DV systems. The results show that the combined CC/DV system may or may not save energy. The difference in energy consumption is a function of supply air temperatures, outdoor airflow rates and cooling loads besides the climate conditions.

With increase of peak cooling loads, the combined system became more economical than the VAV system. Sodec [17] showed this trend for the combined cooled ceiling and mixing ventilation (CC/MV). The study demonstrated that increase of cooling loads increases energy savings with the combined CC/MV compared to the VAV system. The same relationship is evident for the combined CC/DV systems in Fig. 2. Potentials for energy savings with the combined CC/DV system confirmed Brunk [16]. The author compared CC/DV and VAV systems for cooling load of 73 W/m^2 (floor area) and calculated more than 35% lower energy consumption for the CC/DV system than that with the VAV system. This analysis did not include the energy for perimeter heating and, therefore, is not directly comparable to the previous three studies presented in Fig. 2. Nevertheless, the study showed that the combined CC/DV system consumes less energy than VAV for larger cooling loads because

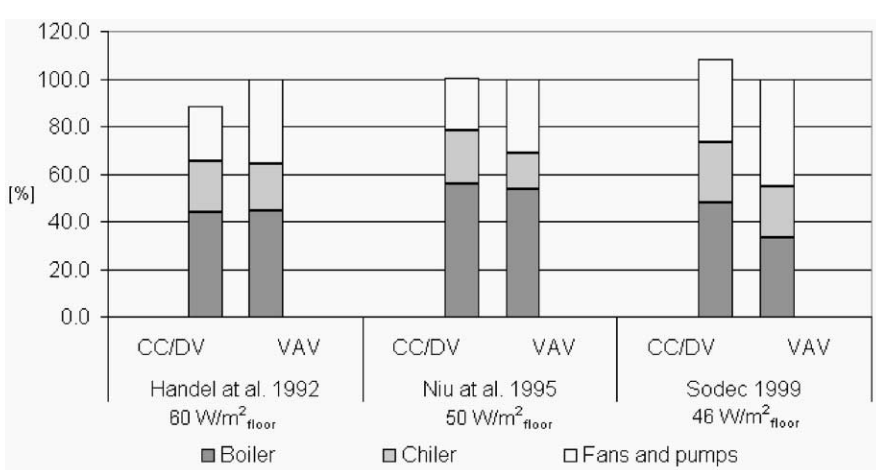


Fig. 2. The comparison of the energy consumption for the combined CC/DV and VAV system.

perimeter heating energy consumption is only a small portion of the total annual energy consumption.

Fig. 2 shows that the heating energy consumption with the combined CC/DV system is greater than or similar to the heating energy consumption of the VAV system. These differences can be explained by different room types. For perimeter rooms, where demand for heating exits during winter, the total heating energy consumption for each of the two systems is similar. For central room type, where cooling demand is dominant during the whole year, the heating energy consumption with the combined CC/DV system is greater because during the winter DV air supply temperature is greater than air supply temperature with the VAV system. Therefore, additional heating is needed [16].

For all three analyses presented on Fig. 2, the combined CC/DV system consumes less electrical energy and more cooling energy than the VAV system. With CC, the following two effects are important for the total electrical energy consumption:

1. reduction of electrical energy consumption for fans due to lower air volume flow rate; and
2. increase of electrical energy consumption for cooling tower and CC pumps.

The overall result is decrease of the total electrical energy consumption with the CC/DV system because of the high thermal capacity of water. The CC system (hydronic system) removes a given amount of thermal energy with only a small part of the otherwise necessary fan energy [29,30]. On the other hand, a larger cooling energy consumption with the combined CC/DV than with the VAV system is due to the free cooling. The outdoor air free cooling for the VAV system can be used for a longer period during year than the free cooling of cooled water by cooling tower for the combined CC/DV system [16].

Compared to the combined CC/MV system, the CC/DV system has a greater annual energy consumption. Sodec [17] compare energy consumption of these systems (Fig. 3) and calculated 17% larger energy consumption with the combined CC/DV system. The reason for this larger energy consumption is higher air supply temperatures with DV. To keep the air dew point temperature below CC temperature, the outdoor air is first cooled and dehumidified, and then heated to the supply temperature for DV. This process considerably increases the annual energy consumption for heating.

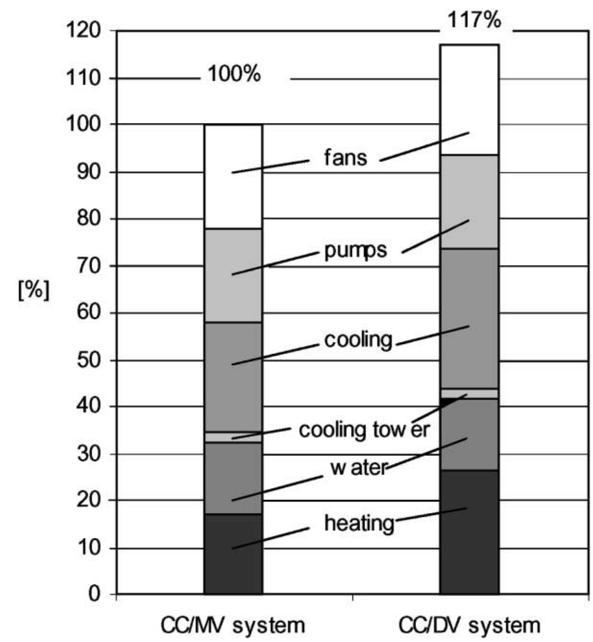


Fig. 3. The comparison of the energy consumption for the combined CC/MV and CC/DV system [17].

4.2. Capital cost

An exact capital cost difference between the combined CC/DV system and equivalent VAV system is difficult to determine. It depends mainly on market prices and contractors, and varies considerably for different countries. In general, for smaller cooling loads, the combined CC/DV systems have a higher capital cost than VAV systems. CC reduces the amount of circulated air, which requires smaller air-handling units and ducts, but extra cost is necessary for CC panels. Handel et al. [28] calculated that the combined CC/DV is approximately 20% more expensive than VAV system for cooling load of 60 W/m^2 (floor area). However, with the increase of cooling capacity, the capital cost for VAV systems grows faster than for the combined CC/DV systems. From a certain cooling capacity, the capital cost for the combined system will be lower than for VAV system. For the German market prices and cooling loads above 55 W/m^2 (floor area) the capital cost for CC/MV system is more favorable than the one for VAV system [17]. If the building space reduction (plenum and equipment room) with CC system is included, the cooling load break point for economical CC/MV system is even lower. A similar relationship for CC/DV and VAV first cost is expected and needs to be determined for the US market.

The capital cost of the combined system can be considerably reduced by reduction of the cooled panel area because CC panel price significantly contributes to the total price of the system. According to Sodec [17], the cost of whole CC system is from 50 to 55% of the total system cost for German market. For US market, this portion is even higher and, according to CC manufacturers, is approximately 75%. The greatest part of this portion is for CC panels. However, a lower panel area requires a lower panel temperature for the same cooling loads and, therefore, higher control demands to prevent condensation. Also, a smaller panel area often increases the operating cost because of the reduced period for free cooling due to the lower water supply temperature [18].

5. Design parameters

To design an energy efficient CC/DV system that satisfies thermal comfort and indoor air quality requirements is challenging because of the complex interactions between CC and DV systems. Therefore, design of the combined system is more difficult than the design of the CC and DV systems working independently. Use of the design guidelines for CC or DV as independent systems is not appropriate for design of the combined CC/DV system.

5.1. Cooling load, ventilation rate and cooled panel temperature

Several studies [8,20,22] suggested important design parameters as cooling loads, ventilation rates and cooled panel temperatures. These studies reported that cooling loads removed by CC and DV in the combined system should be properly adjusted. Fig. 4 shows the cooling load removed by CC as a function of the contamination degree in which Behne's study [20] was for a cooling load of 30–95 W/m^2 by curvilinear regression and Fitzner's investigation [22] was for a cooling load of 15–50 W/m^2 through linear regression. Fig. 4 shows that, for a particular portion of the cooling load removed by CC, the expected mean contamination degree is in the range presented by upper and lower boundary lines. The two studies predicted a similar contamination degree.

The mean contamination degree actually depends on the vertical temperature gradient. Tan et al. [8], showed that a vertical temperature difference between the head and ankle of 2–2.5 °C would not cause thermal discomfort in the occupied zone. They developed a design diagram for the combined CC/DV system shown in Fig. 5, where *R* is the ratio of the cooling load removed by DV over the total cooling load. The diagram gives the relationship between the vertical temperature gradient, cooling load, ventilation rate and CC/DV cooling capacity. For a particular temperature gradient, the total cooling load and *P*/*Q* ratio (ratio between the total cooling load and supply airflow rate), the *R* can be determined by the diagram. For example, to design a temperature gradient of 2.5 °C/min, the *R* would be 0.6, when the *P*/*Q* ratio is 15 kW/(m^3/s), or, for given $R = 0.75$, and a *P*/*Q* ratio of 18 kW/(m^3/s), the temperature gradient in the occupied zone would be 2.2 °C/min.

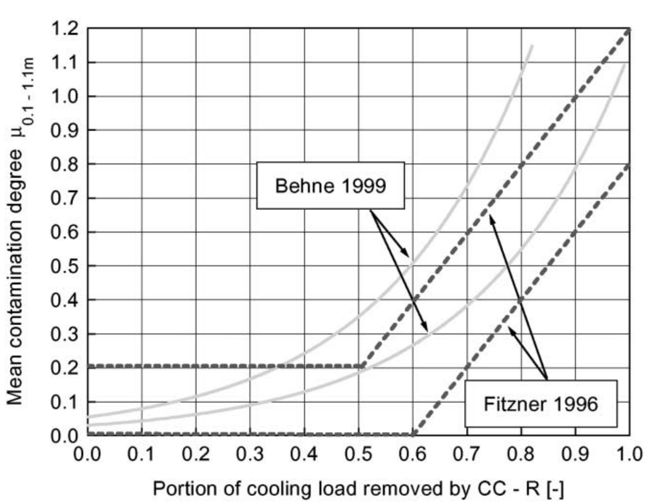


Fig. 4. The cooling load removed by CC vs. the mean contamination degree.

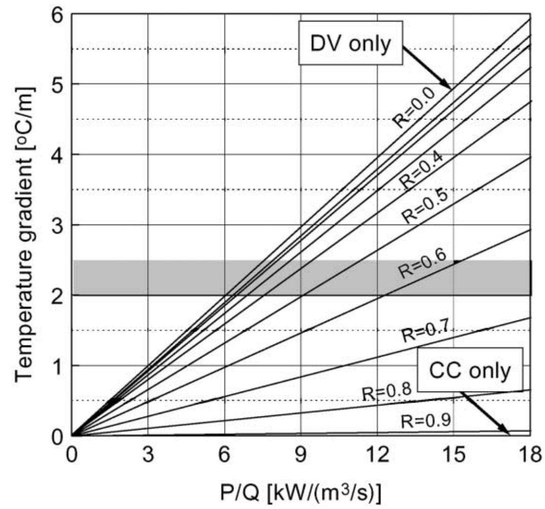


Fig. 5. The design diagram for the combined CC/DV [8].

Tan et al. [8] recommend the maximum *P*/*Q* ratio of 18 kW/(m^3/s). Larger *P*/*Q* ratios would probably cause mixing and lower air quality in the occupied zone. For a temperature gradient of 2.0 °C/min, a minimal portion of cooling load removed by DV should be 33%. Behne [20] constructed a similar design diagram that also includes mixing ventilation systems. Behne [20] pointed out that good thermal comfort and air quality for the combined CC/DV system can be expected when a DV system removes at least 20–25% of the total cooling load, which is close to one-third from Tan et al.'s study [8].

In general, a lower portion of the cooling load removed by DV provides slightly better thermal comfort, while a higher portion results in slightly better air quality. The minimal portion of the cooling load removed by DV that provides good indoor air quality determines a minimal supply airflow rate for DV, because the difference between supply and exhaust air temperature is limited. Table 2 shows the minimum airflow rate for different cooling loads and different portions of cooling load removed by DV when the temperature difference between the supply and exhaust air is 5 °C and room height is 2.8 m. With an estimated maximum occupancy of seven people per 100 m^2 floor area and an outdoor air rate of 10 l/s for a typical office space [31], the air exchange rate is 0.9 h^{-1}. This airflow rate is relatively low compared to those in Table 2. Hence, for a typical office room where a demand for the fresh air is relatively low and the cooling loads are high, total airflow rate is larger than the one needed for ventilation. Because the recirculation of

Table 2
The minimum DV airflow rate for an office space with a temperature difference of 5 °C between the supply and exhaust air

Cooling load (W/m^2 (floor area))	Portion of cooling load removed by DV (%)	Minimal air volume flow rate—air changes per hour (h^{-1})
35	20	1.5
35	33	2.5
70	20	3.0
70	33	5.0

room air with DV brings back pollutants directly to the occupied zone, systems with 100% outdoor air and heat recovery should be used.

Cooling loads in an office building are usually between 35 and 70 W/m^2 (floor area) where higher values are typical for US offices. Several studies have tried to define a maximum cooling load for the combined CC/DV system. Niu and Kooi [26] reported that the combined DV/CC system for cooling load of 50 W/m^2 (floor area) gives rather good thermal performance and almost the same ventilation effectiveness as a DV system with a cooling load of 25 W/m^2 (floor area) with cooled panel area covering 60% of the total ceiling area. The maximum cooling load could be even higher. Alamdari [23] reports 60 W/m^2 (floor area) with a cooled panel area covering 75–85% of the ceiling. Further increase of ceiling area covered by panels is difficult because lighting and exhaust air systems cover a certain part of the ceiling area. For higher cooling loads, chilled beams were recommended [23]. According to Behne's [20] design diagram, this limit can be as high as 100 W/m^2 (floor area). In general, a large cooling load requires a large portion of the load removed by DV, which increases the airflow rate and fan energy consumption.

In addition to those parameters mentioned earlier, other parameters, such as pollutant source, ceiling height, and moisture distributions, are also important in design of the combined CC/DV system. In order to illustrate the importance of these parameters, we use a conference room as shown in Fig. 6 as an example that was studied numerically. The room is located in a building internal zone (no outdoor walls or windows) with 18 occupants. The HVAC system supplies 100% of outdoor air at 20 °C with a flow rate of 5.7 ACH (ASHRAE standard 62). The total cooling load in the room is 2.4 kW with which 60% was removed by the CC that had a ceiling panel temperature of 18 °C. From Figs. 2 and 4, the corresponding mean contamination degree in the occupied zone ($\mu_{0.1-1\ m}$) should be between 0.27 and 0.5, and the room air temperature gradient is 2.3 °C/min.

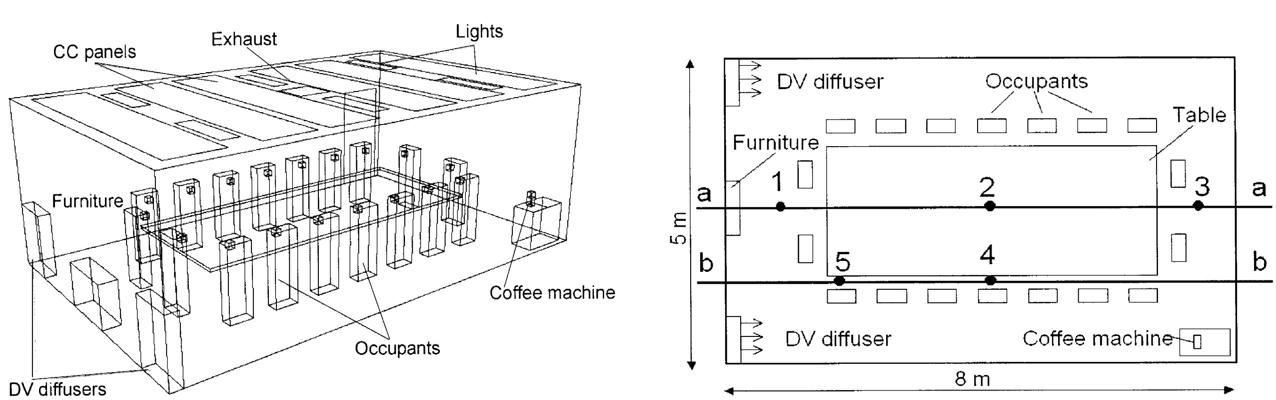
5.2. Pollutant source type

This study used CO_2 as an active pollutant source and VOC from the carpet as a passive pollutant source. Fig. 7 presents the calculated vertical contamination degree for the CO_2 and VOC at five positions in the room (Fig. 6(b)). The vertical line with a removal efficiency of 1.0 that is for perfect mixing condition is used as a reference. The CO_2removal efficiency has a similar profile at different locations in the room. In the occupied zone, μ is smaller than 1.0, and at the upper zone, μ is close to 1.0. In position 2 at 0.7 m above the floor, CO_2 concentration increases sharply because the table in the room blocks the airflow. Otherwise, the average CO_2 removal efficiency in the occupied zone ($\mu_{0.1-1\ m}$) is in the range of Behne's diagram (Fig. 4). The average vertical CO_2 concentration profile represents a typical CO_2 concentration distribution in the room.

On the other hand, the VOC removal efficiency is considerably different from that of CO_2. In some locations, the CC/DV system provides a better air quality than the perfect mixing system, but can be worse in the other locations. The average VOC profile is close to the mixing one that is not typical in the room. The results are similar to those for DV system [27].

5.3. Ceiling height

DV systems perform better in spaces with high ceilings, such as atria, concert halls, and various industrial spaces [2].



(a) (b)

Fig. 6. The layout of the conference room (a) perspective view and (b) plan view.

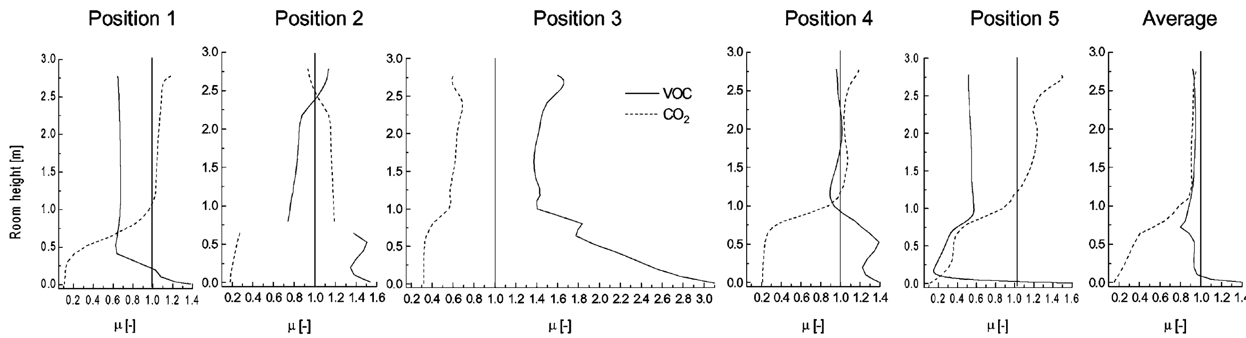


Fig. 7. VOC and CO_2 contamination degree for different positions in room.

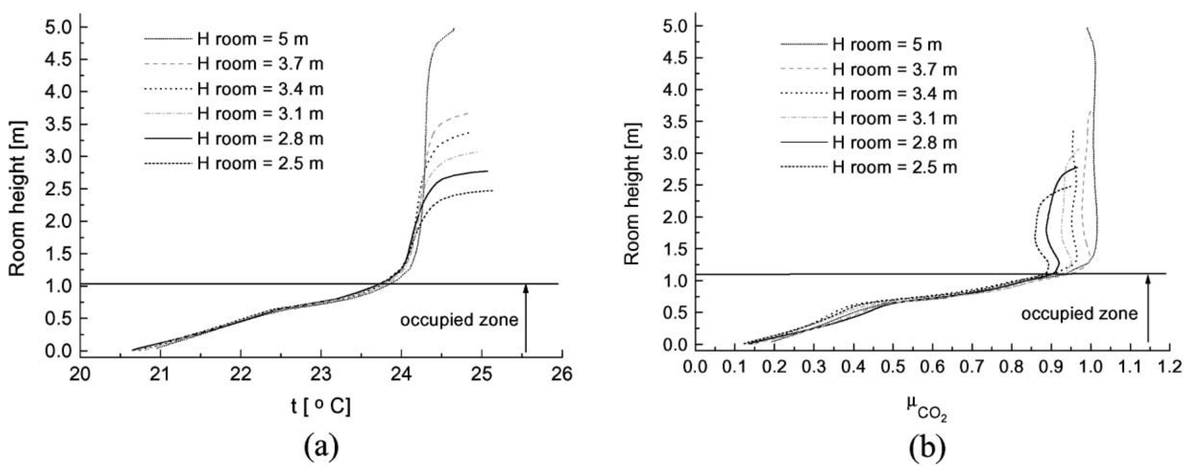


Fig. 8. The influence of room height on (a) the average temperature gradient and (b) average CO_2 vertical distribution.

However, for the combined DV/CC system, the temperature gradient at the breathing zone is almost identical when we varied the ceiling height from 2.5 to 5.0 m as shown in Fig. 8(a). The same trend was found for indoor air quality in which CO_2 was a major contaminant. The cooling load removed by CC is almost the same for different room heights. The average air temperature is higher closer to the ceiling because of the convective heat transfer from the lighting system. The mean contamination degree in the occupied zone, as shown in Fig. 8(b), is approximately 0.5,

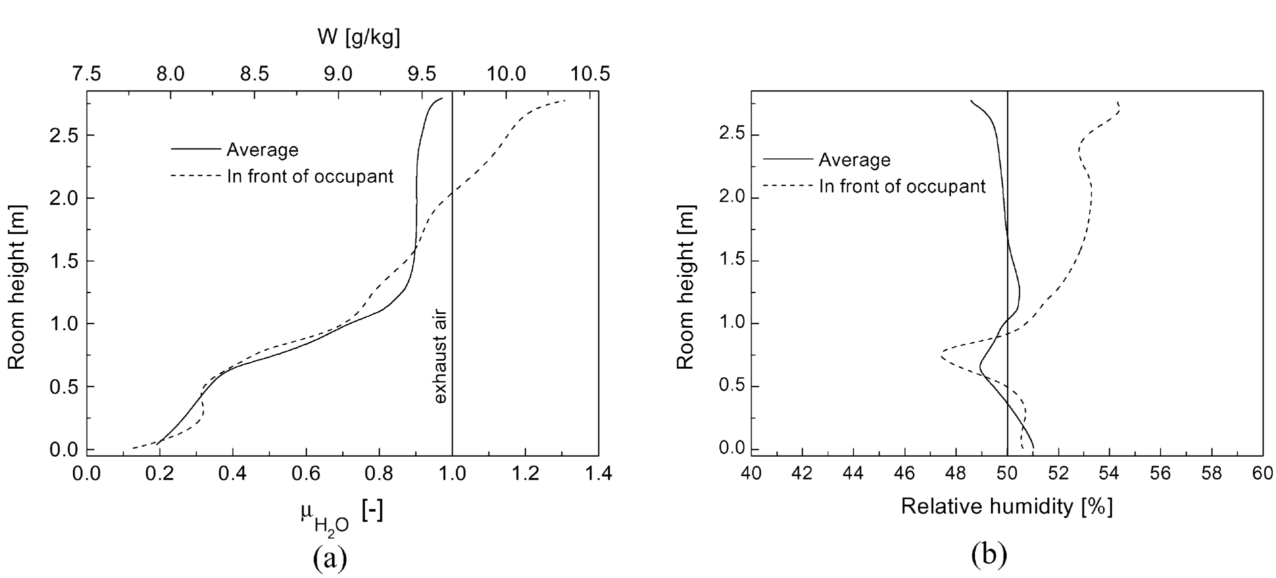


Fig. 9. The room vertical distribution of average and local (a) humidity ratio, and (b) relative humidity.

which is in the expected range (0.27–0.5). In the upper zone, the contamination degree is between 0.85 and 1.0 with higher values for higher ceilings. This indicates a more intensive mixing in the zone for higher ceiling rooms.

5.4. Moisture and condensation on CC

When a mixing ventilation system is combined with CC, it is appropriate to assume a uniform moisture distribution in a room. The dew-point temperature should be the minimum CC surface temperature. However, the assumption of a uniform moisture distribution may not be appropriate for the CC/DV systems. In the conference room (Fig. 6), the moisture sources are the occupants (0.085 kg/h) and a coffee machine (0.1 kg/h). If the humidity ratio in the supply air is 7.5 g/kg, the corresponding humidity ratio at the exhaust air would be 9.7 g/kg. With an exhaust air temperature of 24.5 °C, the relative humidity is 50% and the dew-point temperature is 13.5 °C. Since the moisture sources are active, the moisture distribution in the room is similar to the CO_2 distribution. Fig. 9(a) presents average and local (position 5 in Fig. 6(b)) humidity ratio profiles in the room. The humidity ratio *W* is considerably lower in the occupied zone than that close to the ceiling. However, it does not mean a lower relative humidity in the breathing zone as shown in Fig. 9(b) because the corresponding air temperature is lower too. In fact, the vertical average relative humidity is almost a constant. Therefore, the thermal comfort level would be the same.

However, the high humidity ratio near the ceiling (Fig. 9(a)) that is brought to the zone by the thermal plumes from the occupants may cause condensation problems, even when the CC surface temperature is higher than the mean dew-point temperature. For the example shown here, the CC surface temperature should be at least 1 °C higher than that with perfect mixing systems.

6. Discussion

The above discussed design parameters strongly influence the performance of the combined CC/DV systems. Properly designed systems can provide excellent indoor environment with low energy consumption. Further considerations on CC/DV performance and design are reduction of energy consumption and peak power demand, control strategy and characteristics of CC panels.

6.1. Energy considerations

6.1.1. Energy consumption reduction

The free cooling of CC water by cooling towers is used with the combined CC/DV systems because it has a great potential for reduction of the total energy consumption. It is especially suitable in climates with long transient periods between winter and summer [18] calculated that for the Dutch climate with the free cooling, the annual energy cost reduces from 8 to 15%, depending on the CC panel area. Detailed results are presented in Table 3. Sodec [17] obtained similar results for the German climate, and showed that with the higher cooling loads, the larger energy cost reduction is possible with cooling towers. For increase of the cooling load from 46 to 75 W/m^2 (floor area), the annual energy cost reduction due to the free cooling increase from 9 to 20%. Sodec [17] also investigated a possibility to reduce energy consumption by overnight running of the CC cooling. The analysis showed that the total energy costs with and without overnight cooling is the same because the longer operation of the system incurred expenses equal to the savings. Therefore, for CC/DV is recommended that the system does not work during night, which is so called intermittent operation regime.

Table 3
The energy savings with the free cooling by cooling tower with different CC panel areas [18]

Panel area (% of total ceiling area)	25	40	60
Energy saving for cooling (%)	25	37	49
Total annual energy saving (%)	8	12	15

6.1.2. Peak power reduction

With the intermittent operation regime for CC/DV system, peak space cooling loads occurs earlier during a day than with the intermittent operation regime for all-air system. The main reason for this is the direct absorption of radiant heat by the CC panels and the consequent reduction of accumulated heat in enclosure walls [32]. This may cause higher or lower peak space cooling loads with CC/DV than with all-air systems depending on the heat gain distribution during a day. Feustel and Stetiu [29] calculated 9% larger space cooling loads with an all air system, while Niu et al. [32] calculated 12% larger loads with the combined CC/DV system. However, CC/DV systems have a great potential for cooling peak power reduction mainly because of the airflow rate reduction. With CC/DV system peak electrical energy consumption for fans during the peak cooling loads is avoided because this system usually operates with the constant volume flow rate. Feustel and Stetiu [29] calculated peak power requirement for the conventional all-air systems and combined CC/DV systems. The study found that for small offices in cooling regime, the total peak power (including fan and chiller) reduces more than 40%. The fan peak power reduces to 75%, while the chiller peak power reduces to 25% mainly due to reduction of heat gains from lights and fans (Fig. 10). For the night running of the CC, Sodec [17] found relatively low reduction of the chiller peak power of 2.2%. Further peak power reduction is possible with an additional ice storage system [16,33].

Considerably lower peak power for the combined CC/DV system relative to all-air systems does not imply lower energy consumption, but it has direct influence on capital

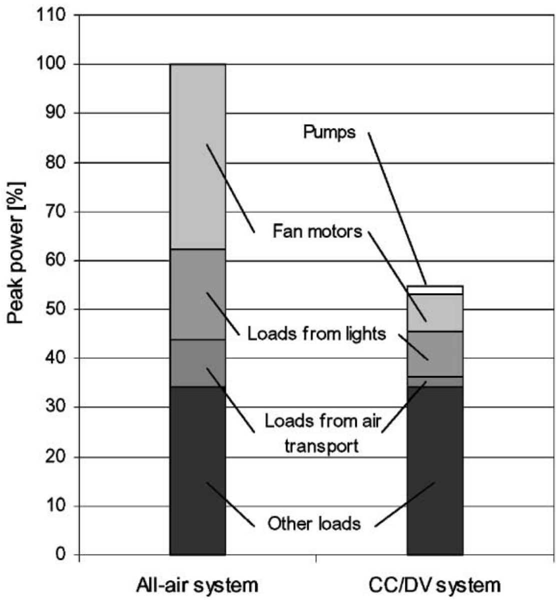


Fig. 10. The comparison of peak power for the all-air system and the combined CC/DV system [29].

cost reduction because of decreased equipment size. Also, reduction of electrical peak power consumption contributes to the stability of electrical network and, in many countries, results in money savings.

6.2. Control strategy

During the operation of the combined CC/DV system, a control strategy has an important impact on the thermal comfort, air quality, energy consumption and protection from moisture condensation. A typical control strategy includes a constant airflow DV system with a fixed air temperature, and variable CC temperature. Consequently, for the off-design conditions, when cooling loads are lower than the design loads, CC temperature is increased and portion of the cooling load removed by CC (R) is reduced. For well-designed CC/DV systems, the increase in CC temperature will not significantly affect thermal comfort and air quality because the room air temperature gradient is only slightly changing with the CC capacity. This change of temperature gradient with reduction of cooling load (P) can be seen on Fig. 5 by following a change of P/Q ratio and R with a change of P. Reduction of cooling loads (P) produces not only reduction of R, but also reduction of P/Q ratio because Q is constant. For example, if design P/Q ratio of 18 kW/(m^3/s) with CC output of 67% drops to 12 kW/(m^3/s), CC output (R) will decrease to 50% and temperature gradient will change from 2 to 2.7 °C/min (Fig. 5). With further drop of P/Q ratio to 6 kW/(m^3/s), CC output will drop to 0 (CC is turned off) and temperature gradient will be again 2 °C/min.

CC temperature control can be provided by:

- constant water temperature with variable flow; and
- constant water flow with variable temperature.

The control with variable flow and constant water temperature is more common then the other one. Furthermore, to avoid condensation, it is important to control panel surface temperature. Conroy and Mumma [34] suggest an additional central dew-point temperature control. With this control, the supply water flow for CC panels increases when the room air dew point temperature is close to the water supply temperature. Another solution is to switch off the cooled water supply as soon as the relative humidity reaches ‘‘dangerous’’ levels. Behne [7] suggested to keep at least 1 °C temperature difference between the panels and room air to insure no condensation on the panels. However, risk from condensation is always present during the startup of the system, after night or weekend breaks. For the outdoor air conditions when outside humidity is higher than the inside one, and during the periods when the system is turned off, there is a certain infiltration of humidity, which increases the air humidity in the room at the system startup time. A possible solution of the problem is an earlier start of DV system and gradual start of CC system [34].

6.3. Characteristics of CC panels

For the performance of the combined CC/DV system, important characteristics of CC panels are a panel capacity, convection/radiation portion, and dynamic response of CC system. The cooling capacity and convection/radiation portion of CC panels depend on air and enclosure temperatures in a room. Therefore, the cooling capacity and convection/radiation portion vary with a type of room, type of heat sources and CC construction.

The variation of convection/radiation portion is from 60/40 to 40/60%. For rooms with large heat gains from radiant heat sources such as lighting system or sun radiation, CC radiation portion is larger. Also, conduction through external walls for perimeter rooms may have significant impact on internal wall surface temperatures and, therefore, on CC convection/radiation portion. However, a construction of CC panels has low influence on this portion.

Influence of CC panel construction investigated Behne [7,35]. The author found that the insulation on upper (ceiling) side of panels and air circulation between CC panels and ceiling has a great influence on cooling capacity, but low on convection/radiation portion. According to this research, panels without insulation and with air circulation on ceiling side have 24% greater cooling capacity than the panels with the insulation. Feustel and Stetiu [29] also reported that the CC panel construction has influence on the cooling capacity. They found that approximately 7% of the CC cooling power cools the ceiling construction and room above the ceiling when the CC panels are without insulation and with intensive air circulation in the plenum above the panels. In general, the panels without insulation have greater cooling capacity, but also greater energy consumption due to the cooling of the plenum and exhaust air above the CC panels.

Water-cooled radiant panels have a small thermal mass and, therefore, a fast dynamic response [36]. Feustel and Stetiu [29] reported that cooling panel systems have dynamic response time comparable to all air systems. Antonopoulos et al. [37] confirmed that in their study, and calculated needed time to reach the summer thermal comfort specified in ANSI/ASHRAE standard 55-1981 for different panel temperatures. For initial room air temperature of 30.5 °C, and panel temperature of 15 °C, they calculated 8 min for reaching summer thermal comfort in the room. Therefore, the combined CC/DV system can be started shortly before the scheduled cooling in a space such as a conference room or an office.

7. Conclusions

Our study reached several conclusions:

- The combined DV/CC system is a HVAC system that may successfully combine advantages of DV related to air quality, advantages of CC related to thermal comfort and increased cooling capacity. However, this system is very sensitive because the stratified boundary layer with high pollutant concentration can be easily suppressed into the breathing zone. To avoid this, certain temperature gradient has to be maintained to provide effective contaminant removal.
- Portion of cooling load removed by CC is a key parameter for temperature gradient and air quality. The higher the portion of cooling load removed by CC, the lower vertical temperature gradient becomes, which means lower contamination degree. With this portion, expected air quality can be estimated by diagram in Fig. 4. Also, based on this portion, complex but very compact and useful design diagram is suggested for the combined CC/DV system (Fig. 5). The diagram combines many design parameters, whose complex interactions had to be studied empirically.
- Depending on cooling load, climate, room type, and system configuration, energy consumption with the combined CC/DV system might be lower or higher than the energy consumption with VAV system. For larger cooling loads, CC/DV system is more economical than VAV. Compared to the combined CC/MV system, CC/DV system has higher energy consumption.
- Typical control strategy for the combined system is constant air volume flow with fixed supply temperature for DV and variable CC temperature or cooled water flow rate. For well designed combined CC/DV system, influence of transient cooling load (variable CC/DV ratio) on thermal comfort and air quality is low. Furthermore, due to the fast dynamic response the system can be started shortly before the scheduled cooling.
- For rooms in which internal cooling load is dominant, the space height has no influence on temperature gradient and air quality in the occupied zone. Also, moisture distribution with the combined DV/CC system is non-uniform. Therefore, a minimal CC surface temperature should be at least 1 °C above the exhaust air dew-point temperature.
- Contamination degree depends on the type and position of pollutant sources. Analysis of contaminant distribution usually found in literature relates to active contaminate sources distribution. When pollutant sources are passive (VOC from building materials), air quality in the occupied zone is variable and for certain room positions can have lower air quality than with the perfect mixing ventilation.

The currently suggested design guidelines primarily deal with the active contaminate sources. Design guidelines that would account for both passive and active contaminate sources for CC/DV system should be developed. Also, further evaluation of this system for the US market is needed in order to provide economical incentives for the CC/DV systems application. The system is especially suitable for the core building zones with no perimeter heating that are typical for US offices. Further study should evaluate suitability of the system for US price ratio for different types of energy and for different US climates. The recommended analysis requires a numerical tool that would combine simulation of airflow pattern, building thermal behavior, HVAC system models, and climate data. The tool can be used for development of design guidelines as well as for design process. Only the combined analysis approach would justify application of the combined CC/DV system for US buildings.

References

[1] DOE, Department of Energy, Information on the Commercial Buildings Sector, Energy Consumption by End Use for 1995: http://www.eia.doe.gov, 2001.
[2] X. Yuan, Q. Chen, L.R. Glicksman, A critical review of displacement ventilation, Energy and Building, ASHRAE Transaction, Vol. 104, No. 1A, 1998, pp. 78–90.
[3] D.L. Loveday, K.C. Parsons, A.H. Taki, S.G. Hodder, L.D. Jeal, Designing for thermal comfort in combined chilled ceiling/displacement ventilation environments, ASHRAE Transaction 104 (1B) (1998) 901–911.
[4] S. Hu, Q. Chen, L.R. Glicksman, Comparison of energy consumption between displacement and mixing ventilation for different US building and climates, ASHRAE Transaction 105 (2) (1999) 453–464.
[5] P.O. Fanger, B.M. Ipsen, G. Langkilde, B.W. Olsen, N.K. Christensen, S. Tanabe, Comfort limits for asymmetric thermal radiation, Energy and Buildings 8 (1985) 225–236.
[6] R.W. Kulpmann, Thermal comfort and air quality in rooms with cooled ceilings—results of scientific investigations, ASHRAE Transaction 99 (2) (1993) 488–502.
[7] M. Behne, Is there a risk of draft in rooms with cooled ceilings? Measurement of air velocities and turbulences, ASHRAE Transaction 101 (2) (1995) 744–752.

[8] H. Tan, T. Murata, K. Aoki, T. Kurabuchi, Cooled ceilings/Displacement ventilation hybrid air conditioning system—Design criteria, Proceedings of Roomvent '98 1 (1998) 77–84.
[9] S.G. Hodder, D.L. Loveday, K.C. Parsons, A.H. Taki, Thermal comfort in chilled ceiling and displacement ventilation environments: vertical radiant temperature asymmetry effects, Energy and Buildings 27 (1998) 167–173.
[10] T. Imanari, T. Omori, K. Bogaki, Thermal comfort and energy consumption of the radiant ceiling panel system. Comparison with the conventional all-air system, Energy and Building 30 (1999) 167–175.
[11] Z. Jiang, Q. Chen, A. Moser, Indoor airflow with cooling panel and radiative/convective heat source, ASHRAE Transaction 98 (Part 1) (1992) 33–42.
[12] J. Niu, J.V.D. Kooi, Numerical investigations of thermal comfort and indoor contaminant distribution in a room with cooled ceiling system, Proceedings of Indoor Air '93 5 (1993) 331–336.
[13] G. Gan, Towards a better indoor thermal environment—A CFD analysis of the performance of chilled ceiling system, in: Proceedings of the 4th International Symposium on Ventilation for Contaminant Control, 1994, pp. 551–556.
[14] F. Alamdari, N. Eagles, Displacement ventilation and chilled ceilings, Technical Note TN 2/96: 1-32, Building Services Research and Information Association (BSRIA), 1996.
[15] H. Brohus, Influence of a cooled ceiling on indoor air quality in a displacement ventilated room examined by means of CFD, Proceedings of Roomvent '98 1 (1998) 53–60.
[16] M.F. Brunk, Cooling ceilings—an opportunity to reduce energy costs by way of radiant cooling, ASHRAE Transaction 99 (2) (1993) 479–487.
[17] F. Sodec, Economic viability of cooling ceiling systems, Energy and Buildings 30 (1999) 195–201.
[18] J. Niu, J.V.D. Kooi, H.V.D. Ree, Energy saving possibilities with cooled-ceiling system, Energy and Buildings 23 (1995) 147–158.
[19] ASHRAE, 1997 ASHRAE Handbook-Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc., Atlanta, 1997.
[20] M. Behne, Indoor air quality in rooms with cooled ceilings. Mixing ventilation or rather displacement ventilation, Energy and Buildings 30 (1999) 155–166.
[21] ASHRAE, ANSI/ASHRAE standard 55-1992, Thermal environmental conditions for human occupancy, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1992.
[22] K. Fitzner, Displacement ventilation and cooled ceilings, results of laboratory tests and practical installations, Proceedings of Indoor Air'96 1 (1996) 41–50.
[23] F. Alamdari, Displacement ventilation and cooled ceilings, Proceedings of Roomvent '98 1 (1998) 197–204.
[24] J. Niu, J. Burnett, Integrating radiant/operative temperature controls into building energy simulation, ASHRAE Transaction 104 (2) (1998) 210–217.
[25] H. Kruhne, Effect of cooled ceilings in rooms with displacement ventilation on the air quality, Proceedings of Indoor Air '93 5 (1993) 395–400.
[26] J. Niu, J.V.D. Kooi, Indoor climate in rooms with cooled ceiling systems, Building and Environment 29 (3) (1994) 283–290.
[27] S. Yang, J. Srebric, Modeling the performance of different ventilation system on VOC removal in a full-scale room, Proceedings of CLIMA 2000, 2001.
[28] C. Handel, S. Lederer, H.W. Roth, Ventilation for energy efficiency and optimum indoor air quality, in: Proceeding of 13th AIVC Conference, 1992, pp. 433–430.
[29] E.H. Feustel, C. Stetiu, Hydronic radiant cooling—preliminary assessment, Energy and Buildings 22 (1995) 193–205.
[30] C. Stetiu, Radiant Cooling in US Office Buildings: Towards Eliminating the Perception of Climate-Imposed Barriers, Ph.D. Thesis, University of California, Berkeley, 1997.
[31] ASHRAE, ASHRAE Standard 62-1999, Ventilation for acceptable indoor air quality, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1999.
[32] J. Niu, J.V.D. Kooi, H.V.D. Ree, Cooling load dynamics of rooms with cooled ceilings, Proceeding of Building Services and Engineering and Research Technology 18 (4) (1997) 201–207.
[33] N. Matsuki, Y. Nakano, T. Miyanaga, N. Yokoo, T. Oka, Performance of radiant cooling system integrated with ice storage, Energy and Buildings 30 (1999) 177–193.
[34] C.L. Conroy, S.A. Mumma, Ceiling radiant cooling panels as a viable distributed parallel sensible cooling technology integrated with dedicated outdoor air systems, ASHRAE Transaction 107, Part 1, 2001, Paper AT-01-7-5.
[35] M. Behne, The influence of the arrangement of cooling ceilings on thermal comfort, Proceedings of Indoor Air '93 5 (1993) 413–418.
[36] V. Prochaska, B. Kegel, P. Kofoed, Control aspects of displacement ventilation with cooled ceiling, Proceedings of Roomvent '92 3 (1992) 53–68.
[37] K.A. Antonopoulos, M. Vrachopoulos, C. Tzivanidis, Experimental evaluation of energy savings in air-conditioning using metal ceiling panels, Applied Thermal Engineering 18 (1998) 1129–1138.