ELSEVIER

Energy and Buildings 30 (1999) 155–166

ENERGY AND BUILDINGS

Indoor air quality in rooms with cooled ceilings. Mixing ventilation or rather displacement ventilation?

Martin Behne [1]

Lawrence Berkeley National Laboratory, Berkeley, CA 94720, USA

Abstract

Experimental investigations and practical experiences in Europe have proved that hydronic cooled ceilings are able to remove high cooling loads without impairing thermal comfort. As hydronic cooled ceilings cannot remove latent loads and pollutants, e.g., CO_2, VOCs, odors, additional ventilation has to be applied. Often, displacement ventilation is used, which is able to provide lower pollutant levels in the occupation zone than mixing flow systems, if the occupants are causing most of the pollution. Unfortunately, the advantage of the displacement flow, in respect to the air quality, can vanish when cooled ceilings are used to remove the major part of the sensible cooling load. For these applications, a combination of a cooled ceiling with a mixing ventilation system might be more appropriate. This paper presents the results of an investigation examining the distribution of tracer gas in a test chamber which is equipped with a radiant cooled ceiling. There is both a displacement flow system and a mixing flow system available, so that the concentrations of the tracer gas within the occupation zone characterizing the air quality can be compared directly and evaluated under similar conditions. The vertical air temperature rise is taken into consideration as well, as it influences the displacement flow and is an important issue for assessing thermal comfort. The results show the interaction of the portion of the cooling load being removed by the supply air, the air quality in the occupied zone and the vertical air temperature rise. The figures and tables presented show which of the two supply air systems investigated have advantages over the other. © 1999 Published by Elsevier Science S.A. All rights reserved.

Keywords: Indoor; Ceiling; Ventilation

1. Introduction

In the past years, European building developers, architects and engineers are deciding more often to use a radiant cooled ceiling to thermally condition spaces with high cooling loads, as the experiences with cooled ceilings are very persuasive [20]. About 10% of all new installations in German non-residential buildings (mostly buildings for banks and insurance companies) are being built or retrofitted with cooled ceilings to provide the best thermal comfort [23]. When cooled ceilings are properly designed and operated favorable operation costs and more usable building space can be achieved as well [7]. Those advantages when compared with all-air systems are provided only by hydronic radiant cooled ceilings, which only remove sensible cooling loads. Latent loads and pollutants have to be removed by an additional ventilation system. Basically, the windows and doors of a building can be used to supply fresh air to the spaces [18], but preferably, mechanical ventilation should be used to always guarantee an energy efficient air exchange rate [21]. For mechanical ventilation, either displacement ventilation [2] or mixing flow systems can be used, with a supply airflow reduced according to the hygienical requirements [2,8,10].

A German survey [23] shows that about 10% of all cooled ceiling applications (by cooling area) rely on natural ventilation, i.e., these were installed without any additional mechanical ventilation system. Both displacement

[1] Dr.-Ing. Martin Behne is visiting researcher with the E.O. Lawrence Berkeley National Laboratory. The research results were derived when he was research associate with the Hermann-Rietschel-Institut in Berlin, Germany.

[2] In German-speaking countries, the term 'Quelluftströmung' = source flow (or less often 'Schichtenströmung' = layer flow) is commonly used for describing the special air flow pattern happening when colder air with a rather low velocity and turbulence intensity is supplied to the lower part of a room with heat sources to thermally condition a space and to provide the best air quality. These terms describe the actual air flow pattern, usually consisting of a fresh air layer (displacement zone), a transient zone and a mixing zone, more apt than the term displacement ventilation, which is commonly being used in English-speaking countries.

0378-7788/99/$ - see front matter
PII: S0378-7788(98)00083-8

ventilation systems and mixing flow systems are being used by about 45% each to remove the latent load and the pollutants from spaces with cooled ceilings.

Displacement flow systems [9,11,14,22], which often require more effort in terms of design, construction and costs, usually provide best indoor air quality, low air velocities and turbulence intensities. The lowest permissible supply air temperature restricts the cooling capacity of a displacement ventilation system significantly. A cooled ceiling can be applied additionally to satisfy the cooling demand. By using such a 'hybrid-system' one also hopes to get both the advantageous features of cooled ceilings with respect to thermal comfort and the favorable air quality characteristics of the displacement flow pattern. However, adding together two favorable characteristics does not necessarily result in a combination providing both advantages entirely. In fact, the characteristics of a radiant cooled ceiling influences the displacement flow, so that the typical flow pattern of the displacement ventilation might vanish and becomes rather a mixing flow pattern instead [3,15]. From this point of view, an ordinary mixing flow systems might remove the pollutants and latent loads as well.

In order to evaluate the air quality and the thermal comfort being provided by these two different ventilation systems each being combined with a radiant cooled ceiling, a comprehensive experimental investigation [4] in a test chamber was carried out at the Hermann-Rietschel-Institut for Heating and Air-Conditioning [3] of the Technical University of Berlin, Germany. One of the goals of this study was to point out the boundary conditions, when a displacement ventilation system combined with a cooled ceiling can comfortably provide better air quality in the occupied zone than a mixing airflow system.

This report presents major results of the air quality investigation considering the thermal comfort level provided as well, since these two features are related to each other when a displacement ventilation system is being used in spaces with high cooling loads.

2. Evaluation of air quality

The air quality being evaluated in this report is only the distribution of pollutants released by people, e.g., CO_2, odor, tobacco smoke. [4] Pollutants, which are supplied by the air-conditioning system, e.g., defective filters, or are released by the carpets or furniture, e.g., solvents, formaldehyde, spread out differently in a space, so the distribution of these pollutants cannot be evaluated by using the results presented here.

For simulating the pollutants released from people, measurements with the tracer gas Dinitrogenoxide (N_2O) were made under steady-state conditions.

The relative air quality in space in steady-state conditions, can be characterized by the contaminant removal efficiency, μ_x, [5] which represents the ratio between the concentration, e.g., of a tracer gas or particles, at one particular point and the concentration in the exhaust air. It is defined as follows:

$$\mu_x = \frac{c_x - c_{supply}}{c_{exhaust} - c_{supply}} \tag{1}$$

where c_x is the measured concentration at the location x, c_{supply} is the measured concentration in the supply air and $c_{exhaust}$ is the measured concentration in the exhaust air.

When the supply air and the room air are mixed entirely, the concentration, e.g., of a particular gas, will be the same at each location in the room representing ideal mixing flow. In that case, the concentration in the exhaust air equals the concentration at every spot in the room and all locations in a space will have a contaminant removal efficiency of $\mu_x = 1.0$. Therefore, the contaminant removal efficiency, μ_x, reflects the ratio between the concentration at one point of the space with a certain airflow pattern and an ideal mixing flow. Values below 1.0 indicate lower concentration values (= better air quality) than an ideal mixing airflow would provide. Values above 1.0 indicate that some contaminated room air is not exchanged by fresh supply air, but stays in the room instead (short circuiting).

Unfortunately, a real mixing ventilation system is not always able to achieve an ideal mixing of the supply air and the room air. The mixing process is mostly influenced by the location and type of the supply air registers and the momentum of the supply air (supply airflow), as well as the locations and intensities of the heat sources inside the room. Especially when mixing ventilation is added to a cooled ceiling system, only small supply airflows are required, which can very likely lead to a balance between the momentum of supply air and the buoyant airflow of the heat sources. As a result, the mixing of the fresh supply air

[3] Hermann-Rietschel-Institut für Heizungs- und Klimatechnik, Marchstraße 4, D-10587 Berlin, Germany. Tel.: +49-30-31424170; Fax: +49-30-31421141; Internet address: http:\\dynamik.fb10.Tu-berlin.de\~hri.

[4] Smoking is usually still permitted in German office buildings, although changes by law might be introduced in the near future.

[5] In Europe, the term ventilation efficiency is usually being used instead of contaminant removal efficiency [8]. In Germany, one uses the term 'Kontaminationsgrad' = contamination efficiency instead of contaminant removal efficiency to characterize the relative air quality in a space. All three terms mentioned describe the same air quality feature, which is also the inverse of the contaminant removal effectiveness (ventilation effectiveness, in Europe, or 'Lüftungseffektivität', in Germany) when there are steady state conditions.

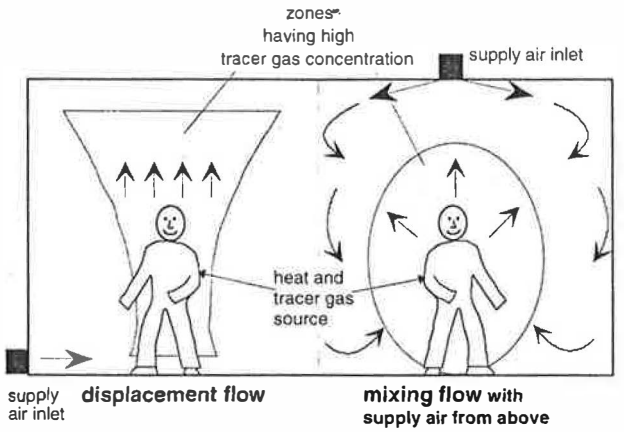


Fig. 1. Spreading of tracer gas near the manikins being both the heat and tracer gas source, with displacement flow and mixing flow.

and the contaminated room air is even more dependent of the heat sources, and locally the contaminant removal efficiency will be more often above 1.0.

To compare directly the air quality with displacement flow and with (real) mixing flow, the ratio of the contaminant removal efficiencies derived for both ventilation systems under comparable boundary conditions is introduced. This ratio is the 'relative contamination efficiency, χ_v' and is defined as follows:

$$\chi_v = \frac{\mu_{v,\text{source}}}{\mu_{v,\text{mixing}}} \tag{2}$$

where $\mu_{v,\text{source}}$ is the contaminant removal efficiency at a certain location with displacement flow and $\mu_{v,\text{mixing}}$ is the contaminant removal efficiency at the same location, but with mixing flow instead.

When the relative contamination efficiency, χ_v, is smaller than 1.0, then the air quality provided by the displacement ventilation system is better than with the mixing flow system applied. Values above 1.0 usually indicate an incomplete mixing of supply air and room air, when the mixing flow system is being operated.

Fig. 1 schematically shows different distributions of pollutants released by a person due to the two different ventilation principles, displacement flow and mixing flow.

3. Air quality and the vertical air temperature rise

When displacement ventilation is used the thermal plume of a person transports the pollutants released by the person itself to the upper zones. If the buoyant airflow of the person is smaller than, or equal to, the supply airflow, the pollutants will be removed entirely from the occupied zone. This usually represents very good air quality in the occupied zones.

However, as the buoyant airflow of a heat source increases (Fig. 2) with the height [9,11,13], there will always exist a zone (often a horizontal layer) where the buoyant airflow exceeds the supply airflow. Within that zone (layer) the airflow pattern changes from a fresh air layer (displacement zone) into a mixing flow pattern affecting the distribution of pollutants as well as the air temperature distribution. In this case, the measured contaminant removal efficiency can be used to indicate the changing of the airflow pattern.

The left part of Fig. 3 shows typical profiles for a displacement airflow pattern having a fresh air layer from the floor up to 1.0 m, a transition zone between 1.0 and 1.7 m and a mixing zone above. The contaminant removal efficiency increases rapidly to values above 1.0 within the transition zone, pointing out this particular zone. The distinct fresh air layer shown incorporates the best air quality, but requires also a significant vertical air temperature rise [3,17].

In order to provide the best thermal comfort, not only the indoor temperature level has to be within a certain range and the velocities and turbulence intensities must not bother the occupants, but also the vertical air temperature rise in the occupied zone must stay small [1,8,10].

When a space is thermally conditioned only by a displacement ventilation system, vertical air temperature differentials will appear depending on the thermal load and the specific supply airflow ($m^3/(h\ m^2)$). If the vertical air temperature difference between 1.1 and 0.1 m, $\Delta t_{0.1/1.1\ m}$, exceeds 2.0 K, more than 15% of all occupants might not feel comfortable, which means the best thermal comfort is not provided anymore [19,21]. Fig. 4 shows the percentage of dissatisfied persons due to the vertical air temperature difference between 1.1 and 0.1 m and the duration of stay. The longer a person is staying in a space with a remarkable vertical air temperature difference, the less likely he or she will complain about it. However, to avoid most complaints, the more critical curve for a stay of 1.5 h is used in the common standards for determining thermal comfort [1,8,10].

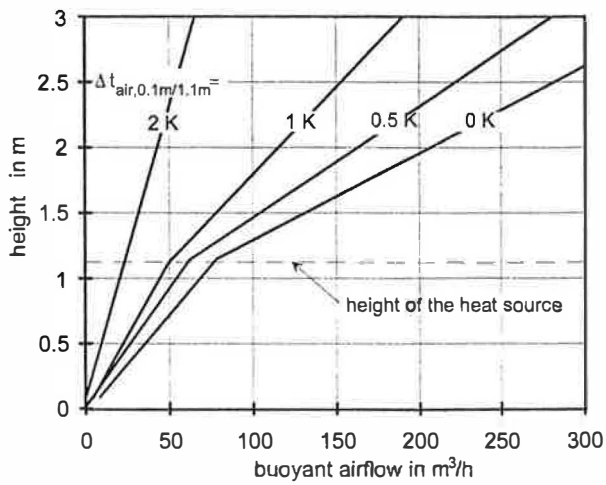


Fig. 2. Buoyant airflow of a sedentary person (heat released = 100 W) as a function of the height and the vertical air temperature difference [11].

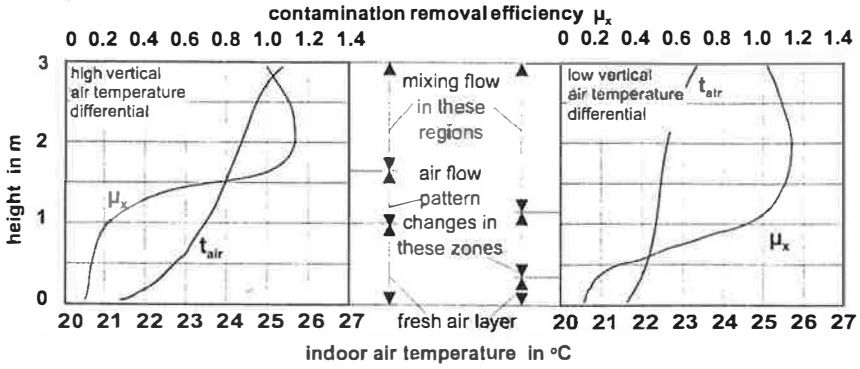


Fig. 3. Profiles of air temperature and tracer gas concentration with displacement flow with 'high vertical air temperature differential' (no cooled ceiling, left) and with 'low vertical air temperature differential' (with cooled ceiling, right).

With a mixing flow system, thermal loads of up to 100 W/m^2 can be removed comfortably, but rather high supply airflows and momentums are required, which are more likely to cause noise and draft in a space than the low velocities and turbulence intensities occurring with displacement ventilation.

Due to thermal comfort issues, the highest removable cooling load with a single displacement ventilation system in offices is limited to about 30 W/m^2. [6] To comfortably remove higher cooling loads from a space, displacement ventilation can be combined with a cooled ceiling. Problems with draft will not occur with cooling loads up to 100 W/m^2 [5]. When the cooled ceiling is removing the major part of the cooling load, the vertical temperature distribution will be more uniform when compared with a single displacement ventilation system. Therefore, a cooled ceiling and a displacement ventilation system can provide thermal comfort in spaces with rather high thermal loads. But at the same time, the buoyant airflow of the heat sources will be increased (Fig. 2), which is lowering the zone where the fresh air layer changes to the mixing airflow pattern (Fig. 3).

The right part of Fig. 3 presents the vertical profiles of the indoor air temperatures and the contaminant removal efficiency for a space with a cooled ceiling and a displacement ventilation system. Apparently, the air quality in the occupied zone with the cooled ceiling is not necessarily as good as with a distinct displacement airflow pattern, but one also has to consider that a single displacement ventilation system supplying 'only' the required fresh air rate would not be able to provide good thermal comfort.

A mixing flow system can be combined with a cooled ceiling as well. This usually decreases the likelihood of uncomfortable velocities and turbulence intensities due to the lower supply airflows. However, it remains uncertain whether a real mixing ventilation system in conjunction with a cooled ceiling would be able to provide the same air quality level as a displacement ventilation system, when this is been combined with a cooled ceiling.

4. Experimental set-up and variations investigated

Indoor air quality and thermal comfort were measured and evaluated under different conditions in a test chamber equipped with a cooled ceiling. The chamber was situated in a big heatable test hall with uniform air temperatures. The walls of the test chamber were insulated and all measurements were made under steady-state conditions. The test chamber was vented (supply airflow ≈ 390 m^3/h; air exchange rate ≈ 2.9/h) by either a displacement flow system with the supply air inlet along one of the walls and on the floor or by a mixing airflow systems having two

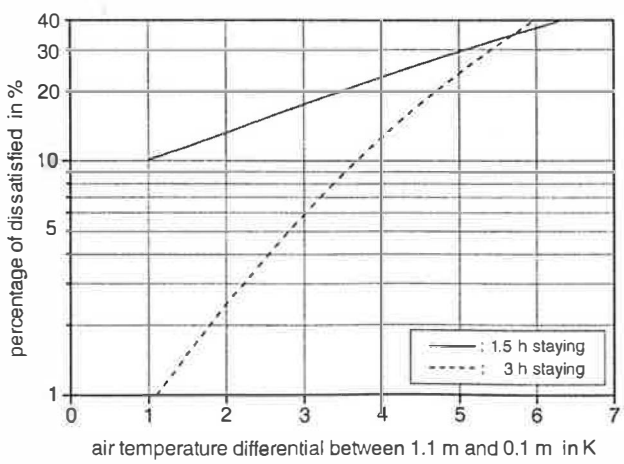


Fig. 4. Percentage of dissatisfied persons as a function of the vertical air temperature differential, $\Delta t_{0.1/1.1\,m}$, and the duration of stay [21].

[6] The supply airflow of displacement ventilation systems is commonly set by the outdoor air requirements. The supply air temperature should usually not be lower than 8 K when compared with the mean room air temperature. With that, 30 W/m^2 is the highest comfortably removable cooling load, if an office provides 5 m^2 per person and the fresh air required is 60 m^3/(h person).

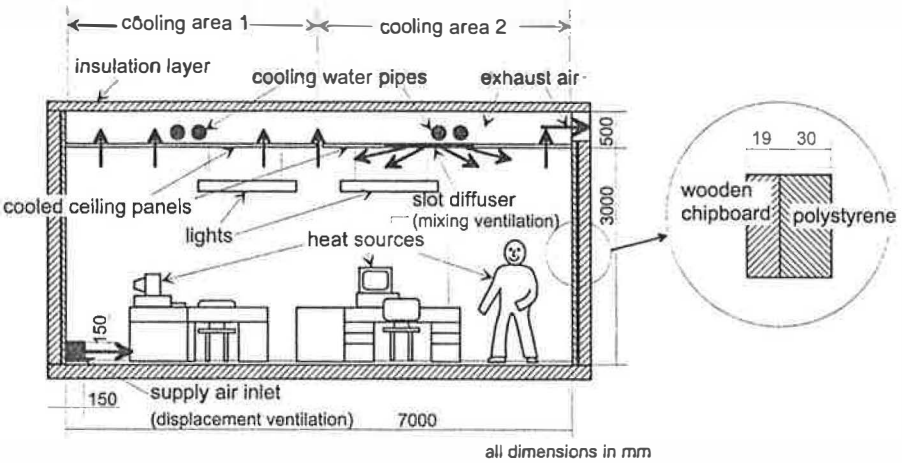


Fig. 5. Cross-section and construction of the wall of the test chamber.

slot diffusers mounted in between the cooled ceiling panels. Fig. 5 presents a cross-section of the test chamber and Fig. 6 shows the floor plan and a reflected ceiling plan. The positions of the supply air devices are stated.

The cooled ceiling investigated was a hydronic panel type and covered about 90% of the entire ceiling area. The portion for heat exchange via radiation was about 45%. The exhaust air was leaving the room through slots between the ceiling panels.

In real buildings, it is not always possible to cover the entire ceiling with cooling panels (or cold water pipes embedded in plaster), as other installations, e.g., lights or construction components, take up some space. In order to be able to investigate the impact of a partly-cooled ceiling on the airflow pattern, the air quality and thermal comfort, the total area of the cooled ceiling was divided into two equally sized parts with individual cold water cycles. Due to this, the cold water flow through each of these cooling areas could be controlled individually and either the entire cooled ceiling or just one half of it was cooled.

Additionally, the heat sources in the test chamber were arranged in three different configurations, so that several

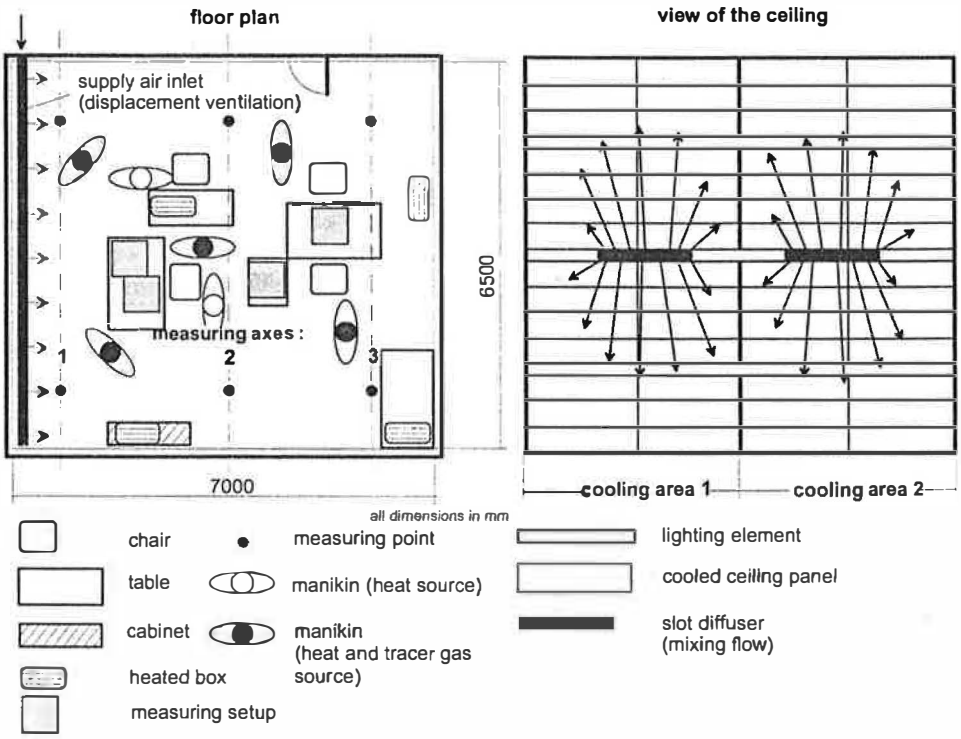


Fig. 6. Floor plan and reflected ceiling plan of the test chamber, showing the heat source arrangement S1,2 and explaining the measuring points.

different arrangements of cooling load [7] (movable heat sources) and cooled ceiling area could be investigated. Table 1 shortly presents the variations investigated. Heated manikins and 'boxes' (simulating computers or printers) were used as heat sources. The movable heat sources released about 75% of the entire cooling load. The rest was being released by the lighting (~ 17%) and the measurement equipment (~ 8%).

Five out of seven heated manikins (movable) were used as tracer gas sources, releasing the tracer gas from below their clothes at a height of about 1 m. The tracer gas concentration and the air temperature was measured at six locations (three axes; see Fig. 6) and different heights. The tracer gas concentration was determined by an infrared gas detector (accuracy ~ 3%) and the temperatures were measured with thermocouples (NiCr–Ni; accuracy of the entire set-up: ~ 0.1 K). The analog measurement signals were digitalized by a data-logger and then controlled online, analyzed, recorded and stored with a PC.

The portion of cooling load removed by the ventilation system ω_{supply} and the cooled ceiling $\omega_{ceiling}$ are calculated according to the following equations:

$$\omega_{supply} = \frac{\dot{Q}_{supply}}{\dot{Q}_{heatsources} - \dot{Q}_{envelope}} \tag{3}$$

$$\omega_{ceiling} = 1 - \omega_{supply} \tag{4}$$

where $\dot{Q}_{heatsources}$ is the electrical input (= heat released) of the heat sources

$$\dot{Q}_{supply} = \dot{V}_{supply} \cdot \rho_{air} \cdot c_{p,air} \cdot \left(\bar{t}_{air,2.9\,m} - t_{supply}\right) \tag{5}$$

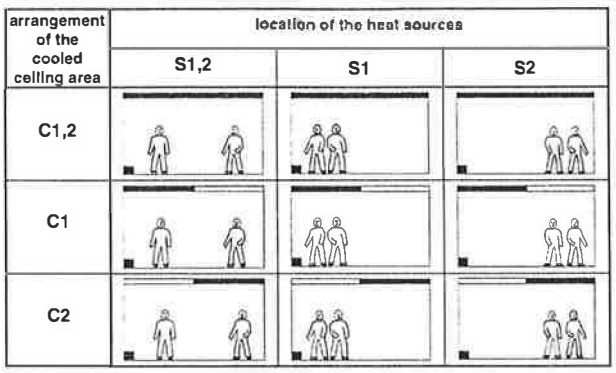
(cooling load removed by the supply air)

$$\dot{Q}_{envelope} = U_{wall} \cdot \left(\bar{t}_{air} - \bar{t}_{o}\right) + U_{floor} \cdot \left(\bar{t}_{air} - \bar{t}_{air,base}\right) \tag{6}$$

(heat entering/leaving the test chamber through the walls and the floor), where $\dot{V}_{supply}$ is the supply airflow, ρ_{air} is the density of the air, $c_{p,air}$ is the heat capacity of the air, $\bar{t}_{air,2.9\,m}$ is the mean indoor air temperature at a height of 2.9 m, $\bar{t}_{supply}$ is the supply air temperature, U_{wall} is the mean heat transfer coefficient of the walls (≈ 0.85 W/m² K), U_{floor} is the mean heat transfer coefficient of the floor (≈ 2.5 W/m² K), $\bar{t}_{air}$ is the mean indoor air temperature, $\bar{t}_{o}$ is the mean air temperature of the environment (test hall) and $\bar{t}_{air,base}$ is the mean air temperature in the basement.

The portions of cooling load removed by the ceiling and the displacement ventilation system were adjusted in many preliminary tests to obtain both reasonable vertical air temperature rises and mean contaminant removal efficiencies in the occupied zone below 1.0. The final variations seem to incorporate a good compromise for both characteristics. The portions of the cooling load removed by the supply air was not always the same for all variations. When the entire cooled ceiling was being cooled, about 80–85% of the cooling load was removed by the cooled ceiling and just 15–20% by the air. When only one half of the ceiling was being cooled, the portion of the supply air was increased to about 25–30% to provide acceptable indoor air conditions.

Table 1
Nine variations of both cooled ceiling area arrangement and heat source locations investigated (the dark parts of the ceiling indicate the cooled part and the manikins point out the heat source arrangement)



The meaning of the 'names' of the columns and rows used are as follows: (Rows) C1,2: both cooled ceiling areas 1 and 2 are being cooled; C1: only cooled ceiling area 1 is being cooled; C2: only cooled ceiling area 2 is being cooled. (Columns) S1,2: heat sources are located below the cooled ceiling areas 1 and 2; S1: heat sources are mainly situated below cooled ceiling area 1; S2: heat sources are mainly situated below cooled ceiling area 2.

The adjustment of a mixing ventilation system and cooled ceiling is not as critical as with the displacement ventilation system. However, the cooled ceiling should remove the major portion of the sensible load to provide a high thermal comfort level. In order to get comparable results, the portions of the cooling load removed by the mixing ventilation system were set similar (15–35%) to the respective variation with the displacement ventilation system.

To better compare the air temperature distribution of all the variations, the local air temperature difference $\Delta t_{air,x}$ is introduced. It is defined as follows:

$$\Delta t_{air,x} = t_{air,x} - \bar{t}_{air} \tag{7}$$

where $t_{air,x}$ is the air temperature at one particular measuring point x and $\bar{t}_{air}$ is the mean air temperature inside the test chamber.

5. Results

The major influences on the airflow in the test chamber are illustrated by presenting two of the nine different

[7] The cooling load was varied between 40 and 65 W/m² while also varying the portions of the cooling load removed by the ventilation system or the cooled ceiling, to find out which cooling loads can be removed without impairing thermal comfort. The cooled ceiling removed between 70 and 95% of the entire cooling load of the test chamber.

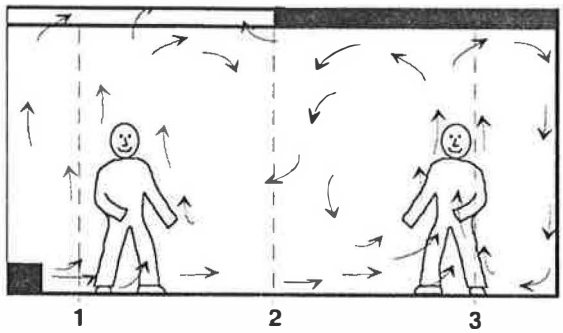


Fig. 7. Schematic airflow in the test chamber with displacement ventilation for the variation C2_S1,2.

variations. The results of the variations C2_S1,2 and C1,2_S1 will be described for both displacement ventilation and mixing ventilation. However, as the mixing ventilation does not create a rather distinctive air movement in the space, only the airflow pattern with displacement ventilation will be shown [8] schematically (Figs. 7 and 10).

The following Figs. 9 and 11 present the profiles of the local air temperature difference $\Delta t_{air,x}$ for the tests with the cooled ceiling and displacement ventilation, as only this combination causes significant temperature profiles. The air temperature distribution with mixing flow was always almost uniform. Figs. 8 and 12 present the relative contamination efficiency directly comparing the tracer gas concentrations with displacement flow and mixing flow. All these figures show the data vs. the height for three different axes, where each point in the graphs represents the average of two values measured in one respective height (compare Fig. 6).

5.1. Variation C2_S1,2

The contaminant removal efficiencies measured with displacement flow are lower than with mixing ventilation in the entire room (Fig. 8). The data of the lower region (0.1 to 1.0 m) indicate that the fresh air was supplied to all axes, although the tracer gas concentration with displacement flow increases towards the measuring axis 3 due to recirculating room air (contaminated) from the ceiling (see Fig. 7).

The uneven mixing of supply air and room air when mixing ventilation was used is one of the reasons for the relatively better air quality at all heights with displacement ventilation. While the displacement flow resulted in contaminant removal efficiencies of 0.05 to 1.1 (0.1 to 1.7 m height), the corresponding values with mixing flow were between 1.1 and 1.4, with the higher values below the

[8] The airflow was observed by injecting fog into the supply air duct. This method of visualizing the airflow pattern works very well with displacement ventilation. The arrows shown in Figs. 7 and 10 depict the way the air moved in the test chamber.

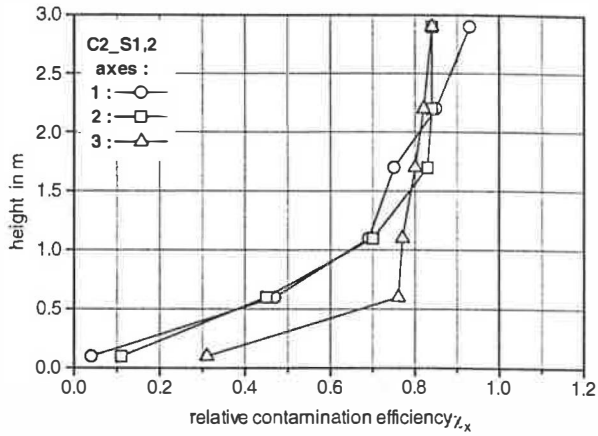


Fig. 8. Relative contamination efficiency at three measuring axes for the variation C2_S1,2.

cooled ceiling area. This indicates for the tests with mixing flow that the fresh air was not entirely being mixed with the room air rising at the heat sources.

This incomplete mixing was always observed when only one half of the ceiling was cooled. Although the flow visualization did not always indicate such an influence and although the results for all variations are varying, it seems to be true that the room air in the zones underneath the cooled ceiling area got less mixed with fresh air than in the other zones. Apparently, the cooled ceiling seem to influence the mixing mechanism stronger than the buoyant airflows of the heat sources, which is quite surprising. However, short circuiting in parts between the tracer gas sources and the measuring points might temporarily have happened as well, as the spreading of the tracer gas in the area around the manikins is very difficult to predict with mixing flow.

The profiles of the air temperature difference with displacement ventilation shown in Fig. 9 point out that the colder supply air warmed up a bit along its way to axis 3 being both convectively heated by the warmer floor and

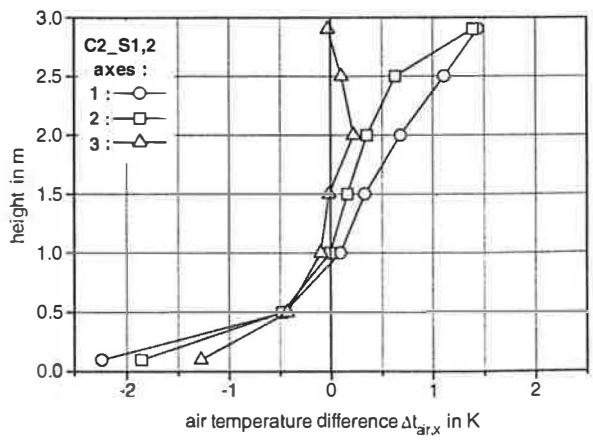


Fig. 9. Vertical air temperature profile at three measuring axes for the variation C2_S1,2 with cooled ceiling and displacement ventilation.

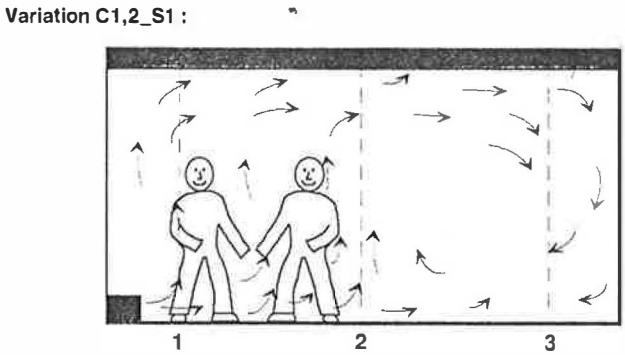


Fig. 10. Schematic airflow in the test chamber with displacement ventilation for the variation C12_S1.

partly mixed with the warmer room air. The comparison of the different air temperature profiles above 1.0 m height shows almost uniform air temperatures in the part of the room under the cooled ceiling and rising temperatures at the axes 1 and 2. The vertical air temperature rise measured at axes 1 and 2 for this variation corresponds to about 15% dissatisfied occupants (compare Fig. 4) and approaches the limit for thermal comfort.

The airflow visualization with fog showed that the cooled ceiling area (above axis 3) forced the rising room air to recirculate to the occupied zone by convective heat exchange at the ceiling's surface. This was causing more uniform air temperatures on the one hand and higher contaminant removal efficiencies at axis 3 on the other hand (compare Fig. 8). The cooled part of the ceiling (cooling area 2) appeared to be not as permeable for the rising room air as the passive part. As a result, the biggest part of the exhaust air passed the uncooled ceiling area to leave the test chamber (Fig. 10).

5.2. Variation C1,2_S1

Fig. 11 shows lower tracer gas concentrations with displacement flow only close to the floor and in 0.5 m

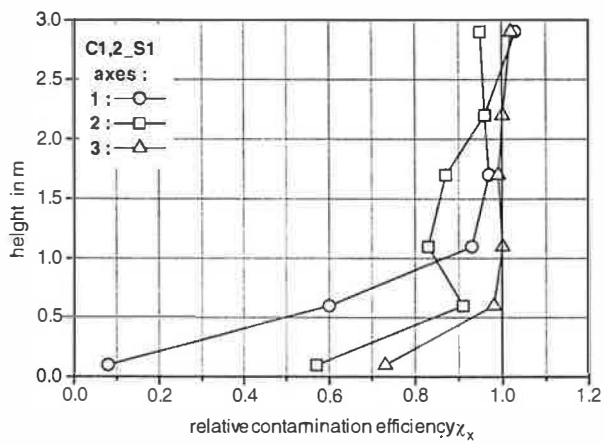


Fig. 11. Relative contamination efficiency at three measuring axes for the variation C1,2_S1.

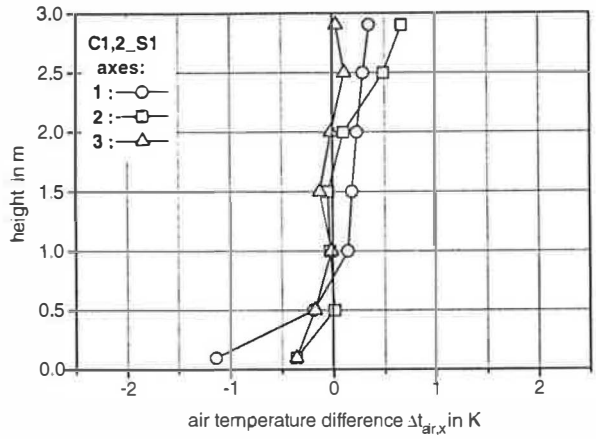


Fig. 12. Vertical air temperature profile at three measuring axes for the variation C1,2_S1 with cooled ceiling and displacement ventilation.

height at axis 1. The profiles indicate that almost the entire fresh air is being drawn up by the thermal plume of the heat sources, which were mostly situated close to the supply air inlet. The rest of the room (except the floor area) did not get fresh air, but already mixed air. Therefore, all other relative contamination efficiencies are very close to 1.0, indicating that there is no significant difference between the displacement flow and mixing flow in terms of air quality.

In fact, a distinct recirculation of rising room air to the lower part of the room was observed with displacement flow, especially at the walls. This marked recirculation is typical for the airflow pattern in a space with a cooled ceiling removing the major portion of the cooling load and occurs due to three reasons. First, the convective heat exchange at the cooled ceiling cools the rising air, which makes it to flow back down. Second, the radiant heat exchange cools the walls, causing buoyant airflows (downwards). Third, the buoyant airflow of the heat sources amounts to much more than the supply airflow [9] (= exhaust airflow) so that, a remarkable flow of room air has to recirculate in the test chamber. These mechanisms are correlated with each other, so it is hard to tell which influences or causes the other.

The slot diffusers (mixing flow) forced the introduced supply air mainly to flow along the ceiling and then down along the walls, so that it was not possible to distinguish

[9] With Fig. 2, one can roughly estimate the maximum buoyant airflow of the heat sources. With an assumed vertical air temperature rise of 0.5 K (Fig. 12), a person causes a buoyant airflow in 3.0 m height of about 250 m^3/h. There are 7 manikins adding up to about 1750 m^3/h (the other heat sources are not considered, but those would increase the buoyant airflow further). Although, this number might not be absolutely true as there is always interaction between the heat sources and other surfaces [13], it clearly shows that the buoyant airflow very likely is much higher than the supply airflow of about 400 m^3/h and thus, recirculation has to take place.

whether the cooled ceiling supported this air movement additionally. Unlike the variations with only one half of the ceiling being cooled, the fresh air was mixed very well with the room air, causing contaminant removal efficiencies of 1.0 to 1.1 at all places for all variations with entirely cooled ceiling area.

The rather uniform air temperature profiles presented in Fig. 12 prove that, the supply air was mixed with the room air very well, although displacement ventilation was operated. Only very close to the supply air inlet (lowest points at axis 1), a significantly cooler air temperature was measured.

The vertical air temperature differentials between 1.1 and 0.1 m with entirely cooled ceiling area and displacement ventilation were always below 1.5 K pointing at good thermal conditions. The dominating characteristic of the cooled ceiling equalized the airflow pattern, so that almost no local differences appeared. The arrangement of the heat sources did not influence the temperature distribution significantly.

6. Discussion

The results presented make clear that when the air quality provided with displacement ventilation and a cooled ceiling is compared with the one achieved with mixing flow and a cooled ceiling, one has to consider the thermal conditions as well. By evaluating the data of both the contaminant removal efficiency and the vertical air temperature rise of the many different results for the variations with displacement ventilation, an interaction between these characteristics becomes apparent.

As mentioned before, the portion of the cooling load removed by the cooled ceiling strongly influences the airflow pattern in a room with a displacement ventilation system. Monitoring the vertical air temperature profiles, which again involves the buoyant airflow of the heat sources, and the tracer gas distributions, helps to explain the impact of the cooled ceiling on the airflow pattern.

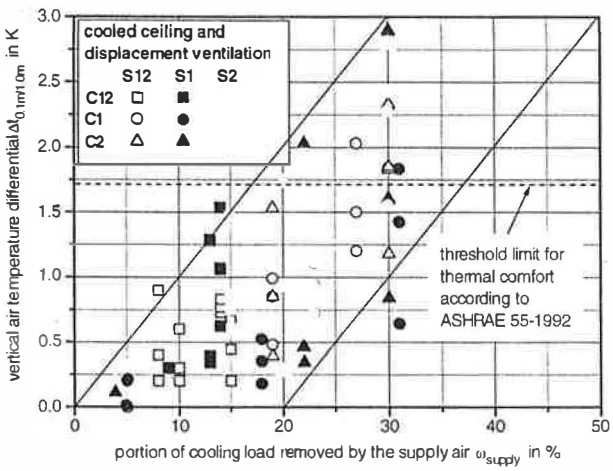


Fig. 13. Vertical air temperature differential between 1.1 and 0.1 m as a function of the portion of cooling load being removed by the supply air and the test variation.

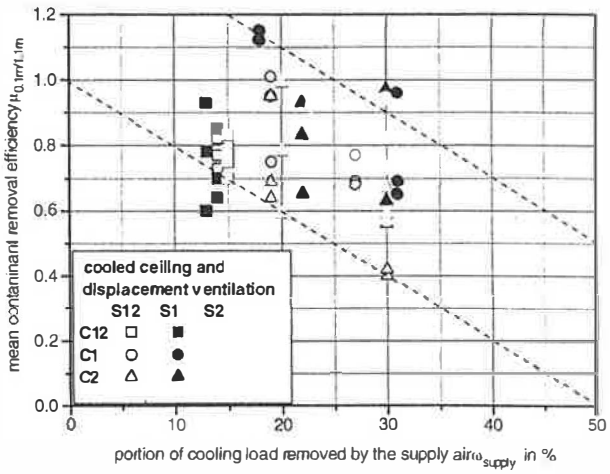


Fig. 14. Mean contaminant removal efficiency of the occupied as a function of the portion of cooling load being removed by the supply air and the test variation.

Fig. 13 presents the vertical air temperature difference between 1.0 and 0.1 m, $\Delta t_{0.1/1.0\ m}$, for the different arrangements vs. the portion of cooling load removed by the supply air ω_{supply}. To assign the results to a particular test variation different symbols are used, in which each point represents the temperature differential at one particular axis with one of the variations. [10] Although the results of the many different variations vary quite a bit, it can be seen that the more cooling load the supply air removes, the higher the vertical air temperature difference becomes in general. This means, that to provide good thermal conditions with displacement ventilation, the cooling load removed by the ventilation system has to be limited.

Fig. 14 presents the mean contaminant removal efficiency, $\mu_{0.1/1.1\ m}$, for all variations investigated with displacement ventilation as a function of the portion of cooling load removed by the supply air. [11] The points depict the average of all measurements at one particular axis and the heights 0.1, 0.5 and 1.1 m. Although the dependence of the mean contaminant removal efficiency to the portion of cooling load removed by the supply air is not as obvious as for the vertical air temperature (shown in Fig. 12), the influence of the portion of the cooling load removed by the supply air can be suggested. This assumption is also been supported by other experimental results [3,15].

[10] Example: with variation C1_S1 (•), $\Delta t_{0.1/1.0\ m}$ ranged between 0.7 (axis 3) and 1.8 K (axis 1), when ω_{supply} was 31%.

[11] Example: with variation C1_S1 (•), $\mu_{0.1/1.1\ m}$ ranged between 0.65 (axis 1) and 0.96 (axis 3), when ω_{supply} was 31%.

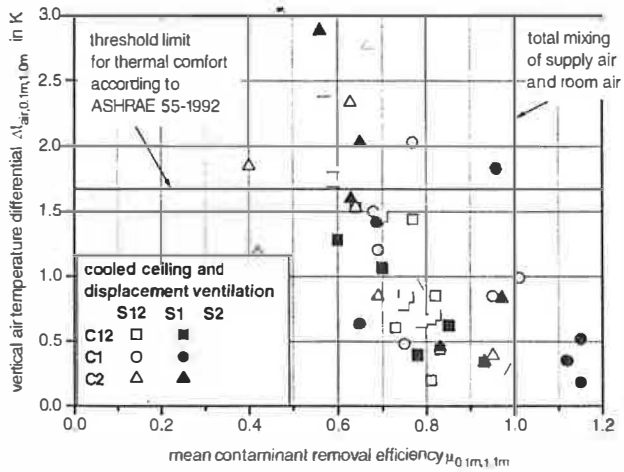


Fig. 15. Vertical air temperature differential vs. the mean contaminant removal efficiency for all variations investigated with displacement ventilation.

Finally, the results of Figs. 13 and 14 are combined in Fig. 15 to present the vertical air temperature difference, $\Delta t_{0.1/1.0\ m}$, vs. the mean contaminant removal efficiency, $\mu_{0.1/1.1\ m}$, when displacement ventilation and the cooled ceiling are operated.

The low mean contaminant removal efficiencies indicating good indoor air quality in the occupied zone are very often associated with 'high' vertical air temperature differences, pointing at a possible problem with thermal comfort. The less critical variations are obviously the ones with an entirely cooled ceiling. These arrangements provided always acceptable vertical air temperature differences and better air quality with displacement flow than with mixing ventilation.

If it is not possible to cover the entire ceiling with cooling panels or if the cooling panels cannot be distributed regularly, the cooled ceiling area should be arranged close to the displacement ventilation inlet rather than on the opposite side. Such an arrangement offers better thermal comfort and the air quality might be better than with a mixing flow system as well. However, the arrangement of the heat and pollutant sources influence the airflow pattern significantly.

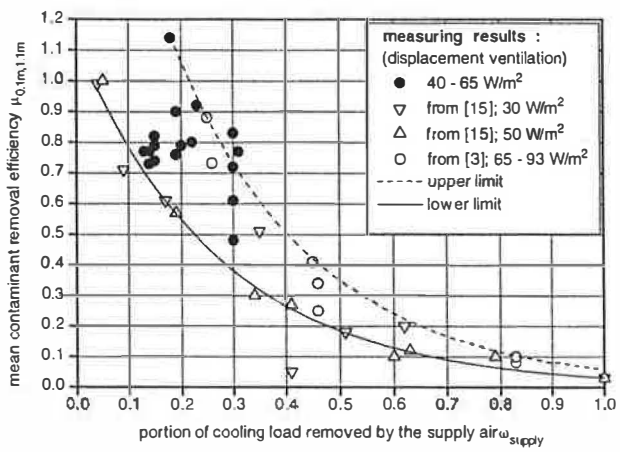


Fig. 16. Mean contaminant removal efficiency as function of the portion of cooling load being removed by the supply air, when displacement ventilation is being used with a cooled ceiling.

Table 2
Experimental coefficients for Eq. (8)

	Lower limit	Upper limit
A	1.13	0.12
B	−3.61	−3.70
C	0	−0.79

The following Fig. 16 summarizes the results from different studies to show the clear impact of the portion of the cooling load removed by the displacement ventilation system ω_{supply} on the mean contaminant removal efficiency $\mu_{0.1/1.1\ m}$ in the occupied zone. The curves for the lower and upper limits were derived with curve fitting and are described by the following equation (Table 2):

$$\overline{\mu}_{0.1/1.1\ m} = A \cdot e^{(B(\omega_{supply} + C))}. \quad (8)$$

One issue has not yet been mentioned: Several experimental investigations have shown that the air quality around a person is normally better than the air quality farther away, if displacement ventilation is being used [6,12,15,16]. Although most studies did not investigate combinations of cooled ceiling and displacement flow systems, a slight improvement of air quality in the 'boundary layer' of a person can also be measured with such systems [4,15].

7. Conclusion

Considering the results presented, none of the ventilation systems investigated can be recommended to supplement a cooled ceiling without carefully considering the pros and cons.

If the air quality in the occupied zone is top priority and the cooling capacity of a single displacement ventilation system is not satisfying the load, a cooled ceiling can be combined with a displacement ventilation system. Pollutants released by persons are removed from the occupied zone to a great extent and good air quality can be achieved associated with comfortable thermal conditions. However, the portions of cooling load removed by both of the components have to be adjusted properly. For office buildings, good results for both thermal comfort and air quality can be expected when the displacement ventilation system removes about 20–25% of the cooling load of a space. When the cooling capacity is exceeding the design conditions, the cooling capacity of the displacement ventilation system should not be increased to maintain thermal com-

fort continuously. Mixing air quality is very likely to occur and ought to be tolerated in such a case.

A mixing ventilation system can provide uniform pollutant distribution at best. If the favorable characteristics of a radiant cooled ceiling with respect to thermal comfort are most important, an air-conditioning concept can be realized with mixing ventilation and a cooled ceiling. Usually such a system will not involve any problems with thermal comfort when the cooling load does not exceed 100 W/m^2 [4]. But one has to make sure that the supply air diffusers are properly designed, situated and adjusted to avoid any draft problems. To obtain the best possible mixing airflow pattern, the entire ceiling ought to be covered by cooled areas. If this cannot be done, the mixing of fresh air and supply air might not be the best at all times and places and the air quality in the occupied zone can be varying.

Appendix A

To illustrate the possible combinations of ventilation system and cooled ceiling the following Fig. 17 and Table 3 were created. For this purpose, some assumptions had to be made: The cooling capacity of a radiant cooling ceiling usually does not significantly exceed 70 W/m^2. Thus, this value and results from Eq. (8) as well as a temperature differential between exhaust air and supply air of 6 K (displacement flow) or 12 K (mixing flow) were used to draw the lines and areas in Fig. 17. Cooling loads beyond 100 W/m^2 are not considered, as it is not possible to exclude problems with thermal comfort totally. In conjunction with Table 3, Fig. 17 allows one to estimate the air quality when a cooled ceiling is combined with a mechanical ventilation system.

A.1. Example for using Fig. 17 and Table 3

An open plan office provides 8 m^2 for each working place and the outdoor air requirement shall be 60 m^3/h per person. This results in a supply airflow of 7.5 m^3/(h m^2).

The cooling load may consist of heat released by people (10 W/m^2), computers (25 W/m^2), lighting (10 W/m^2) and an exterior heat gain (20 W/m^2) and amounts to 55 W/m^2.

For this data, a cooled ceiling combined with either mixing flow or displacement flow would be able to thermally condition the space (area ② in Fig. 17). Displacement flow could provide a slightly better air quality in the occupied zone than a mixing flow system (Table 3).

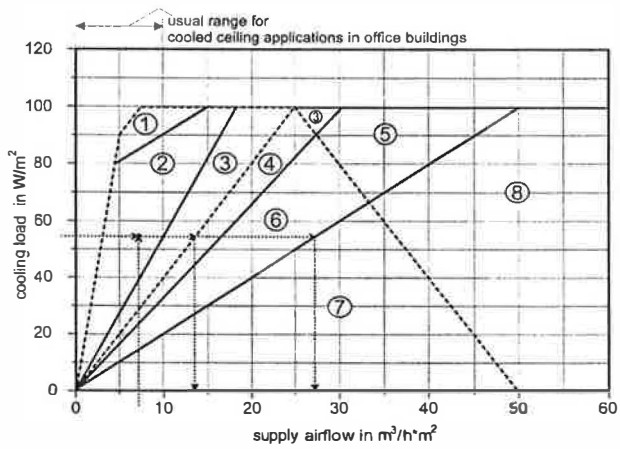


Fig. 17. Chart to determine the possible combinations of cooled ceiling with either displacement or mixing ventilation (compare Table 3).

Table 3
Features of areas shown in Fig. 17

area	combination of HVAC-components	portion of cooling load removed by the supply air ω_{supply}	mean contaminant removal efficiency in the occupied zone
①	CC and MV	20 - 50 %	$\mu \geq 1.0$
②	CC and MV	20 - 55 %	$\mu \geq 1.0$
	CC and DV	10 - 25 %	$\mu \geq 0.9$
③	CC and DV	25 - 50 %	$0.5 > \mu \geq 0.9$
④	MV	100 %	$\mu \geq 1.0$
⑤	CC and DV	50 - 100 %	$0.1 > \mu \geq 0.5$
⑥	MV	100 %	$\mu \geq 1.0$
	CC and DV	50 - 100 %	$0.1 > \mu \geq 0.5$
●	DV	100 %	$\mu \leq 0.1$
	MV	100 %	$\mu \geq 1.0$
⑧	DV	100 %	$\mu \leq 0.1$

CC: cooled ceiling.
MV: mixing ventilation.
DV: displacement ventilation.

A.2. For comparison

If an all-air-system was supposed to remove the loads from the space, the supply airflow would have to be increased to about 14 m^3/(h m^2) (with mixing ventilation) or even up to 28 m^3/(h m^2) when displacement ventilation ought to be used.

References

[1] ASHRAE Standard 55-1992, Thermal Environmental Conditions for Human Occupancy, American Society of Heating, Refrigeration and Air Conditioning Engineers, 1992, Atlanta, GA.
[2] ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality, American Society of Heating, Refrigeration and Air Conditioning Engineers, 1992, Atlanta, GA.
[3] M. Behne, Luftführung im Operationssaal—Untersuchung eines OP-Klimasystems mit Verdrängungsströmung und Deckenkühlung Heizung-, Lüftung-, Klima- und Haustechnik (HLH) 41, VDI-Verlag, Düsseldorf, Germany, 1990, Nr. 7.
[4] M. Behne, Temperatur-, Luftgeschwindigkeits- und Konzentrationsverteilungen in Räumen mit Deckenkühlung, Dissertation, TU-Berlin, 1995, Germany, Verlag für Wissenschaft und Forschung, Berlin, Germany, ISBN 3-930324-36-9.
[5] M. Behne, Is there a risk of draft in rooms with cooled ceilings?

Measurement of velocities and turbulences, ASHRAE Transactions 2 (1995) 100.

[6] H. Brohus, P. Nielsen, Contaminant Distribution Around Persons in Rooms Ventilated by Displacement Ventilation, Roomvent, 1994, Krakow, Poland (published in the proceedings).

[7] M.F. Brunk, Cooling ceilings—an opportunity to reduce energy costs by way of radiant cooling, ASHRAE Transactions 2 (1993) 100.

[8] CEN/TC 156; WG 6, Ventilation for Buildings: Design Criteria for the Indoor Environment Draft, 1993.

[9] P.-O. Danielson, Convective Flow and Temperature in Rooms with Displacement System Roomvent, Stockholm, 1987 (published in the proceedings).

[10] DIN 1946, Teil 2: Raumlufttechnik; Gesundheitstechnische Anforderungen, Beuth-Verlag, Berlin, Germany, 1994.

[11] K. Fitzner, Quell-Lüftung für Büros, Versammlungsräume und Industriehallen, TGA-Congress, Berlin, 1988 (published in the proceedings).

[12] R.B Holmberg, L. Eliasson, K. Folkesson, O. Strindehag, Inhalation-Zone Air Quality Provided by Displacement Ventilation, Roomvent, Oslo, 1990 (published in the proceedings).

[13] P. Kofoed, Thermal Plumes in Ventilated Rooms, PhD Thesis, Instituttet for Bygningsteknik, Aalbrg, Denmark, 1991.

[14] R. Külpmann, Untersuchungen zum Raumklimatisierungskonzept Deckenkühlung in Verbindung mit aufwärtsgerichteter Luftführung, Dissertation, TU-Berlin, Germany, 1991.

[15] H. Krühne, Experimentelle und Theoretische Untersuchungen zur Quelluftströmung, Dissertation, TU-Berlin, Germany, 1995.

[16] M. Mattson, M. Sandberg, Displacement Ventilation—Influence of Physical Activity, Roomvent, Krakow, Poland, 1994 (published in the proceedings).

[17] E. Mayer (Ed.), Menschengerechte Raumklimatisierung durch Quellüftung und Flächenkühlung, Bauforschung für die Praxis, Band 13, IRB Verlag, 1995.

[18] F. Nüßle, Kühldecken: Sinnvoll auch ohne Raumlufttechnik?, Clima Commerce International (CCI), Promotor-Verlag, Karlsruhe, Germany, 1993, Nr. 12.

[19] B. Olesen, M. Schøler, P.-O. Fanger, Discomfort Caused by Vertical Air Temperature Differences, Indoor Climate, Kopenhagen 1978.

[20] W. Radkte, Klares Votum für Kühldecke, Clima Commerce International (CCI), Promotor-Verlag, Karlsruhe, Germany, 1995, Nr. 9.

[21] H. Schiller, Kombination von Kühldecken und Lüftung aus wärmephysiologischer und wirtschaftlicher Sicht—unter Zugrundelegung ganzjähriger Betrachtung Kühldecken—Erfahrungen und Entwicklungstendenzen, FGK-Informationsschrift, 1994.

[22] E. Skåret, Displacement Ventilation, Roomvent, Stockholm, 1987 (published in the proceedings).

[23] M. Stahl, Anstieg um 158 Prozent—Vorsicht mit den Zahlen, Clima Commerce International (CCI), Promotor-Verlag, Karlsruhe, Germany, 1994, Nr. 1.