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DISPLACEMENT VENTILATION AND COOLED CEILINGS

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ABSTRACT

The performance and effectiveness of any ventilation and cooling strategy depends largely on the method of air distribution and heat removal system. The consequences of poor air distribution and cooling systems are draughts, air stagnation, large temperature gradients and radiation asymmetry. These factors are the chief cause of the occupants' dissatisfaction with their thermal environment, and are major contributors to the so-called 'sick building syndrome'.

Cooled ceilings combined with displacement ventilation, sometimes known as 'comfort cooling', has gained popularity in recent years. In the UK, the traditional cooling strategies such as fan coil and vav is now being challenged with static cooling systems in majority of the new and refurbishment projects. The increasing trend towards the use of these systems has led to a number of research programmes to study the air movement, thermal environment and condensation risk of these systems (Alamdari, et al 1993, 1996; Martin et al 1997, Butler 1997).

The room air distribution and thermal environment of the combined displacement ventilation and cooled ceiling systems are presented in this paper.

KEYWORDS

Displacement ventilation, Cooled ceiling, CFD, Air flow pattern, Comfort.

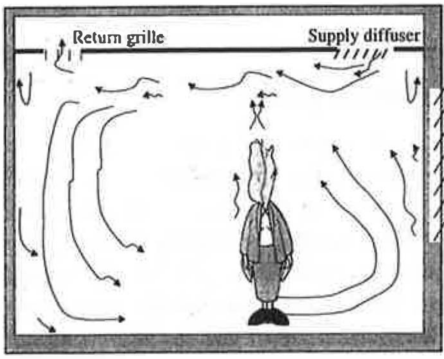
INTRODUCTION

Concerns about energy and environmental performance of buildings continue to be important design issues for the building design community. Critical attitudes towards conventional mechanical ventilation and air conditioning systems and occupants' dissatisfaction with their thermal environment have produced demands for more energy conscious ventilation and cooling system designs and higher levels of thermal comfort and indoor air quality.

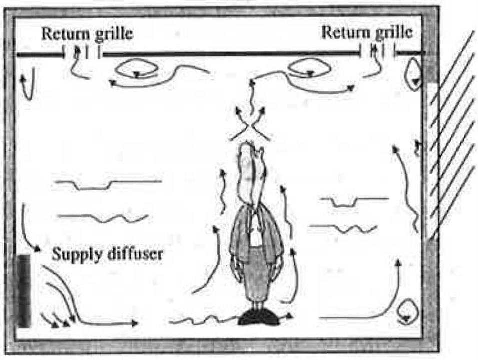
DISPLACEMENT VENTILATION

Displacement ventilation is a method that provides conditioned air to indoor environment with the view to improve air quality whilst reducing energy usage.

In the UK, buildings were ventilated traditionally by diluting the contaminated indoor air with 'fresh' incoming air and removing the polluted air from a suitable location. In recent years, however, the application of displacement ventilation systems in office applications has gained popularity. These systems have been viewed as a more efficient ventilation method, particularly, in providing a higher order of indoor air quality. In these systems, low velocity air is supplied from a low-level supply device directly into the occupied zone at a temperature slightly cooler (usually around 19°C) than the design room air temperature, while warm air is extracted at high-level. The driving force in displacement ventilation is therefore the internal heat sources. The free convection from these heat sources creates a vertical air movement in the room. The momentum flux from the supply air terminal is very low with no significant importance to the general room air movement (Neilsen 1993; Jackman 1990; Munt 1996). Figure 1 compares the airflow characteristics of displacement and mixing ventilation strategies.



(a) Mixing-flow Ventilation



(b) Displacement-flow Ventilation

Figure 1 Room airflow characteristics

In rooms with displacement ventilation there is always a vertical temperature gradient due to upward air movement and flow of heat to the ceiling region. The vertical temperature gradient divides the room into two zones separated by the so-called 'stratified boundary' (Figure 2), a lower zone with 'clean' unidirectional flow and an upper recirculation polluted zone. The stratification boundary, which needs to be kept above the occupied zone, occurs where the flow rate of the plumes (ie net upward airflow rate) is equal to the incoming airflow rate. Vertical air movement occurs in the plumes, close to the supply air terminals and along the external wall/window. In summer conditions, the wall/window temperature below the stratification layer is higher than the room air temperature, and air therefore flows upward (see Figures 1-b and 2). Above the stratification layer the boundary layer may be cooler than the mixed warm air which can result in a downward airflow. Outside the plume and away from the walls there are stratified flow regions with layers of lighter and warmer air floating horizontally (Etheridge et al 1996).

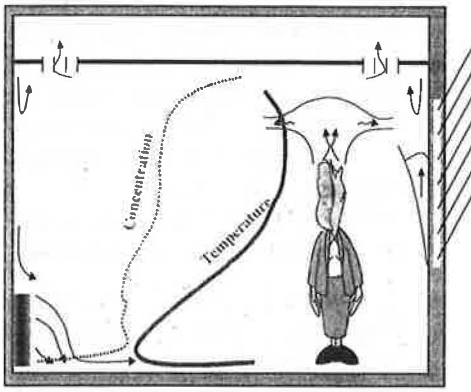


Figure 2 Temperature and concentration

Laboratory and site studies have shown that displacement ventilation is superior to mixing ventilation strategies in providing better indoor air quality, while maintaining acceptable thermal comfort requirements (Wyon 1990; Alamdari et al 1993, 1996; Breum et al 1989, 1992). Consistent studies of the occupants' perceptions of the thermal environment and air quality of these systems provided further confidence of the superiority of displacement to mixing ventilation systems (Ørchede et al 1996). In addition, with displacement ventilation it is possible to remove exhaust air from the room where the temperature is several degrees above the temperature in the occupied zone. This allows an efficient use of energy, because the supply air temperature can be higher than that in the case of mixing ventilation. However, high vertical temperature gradient may impose some risk of cold discomfort for the legs and feet, and heat discomfort at the head (Wyon 1990). Therefore, to provide good indoor air quality and thermal comfort it is necessary to use a higher airflow rate (rather than cooler air) to keep the stratification boundary above the occupied zone and to avoid low temperatures at low level.

These requirements impose limitations on the cooling capability of displacement ventilation in office applications. Sandberg et al (1989) has recommended a maximum load of 25 W/m^2, although other researchers suggest higher limits as high as 40 W/m^2 (Kegel et al 1989). The author, however, has found 25 W/m^2 is more likely to be the maximum limit for thermal comfort criteria.

COOLED CEILINGS

One way of increasing the cooling capacity of displacement ventilation and reducing the vertical temperature gradient is to remove surplus heat via cooled ceiling systems. These systems use chilled or cooled water as the cooling medium. There are many different types of cooled ceiling devices, but essentially they fall into three main categories (Figure 3): radiant cooled panels, passive chilled beams and active chilled beams. With active chilled beams ventilation is an integrated part of the beams, however, with passive chilled beams and panels ventilation has to be introduced separately, either by mixed-flow or displacement-flow ventilation methods.

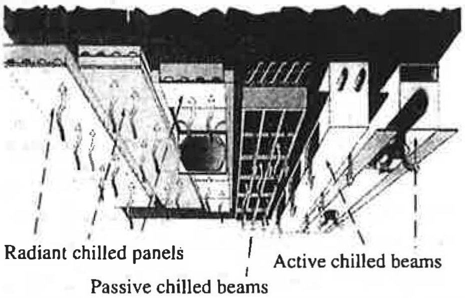


Figure 3 Cooled ceiling devices

The main difference between passive chilled beams and cooled panels is that in a cooled panel the cooling capacity is distributed across the ceiling using serpentine pipework, whereas with beams the cooling is concentrated in finned-coils similar to conventional heat exchangers. From an aesthetic viewpoint the cooled panel is probably better as it may be integrated more easily into the false ceiling. The fundamental difference between radiant cooled ceilings and the more conventional cooling (vav and fan coil) systems, is that cooled ceilings directly cool air, objects and people rather than just the air.

Cooling performance tests of a number of cooled ceiling products has shown that, for cooling loads up to about 60 W/m^2 cooled panels may be used (Alamdari, *commercial-in-confidence*). However, to provide 60 W/m^2 cooling about 75-85% of the ceiling area may need to be covered by cooled panels. For loads higher than 60 W/m^2, the application of chilled beams, which have a higher cooling capacity becomes essential.

When cooled ceilings and beams are used in combination, it has been found that the ceiling layout is an important factor in the resulting room air movement and thermal field (Alamdari, *commercial-in-confidence*). Figure 4 shows a successful combination of chilled beams and cooled panels installed in an office with external wall/window in which a controlled chilled beam is used in the perimeter zone to neutralise solar heat gains.

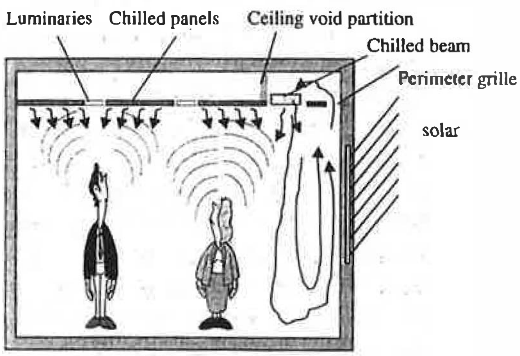


Figure 4 Cooled ceilings and beams

In principle, combining cooled ceilings with displacement ventilation has the potential to remove convective heat gains generated by people and equipment with displacement ventilation, while cooled ceilings provide radiant cooling and reduce the overall temperature gradient by cooling air in the upper part of the room. However, in practice this is largely dependent on the strength of the downward cold convection from the cooled ceilings. For a combined system to be successful, it is necessary for the downward convection from cooled ceilings to cause little or no disturbance to the upward displacement airflow.

Recent research projects funded by the UK government in partnership with the industry studied the performance of displacement ventilation with and without cooled ceiling devices, and the effectiveness of condensation control strategies (Alamdari et al 1996; Martin et al 1997; Butler 1997). In the present contribution, a summary of the work related to the room air movement and thermal energy of these systems is outlined.

METHODS

To assess the performance of a ventilation, heating and cooling system, it is necessary to obtain values of air velocities, temperatures and concentrations of pollutants within the space. This can be achieved through site measurements, laboratory-based physical modelling, computer simulations or a combination of these techniques. These data can then be used to analyse environmental thermal comfort, based on the perception of satisfaction or dissatisfaction that a person may experience within the thermal environment (ISO Standard 7730, 1984).

In the present contribution, a 'field' flow model, FLOVENT, based on computational fluid dynamics (CFD) was used to simulate comparative environmental performance of displacement ventilation with and without cooled ceilings and beams.

The numerical studies were supported by laboratory-based physical modelling which was used to validate and 'tune' the CFD models for accuracy of results.

CFD MODELLING - BACKGROUND

Building airflow and associated processes such as temperature and pressure distribution, and contaminant concentration are governed by the principles of conservation of mass, momentum, thermal energy and chemical species. These conservation laws may each be expressed in terms of 'elliptic' partial differential equations, the solution of which provides the basis for a CFD model. CFD models solve, numerically, the governing conservation equations in order to generate field values for the velocity components, the temperature and concentration of chemical species.

A common governing equation form, represents time-averaged equations for the turbulent flow, thermal energy and chemical species (Patankar 1980).

$$\partial(\rho\phi)/\partial t + \partial(\rho u_j\phi)/\partial x_j = \partial\{\Gamma_\phi(\partial\phi/\partial x_j)\}/\partial x_j + S_\phi$$

Where u_j (u_1,u_2,u_3) are the time-averaged (mean) velocity components in co-ordinate direction $x_j(x_1,x_2,x_3)$, ϕ are any of the dependent variables, Γ_ϕ are the effective diffusion coefficients, S_ϕ are the sources or sinks, ρ is the air density, and t is time. Mathematical expressions for the diffusion coefficients and source terms for each variable can be found elsewhere (Alamdari et al, 1986).

In the model employed here, the standard two-equation k-ε turbulence model was used to characterise the local state of turbulence turbulent flows in terms of the turbulence kinetic energy, k, and its dissipation rate, ε (Launder et al 1974).

To bridge the steep dependent variable gradients close to the solid surface, the code employed here applies the standard 'wallfunction' (Launder et al, 1974). These are simply based on the bilogarithmic behaviour of the mean velocity and temperature near solid surfaces.

The solution procedure was based on SIMPLE (semi-implicit method for pressure linked equations) calculation procedure (Patankar 1980). This involves an initial guess of the field values and then an iterative solution and correction procedure. The iterative solutions continue until the imbalance or error in the equations is sufficiently small to be considered negligible.

VERIFICATION OF THE MODEL

The measurements of air speeds and temperatures within a laboratory-based office model were used to verify the predicted data obtained by the computer model.

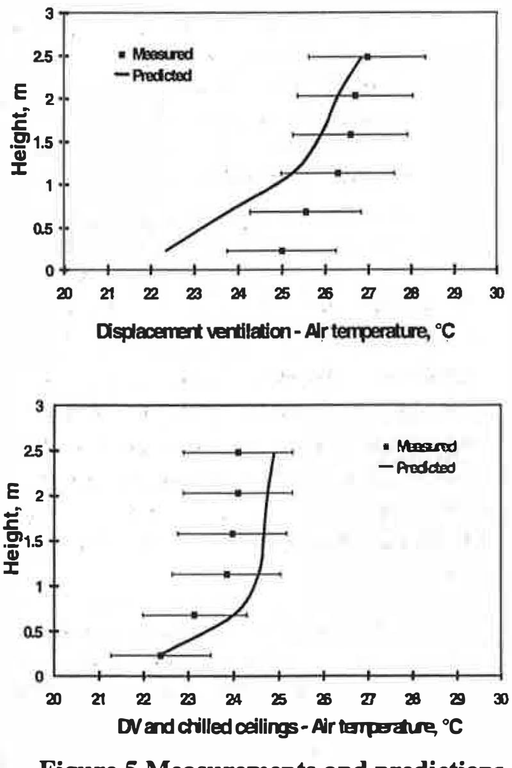


Figure 5 Measurements and predictions

Figure 5 shows the predicted and measured plane-averaged values of air temperature for displacement ventilation with and without cooled ceilings.

The differences between the predicted and measured values were thought to be in part due to the modelling assumptions, and in part due to experimental errors. These are outlined in detail elsewhere (Alamdari et al 1996). Nevertheless, acceptable confidence was achieved for the application of the model in simulating room air movement and temperature distribution with displacement ventilation and cooled ceiling devices. The computer model was subsequently used to study the comparative indoor climate and room air movement for displacement ventilation with and without cooled ceiling devices.

RESULTS

The geometry of the model considered is given in Figure 6, representing a mid-floor space with an external wall (45% glazing).

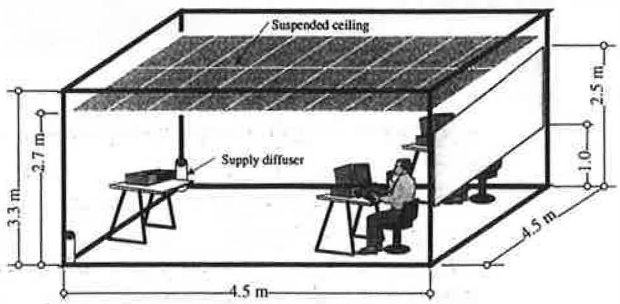
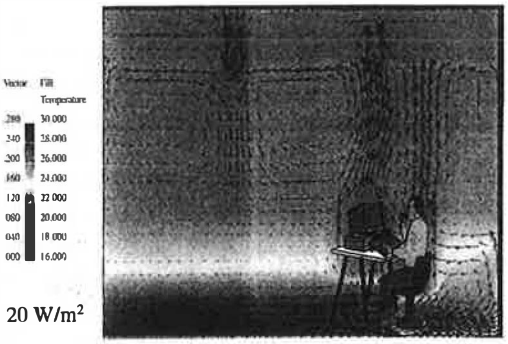


Figure 6 Space considered

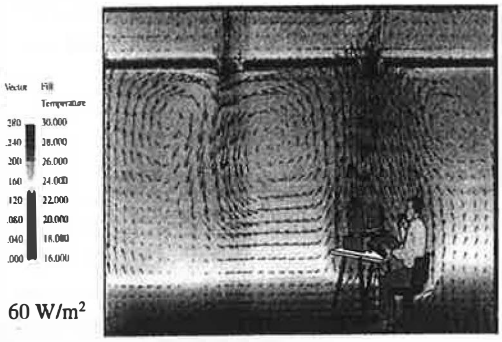
Summer simulations were carried out for a typical outside air temperature of 25°C. The thermal transmittance of the external wall and window were 0.5 and 3 W/m^2K respectively. The supply airflow rate was estimated to be 3.5 ac/h, based on thermal comfort design strategy (Jackman, 1990), by assuming a vertical temperature gradient of 1.5 K/m.

Internal loads of 60 and 20 W/m^2 were considered for displacement ventilation with and without cooled ceilings, respectively.

Figure 7(a) shows the predicted velocity vectors and air temperatures within the office with displacement ventilation.



(a) Displacement ventilation



(b) DV with chilled ceilings

Figure 7 Airflow and temperature Predictions

The predictions indicated that displacement airflow patterns were established when the thermal loads were matched to the cooling capacity of the displacement ventilation system. The warm mixed airflow region extends from the top of the seated people to the ceiling. Although the depth of this mixed-flow region can be reduced by adopting a higher ventilation rate, based on air quality-based calculation methodology (Jackman, 1990), this will increase the airflow rate to almost double the one considered here. However, to achieve the same comfort level higher supply air temperatures will then be required (≈ 21-22°C). It is important to note that the quality of the air, which would be breathed by an occupant, may be different from that at breathing height in the room, as the convection current created by the person draws it from below.

When displacement ventilation with cooled ceilings were considered the predicted airflow patterns indicated more downward convection than in the pure displacement ventilation case (see Figure 7b). However, upward convection was dominant in the vicinity of the occupants. The flow field resulting from the chilled beam cases gave a more mixed condition within the occupancy zone than in the cooled panel. With chilled beams the downward convection was increased to the extent that the resulting flow field was virtually identical to a conventional mixed airflow system case (Alamdari et al, 1996).

The plane-averaged vertical variation of environmental (ie air temperature and air speed) and thermal comfort parameters (predicted percentage dissatisfaction and predicted mean vote) are shown in Figure 8. In the case of displacement ventilation, with and without cooled panels, the vertical temperature gradients in the occupancy zone are about 1.2-1.7 K/m, whilst with the chilled beams they are about 0.6 K/m.

The predicted mean vote and the percentage of occupants' dissatisfaction were well within the recommended ranges specified by ISO 7730 (1984). Indeed, in all of the cases considered, thermal comfort was of a very high order, ie better than or equal to 90% occupant satisfaction (see Figure 9).

CONCLUSIONS

Predictions of air flow and thermal energy have shown that displacement airflow patterns will be established when the thermal loads are matched to the cooling capacity of the displacement ventilation system. However, for thermal loads higher than 25 W/m^2 there would be a risk of occupant thermal discomfort at low levels in the room.

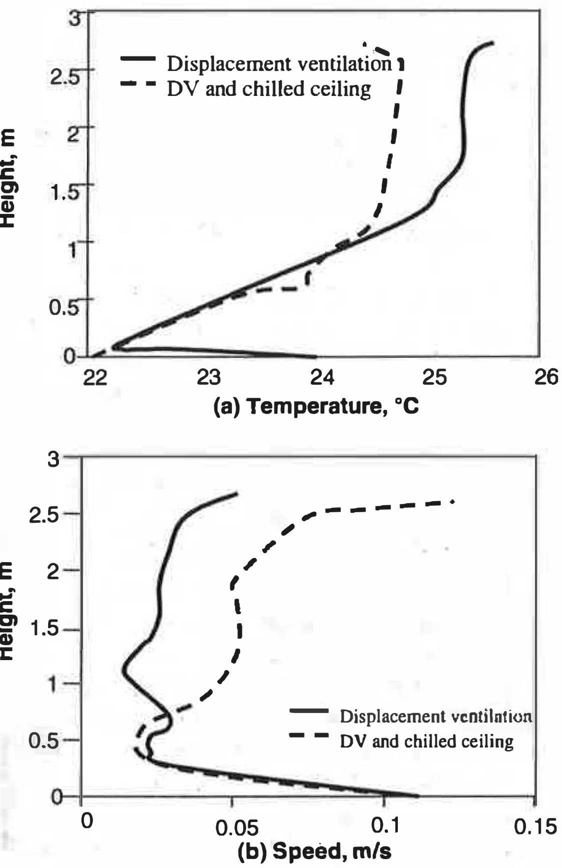


Figure 8 Temperature and speed profiles

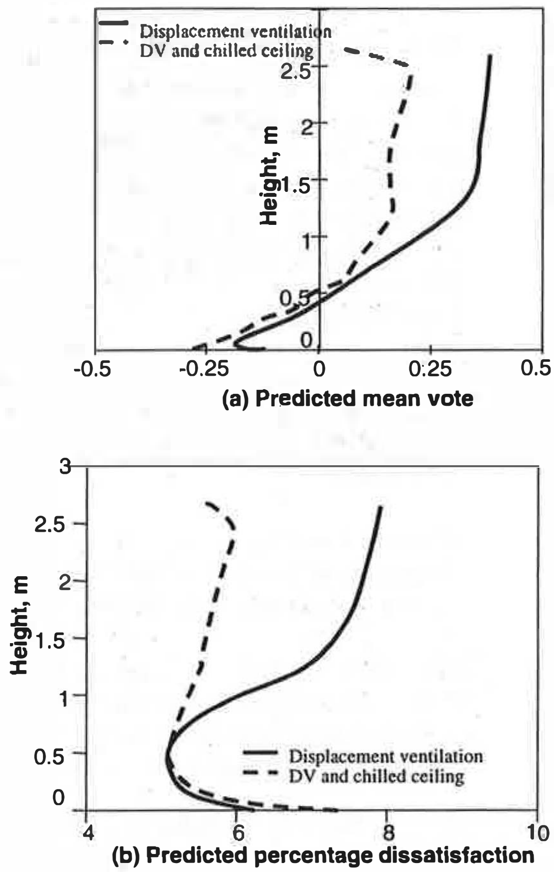


Figure 9 Thermal comfort parameters

Additional cooling can be provided by cooled ceiling (panels) to offset higher thermal gains. However, the addition of cooled panels affects the air distribution characteristics of displacement ventilation systems. Cooled panels change the air temperature near the ceiling, which creates downward convection, and hence increases the depth of the mixed warm and contaminated upper region. Furthermore, radiation heat transfer between the cooled panels and walls reduces the room surface temperatures below the room air temperature, which causes downwards convection near the wall. This may therefore transport pollutants from the mixing region into the supply air and occupation area.

The numerical predictions and laboratory testing confirm that very high environmental thermal comfort conditions are achieved with displacement ventilation combined with cooled panels.

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