TECHNICAL NOTE TN 2/96

DISPLACEMENT VENTILATION AND CHILLED CEILINGS

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BSRIA

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PREFACE

In recent years, the application of displacement ventilation systems and chilled ceiling devices in the office environment has gained popularity in the UK and is increasingly replacing the traditional mixed-flow ventilation and cooling strategies. Although displacement ventilation and chilled ceiling systems individually have been in use on the Continent for many years, their combined performance and effectiveness has been in doubt.

A programme of group-sponsored research work has been carried out at BSRIA in 1994-95 to address issues related to room air movement, thermal energy and environmental thermal comfort of displacement ventilation systems with and without chilled ceiling devices under various internal heat loads. The summary of the work carried out is outlined here.

EXECUTIVE SUMMARY

There is currently an upsurge of interest in the application of buoyancy-driven displacement ventilation systems and chilled ceiling devices in the office environment within the UK's heating, ventilation and air conditioning industry.

Buoyancy-driven displacement-flow ventilation is a method that provides conditioned air to indoor environments with the aim of improving the indoor air quality whilst reducing energy usage. Although these systems have some cooling capability, their prime function is to provide the fresh air requirements of occupants. The additional cooling requirement, however, may be provided by the adoption of appropriate 'static' cooling techniques whilst preserving the displacement flow concept. It has been asserted that this static cooling may be achieved by chilled ceiling devices such as chilled beams and/or panels. However, doubts have been expressed as to whether it is possible to preserve the upward displacement flows in the presence of the downward convective flows generated by these devices.

As part of a series of research projects in the field of displacement ventilation, the BSRIA has undertaken a programme of work to investigate whether it is feasible to combine displacement ventilation systems and chilled ceiling devices while still maintaining adequate levels of thermal comfort and a predominant upward air movement. These studies were achieved by using both physical measurements and numerical modelling based on computational fluid dynamics (CFD) techniques.

When displacement ventilation without cooled ceilings was considered, the airflow patterns were chiefly upward when the internal thermal loads were equivalent to the cooling capacity of the displacement ventilation system. On condition that the supply air temperature and air velocity were maintained within the recommended values, a high order of thermal comfort and air quality were predicted. The addition of chilled beam devices to offset higher internal thermal gains progressively eroded the predominant upward air flow region as thermal loads were increased. Indeed when the cooling load of the chilled ceiling devices was about three times that of the displacement ventilation system, the flow field was virtually similar to a conventional mixed airflow system, except in the vicinity of heat sources where upward convective plumes entrain air from the displacement cool air layer at floor level. The simulation of displacement ventilation with chilled panels, however, showed that the radiant cold panels slightly increased the depth of the mixed warm and contaminated upper region, but it did not affect the displacement airflow characteristics of the lower part of the room.

The environmental thermal comfort conditions, however, were of a very high order in all cases considered.

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1. INTRODUCTION

There is currently an upsurge of interest in the application of buoyancy-driven displacement ventilation systems and chilled ceiling devices in the office environment within the UK heating, ventilation and air conditioning (HVAC) industry. Although these systems individually have been in use on the Continent for many years, their combined performance and effectiveness have been in doubt with regard to room air movement and ventilation effectiveness.

Many independent research studies on displacement ventilation with and without chilled ceiling devices are in progress world-wide and information about the performance of these systems and their design methods appears continuously. The BSRIA has undertaken a series of research projects to evaluate the environmental performance of these systems, as part of the UK Department of the Environment 'Partners in Technology' programme. Work is currently underway to further this research and to bring together the results of the earlier studies to provide a code of practice for displacement ventilation systems with and without chilled ceiling devices.

1.1 DISPLACEMENT VENTILATION

1.1.1 Background

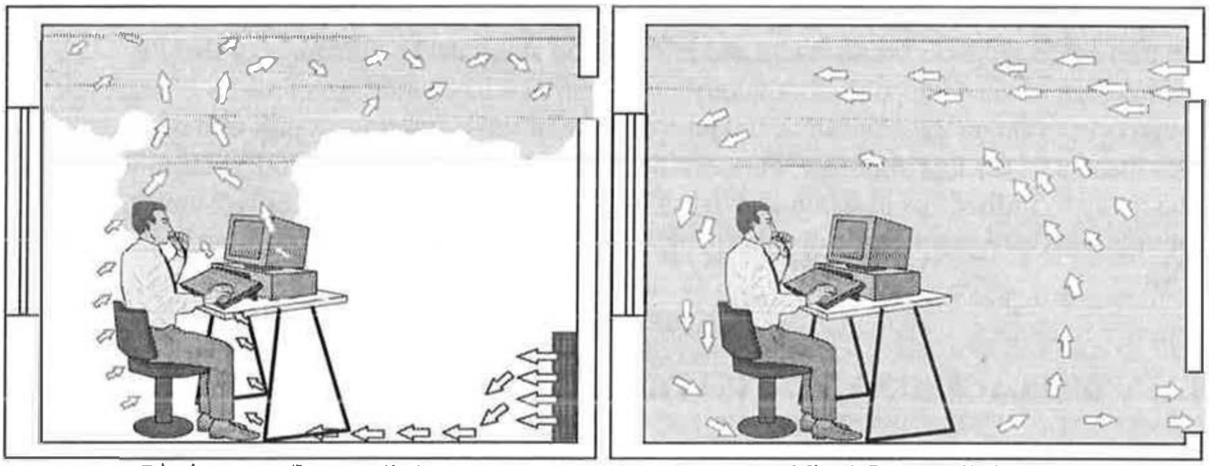
In buoyancy-driven displacement-flow ventilation systems, air is supplied at low velocity from low-level wall-mounted or floor-mounted supply air terminal devices directly into the occupied zone at a temperature slightly cooler than the design room air temperature. The air from a wall-mounted air terminal flows downward to the floor due to the gravity and it moves around the room, close to the floor, creating a cool thin layer. Natural convection from internal heat sources, such as occupants and equipment, causes upward air movement in the room. The warm, contaminated air forms a stratified region above the occupied zone which is then exhausted at high level. In displacement ventilation systems, therefore, it is natural convection or buoyancy which controls the overall room air movement and in that the momentum of the supplied air has no major influence.

Figure 1 shows the displacement ventilation concept compared with traditional mixing ventilation systems.

The airflow in displacement ventilation has both horizontal and vertical air movement characteristics. The horizontal air movement occurs in the thermally stratified layers which are formed between the upper and lower parts of the room, and have warmer and cooler air temperatures, respectively. Figure 2 shows the horizontal and vertical air movement in displacement ventilation. The vertical air movement, however, is caused by cold and warm objects in the space. Whilst warm objects such as people and small power loads create upward convection currents, cold objects such as cold windows and walls cause downward currents (see Figure 3).

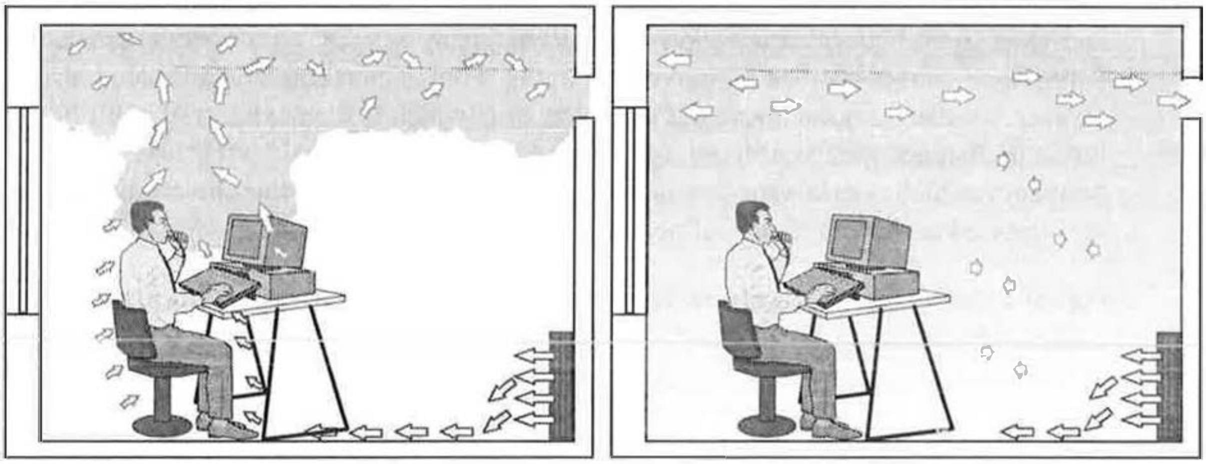
For a given ventilation rate and pollutant discharge, the air quality in the occupied zone of a room with a displacement ventilation system can be higher than that utilising a mixed-flow ventilation method. In displacement ventilation the air movement above the occupancy zone is often rather mixed and it is when this mixed region extends down with the occupied zone that the air quality becomes similar to that in a mixed-flow system. Figure 3 shows vertical contaminant distribution in a room with displacement ventilation.

Figure 1 Room air movement with displacement and mixed-flow ventilation systems



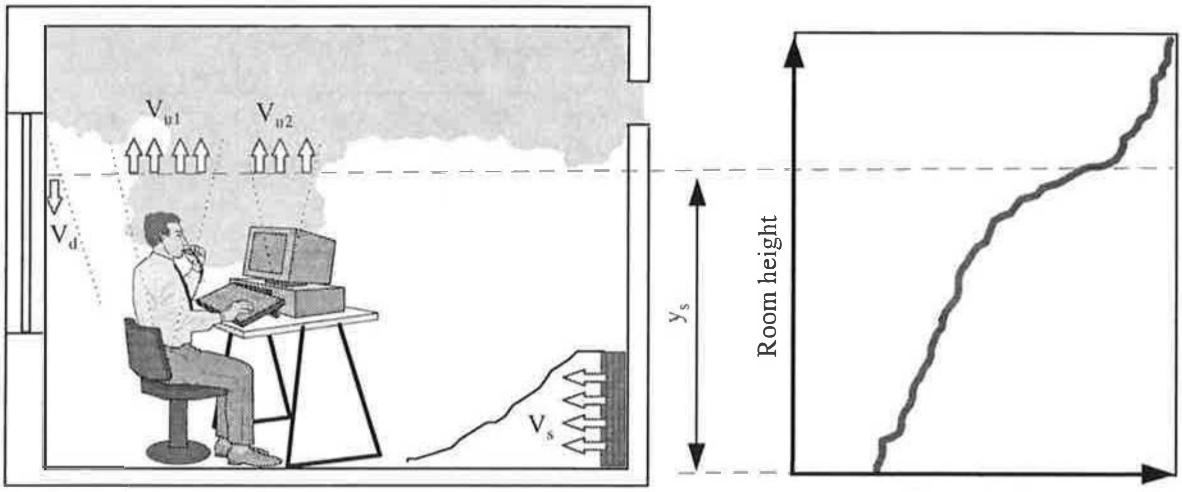
Displacement-flow ventilation Mixed-flow ventilation

Figure 2 Room air movement with displacement ventilation systems



Vertical air movement Horizontal air movement

Figure 3 Convection currents and concentration profile



1.1.2 Basic Design Strategy

The design of a displacement ventilation system may be based on either thermal comfort or indoor air quality. Jackman[2] has outlined the following basic procedures for these design strategies:

Thermal comfort-based design

In this design strategy the main comfort criteria are the thermal environmental conditions, therefore the design procedure does not differ from mixed-flow ventilation strategies. The basis of design is illustrated in Figure 4.

The starting point here is to select an acceptable temperature gradient limit. The overall temperature difference between the supply and exhaust air may then be calculated from the following empirical equation[2].

$$t_e - t_s = 1.7\, t_g\, (h_r - 0.1) \qquad (1)$$

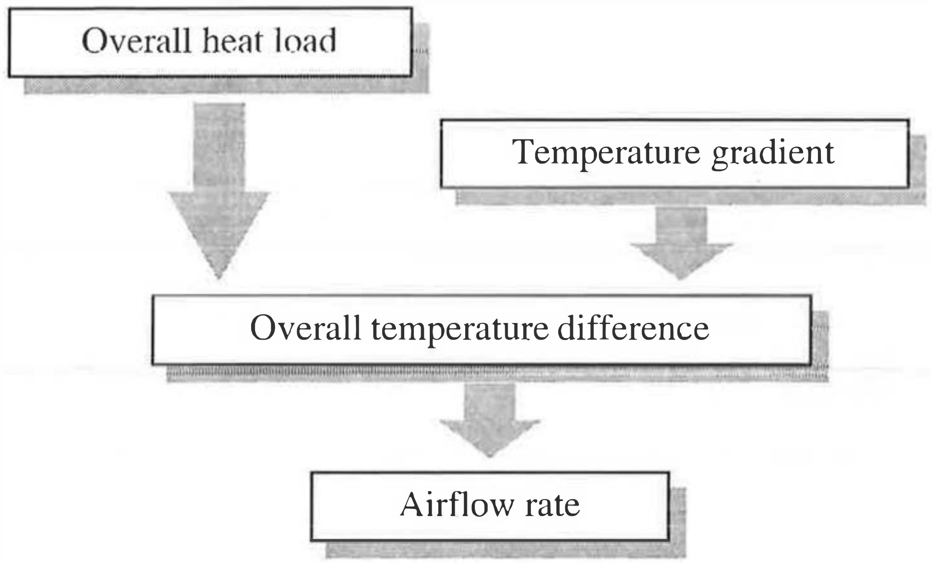
where t_e = exhaust air temperature, °C
ts = supply air temperature, °C
t_g = air temperature gradient, K/m
h_r = floor to ceiling height, m

The required air flow rate then can be calculated from the equation:

$$V = Q/[\rho\, C_p\, (t_e - t_s)] \qquad (2)$$

where V = air volume flow rate, litres/s
Q = total estimated heat load, W
ρ = air density, kg/m^3
C_p = specific heat of air, kJ/kg K

Figure 4 Thermal comfort-based design strategy



Air quality-based design

One of the main benefits of a displacement ventilation system over a mixed-flow system is its ability to achieve high indoor air quality. This is due to the separation of warm contaminated air from fresh air by vertical temperature gradient in the space. The warm contaminated air will rise and form a stratification boundary above the heat sources. The stratification boundary will occur at a height, y_s, where the resulting upward convection currents equal the supply air flow rate (ie $\sum V_u - \sum V_d = V_s$, see Figure 3).

The design of displacement ventilation based on air quality, therefore, involves the calculation of the convection currents from difference sources in the space. Although this design strategy is more complex than the thermal comfort based method, it is more likely to satisfy the indoor air quality requirements.

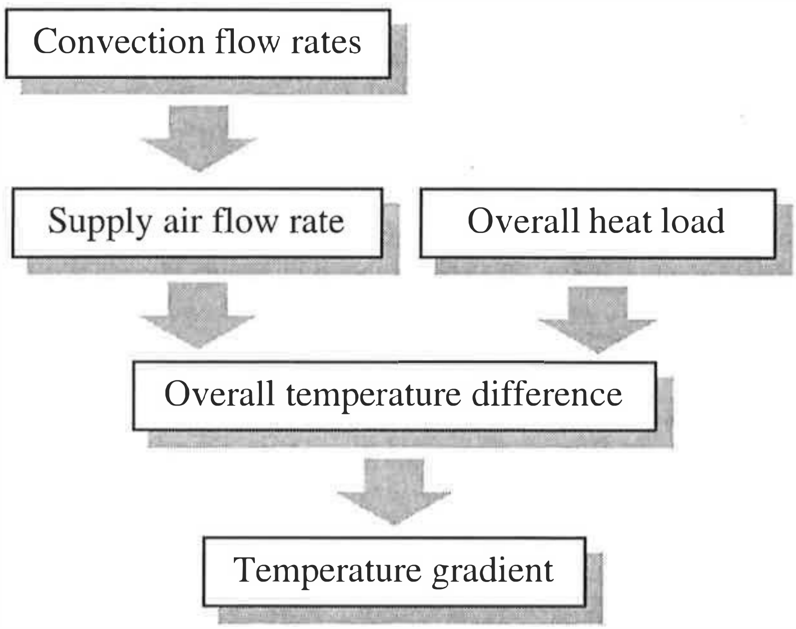
Based on experimental measurements, analytical equations have been driven for idealised sources[2, 3 ,4]. The basis of air quality-based design is illustrated in Figure 5.

1.1.3 Applications and Limitations

Displacement ventilation may be employed for many applications and building types, however there are conditions under which this system is less effective than the traditional mixed-flow ventilation strategies. For example[2]: -

i. where the supply air is warmer than the room air (except under particular circumstances where cold downdraughts exists over the supply position)
ii. where contaminants are cold and/or more dense than the surrounding air
iii. where surface temperatures of sources are low
iv. where ceiling heights are low
v. where disturbance to room air flows are unusually strong

Figure 5 Air quality based design strategy



In addition, other specially occurring factors should also be considered: -

Room layout

The supply air terminals in displacement ventilation systems are often larger than diffusers in mixed-flow ventilation , and therefore require more wall space. Further, they need to be located at low levels within the occupancy zone.

In addition, the airflow at low levels may be influenced by furniture and partitions within the room. Although conventional office furniture may only have a small influence on the air movement, obstacles placed directly on the floor will block the flow, and any opening between them may act as new supply opening because of the thermally stratified layers in the room[3].

Vertical temperature gradient

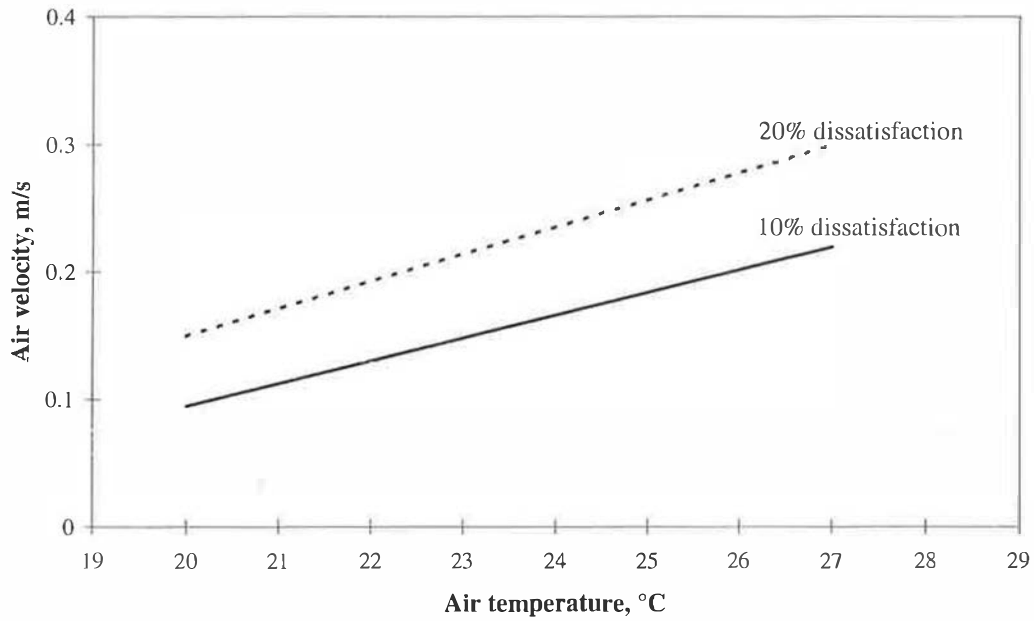
Unlike mixed-flow ventilation where there is little or no temperature gradient within the occupancy zone, with displacement ventilation a vertical temperature gradient is unavoidable in this region. Therefore there will be a temperature differential between ankle and head levels which, if excessive, may cause discomfort. ISO Standard 7730[5] recommends that the vertical air temperature gradient for sedentary occupants should be less than 3 K. This gives a gradient of about 3 K/m. Other sources[2,4] recommend a lower temperature gradient value of 2 K/m. CIBSE Guide[6] recommends that the variation of air temperature across the space should be within a 3 K limit, that is ±1.5 K about the mean room air temperature.

The vertical temperature gradient can also produce low and high surface temperatures at floor and ceiling levels, respectively. This can increase the radiant asymmetry, and hence cause discomfort. For comfort, a difference of 5 K between the thermal radiation from ceiling and floor levels of a room has been quoted[2,5] in relation to sedentary occupation in winter.

Air velocity

Whereas with mixed systems the air is normally introduced to the space at locations outside the occupied zone, with displacement ventilation systems the air is discharged directly into the occupied zone at low level, therefore there is a potential for local discomfort in these regions, particularly, in the vicinity of the air terminals. The ISO Standard[5] recommends a velocity of less than 0.15 m/s in winter and 0.25 m/s in summer for sedentary occupants. However, a study by Wyon and Sandberg[7] has derived relationships between the air velocities, temperatures and the percentage of sedentary people dissatisfied. These relationships which are graphically displayed in Figure 6, reflect human sensitivity to cooling at low levels and suggest that near-floor temperatures below 22°C are unlikely to be acceptable, if accompanying air movement exceeds about 0.15 m/s.

Figure 6 Thermal comfort air temperature and speed limits



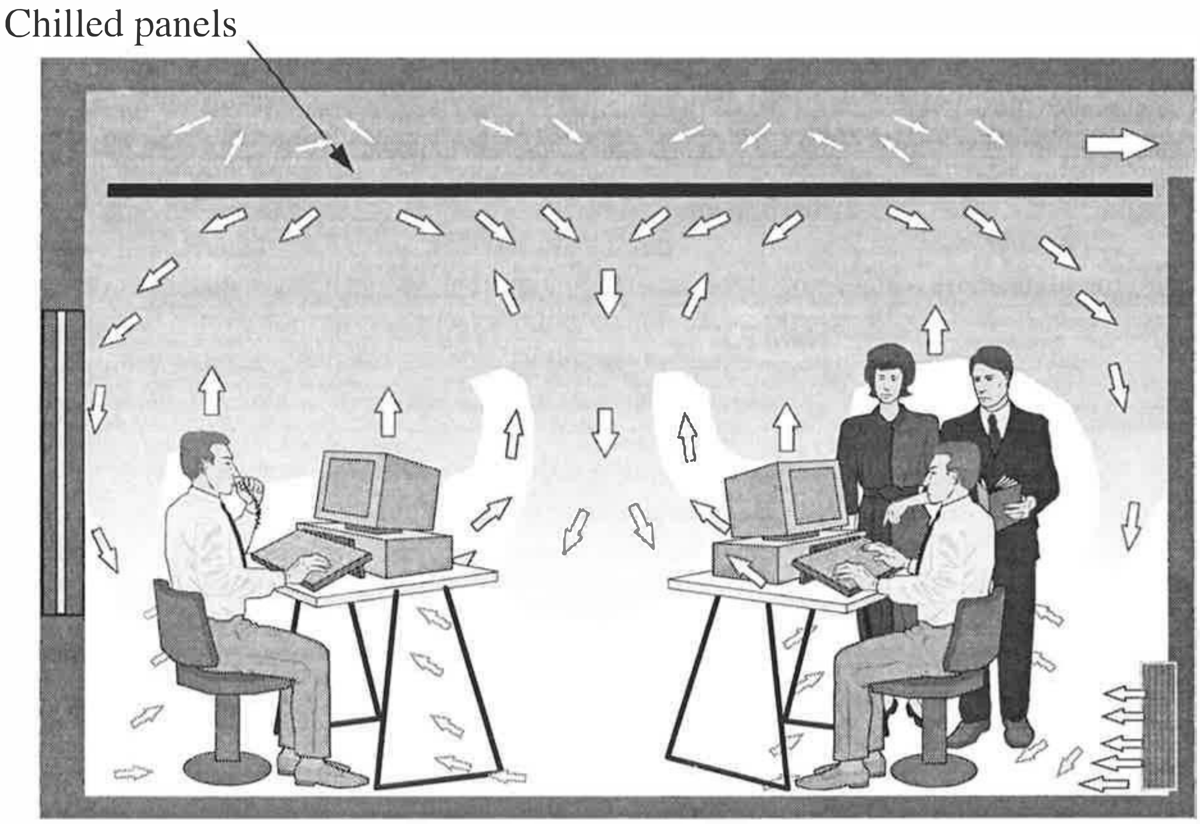
Cooling capacity

In office applications, the cooling capacity of displacement ventilation is limited due to the maximum acceptable vertical temperature gradient. Sandberg[8] has recommended a maximum load of 25 W/m^2. Although other researchers suggest higher limits of 30 W/m^2 and for certain situations even as high as 40 W/m^2 (eg Kegel[9]), the authors have found Sandberg's recommendation of 25 W/m^2 is more likely to be the maximum limit for thermal comfort criteria.

One way of increasing the cooling capacity of displacement ventilation systems to remove surplus heat is via radiant cold ceiling panels and/or convective chilled beams (Figure 7). In principle, the combination of chilled ceilings and displacement ventilation may offer both thermal comfort and air quality in spaces with high internal loads and pollutant rates; in practice however, this may counteract the displacement flow concept. This is because the chilled ceilings cool the warm air at the ceiling level which may cause cold downward convection into the occupied zone.

To investigate whether it is feasible to combine displacement ventilation systems and chilled ceiling devices while still maintaining adequate levels of thermal comfort and a predominant upward air movement, a programme of work[1] was carried out at BSRIA. This involved both physical measurements and numerical modelling based on computational fluid dynamics (CFD) techniques. The summary of the work carried out is outlined in the following sections of this document plus an analysis of the comparative environmental performance of displacement ventilation with and without chilled ceiling devices.

Figure 7 Displacement ventilation and chilled ceilings



2. ENVIRONMENTAL PERFORMANCE ANALYSIS

2.1 THE METHOD

Considerations of indoor climatic quality, dissatisfaction with the indoor thermal environment and awareness of airborne contaminant concentration have a high profile amongst designers in the application of any novel HVAC design, such as displacement ventilation and chilled ceilings.

To study whether it is feasible to combine displacement ventilation systems and chilled ceiling devices in the office environment to obtain adequate levels of thermal comfort and a predominant upward displacement flow, it is necessary to predict the air movement and related phenomena, such as thermal energy and airborne pollutant concentrations within the space. The predictions may be achieved by one or a combination of physical modelling and numerical modelling techniques. The predicted data obtained from physical and/or numerical simulations may then be used for environmental thermal comfort and/or air quality assessment.

2.1.1 Physical Modelling

Physical modelling techniques involve the actual measurements of air velocity, temperature and concentration of species within a laboratory based full size or scale model, to pilot new designs. These techniques facilitate 'field' values of the environmental parameters, such as air speed, air temperature, radiant temperature and contaminant level that are required to evaluate environmental thermal comfort and ventilation effectiveness and could include basic visualisation techniques, for example using smoke as a tracer.

Thermal gains such as lighting, equipment and occupants in the model may be simulated by using actual equipment and people. However using humans is impractical and costly, therefore, in modelling practice, some compromises may be made. For overall field air temperature and velocity assessment, occupants may be represented by black boxes which have a surface area approximately equivalent to human body and an appropriate heat output. For quantitative air flow and air quality tests around a person, however, a full breathing mannekin is essential. This gives results which are mutually comparable with real situations.

Air velocities, air temperatures, mean radiant temperatures and turbulence intensity in the test facility can be obtained by using a micro-processor based data acquisition system, such as the DANTEC multi-channel flow analyser which was used in this study. The measurement probes of this unit combine air speed and temperature measuring transducers in close proximity, effectively enabling measurements of both variables at a point location. To monitor, measure and control the temperatures of all boundary surfaces, supply and extract air, flow and return water, additional thermocouple sensors and instruments are required.

To evaluate the thermal field and air flow within the occupancy zone , air temperatures and air speeds may be sampled in columns in the test facility at a number of heights, at the prescribed measurement grid points on the floor.

2.1.2 Numerical Modelling

Air movement, temperature distribution and airborne contaminant concentrations 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 microclimate computational fluid dynamics (CFD) model.

The finite-volume form of the differential equations may be achieved by dividing the flow domain into a finite set of small cells or control volumes, and integration of the equations over each cell volume. The finite-volume equations are then solved in the CFD model by an iterative manner, to generate field values for all dependent variables, such as air velocity components, air temperature, pressure, concentration of chemical species, etc.

Particular benefit of CFD is visual representation for analysis and appreciation of otherwise invisible circumstances.

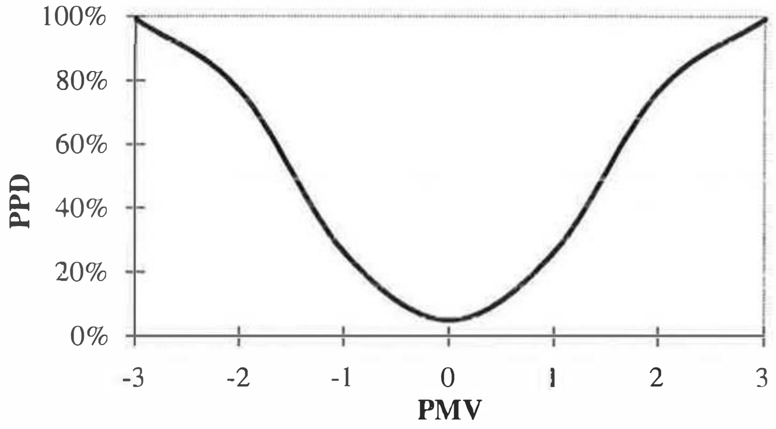
2.1.3 Thermal Comfort Analysis

Thermal comfort is defined as that condition of mind in which satisfaction is expressed with the thermal environment. Thermal comfort assessment, therefore, can be based on the perception of satisfaction or dissatisfaction that a subject experiences with the thermal environment. In this context, thermal comfort indices may be defined which can be used to evaluate how warm or cool people will feel in a given environment.

For guidance on thermal comfort, the model suggested in ISO Standard[5] may be used. This thermal comfort model is based on laboratory-based experiments, therefore, although it may require some modifications for their direct applications in real buildings, it is a useful predictive comparitor which may be applied to analyse the relative thermal comfort performance of systems in a given situation. The model uses the predicted mean vote (PMV) index, which combines the influence of air temperature, mean radiant temperature, air movement and humidity with clothing and activity level into one value on a 7-point thermal sensation scale[3,4], from hot (+3) to cold (-3). To predict the number of people likely to feel uncomfortably warm or cool, a further index, PPD (predicted percentage of dissatisfied), has been defined which gives a quantitative prediction of the number of thermally dissatisfied people. The relationship between the PMV and the PPD indices are displayed[5] in Figure 8.

Compatible to real-life situations the predictions suggest that there will always be some dissatisfaction, because of personal differences and the ability of individuals to make adjustments (to their clothing level, for example), low PPD values indicate a higher level of acceptability than the percentage value might convey. In practice, therefore, when the thermal environment is predictably acceptable to at least 80% of the occupants (ie 10% dissatisfaction on either side of the scale, $-0.5<PMV<+0.5$), the design may be considered satisfactory.

Figure 8 Thermal comfort indices

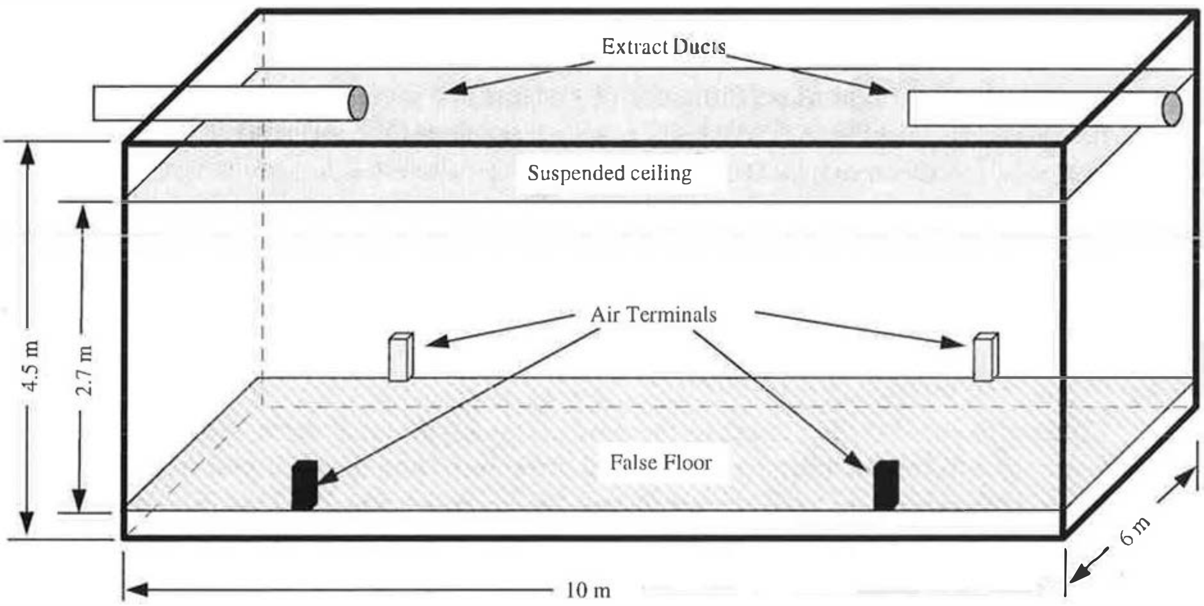


2.2 PHYSICAL MODELLING

2.2.1 The Environmental Test Chamber

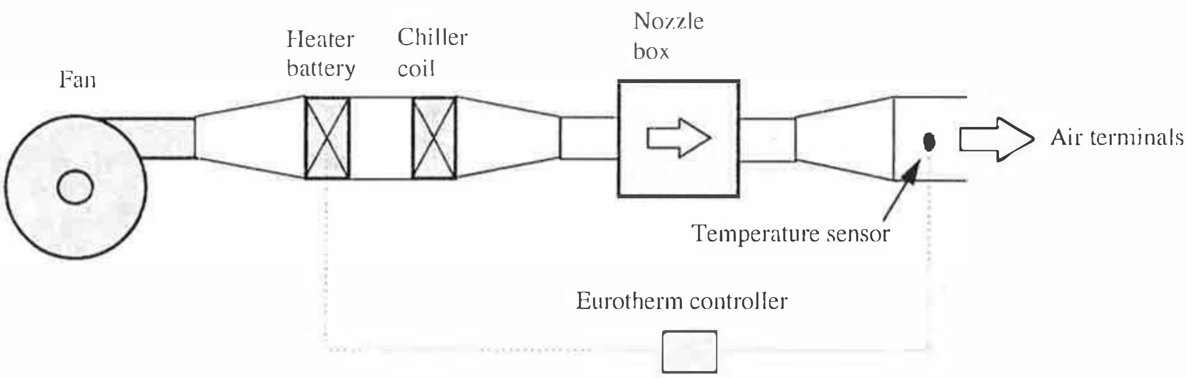
A large test chamber representing an open-plan office environment with internal dimensions of 10 m by 6 m by 2.7 m in height was used to simulate the climatic conditions with the displacement ventilation and chilled ceilings (Figure 9).

Figure 9 Schematic diagram of the test facility



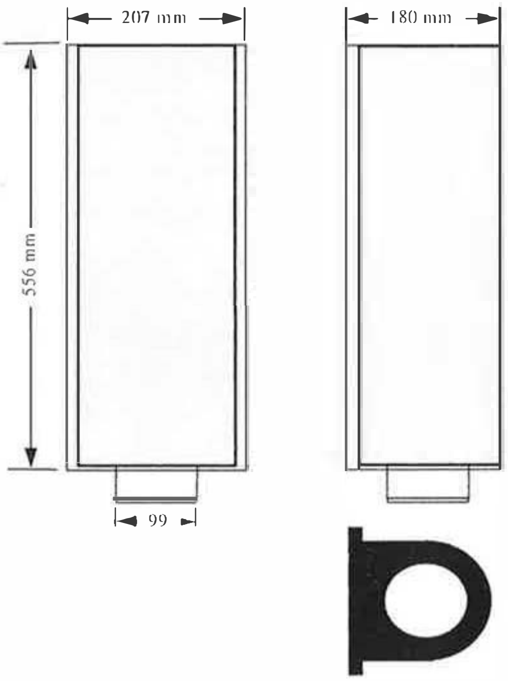
The ventilation plant comprised a centrifugal fan, chiller and expansion coil, an electric heater battery, and a feedback sensor controller (see Figure 10).

Figure 10 Air distribution plant



A damper at the fan inlet was used to regulate the air volume flow rate through the system. Conditioned air was supplied to the room via air distribution ducting under the floor void to four floor standing semicircular displacement air terminals (Figure 11), with maximum recommended flow of 40 l/s per terminal. Air was extracted through the ceiling void via two 150 mm diameter ducts which emerged at high level through the walls of the facility and were routed to the inlet of a centrifugal fan. A damper on the outlet of each fan was used to regulate the rate of air extraction in equal amounts such that the net pressure differential between the facility and the laboratory was zero when the access door was closed.

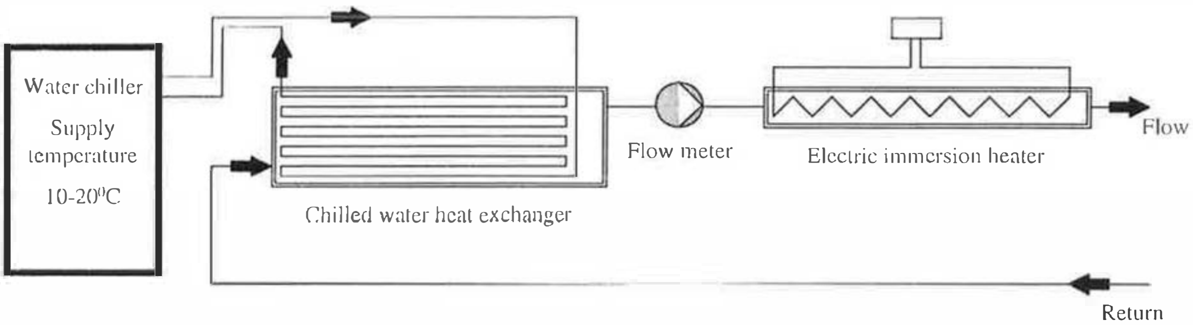
Figure 11 Schematic diagram of the displacement air terminals



2.2.2 Chilled Ceiling Systems

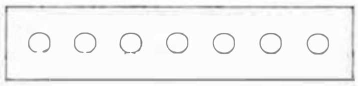
A water-to-water refrigeration plant was used to supply chilled water to the cooled ceiling devices. Primary water was pumped to a heat exchanger coupled to the secondary water circuit which supplied the cooled ceiling systems. Water in the secondary circuit passed through another small vessel containing a temperature regulated electric immersion heater before supply to the cooled ceiling devices. The electric immersion heater was controlled by a sensor in the water flow which was linked to a Eurotherm Controller. Figure 12 displays the chilled water plant system.

Figure 12 Chilled ceiling plant system



Two types of chilled beams and one type of chilled panel were considered in this programme of work. These are listed in Table 1.

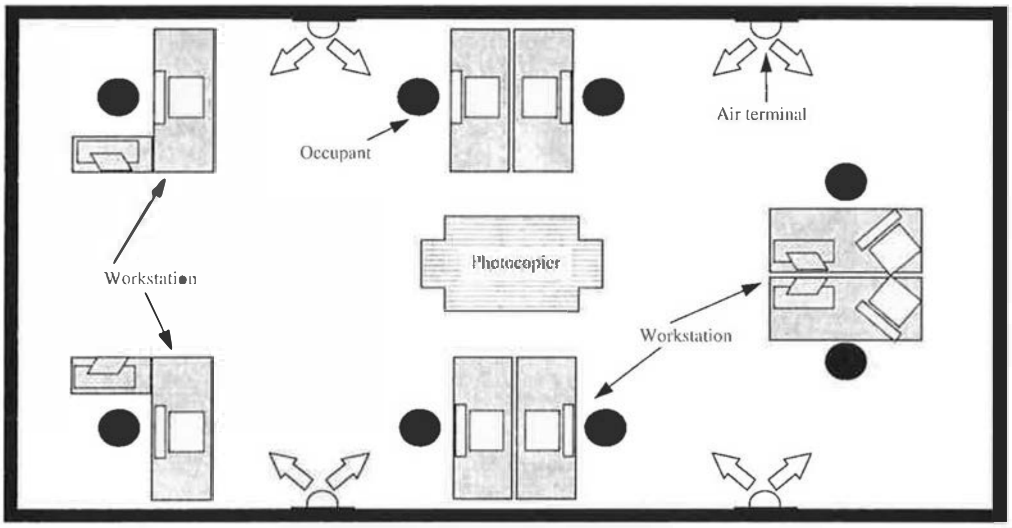
Table 1 Chilled ceiling devices considered

Device	Mode of heat transfer	Schematic
Capped Chilled Beams	80-90% convective 20-10% radiant	
Uncapped Chilled Beams	85-90% convective 15-10% radiant	
Chilled Panels	40-50% convective 60-50% radiant	

2.2.3 Room and Ceiling Layouts

Figure 13 illustrates the internal layout of the test room.

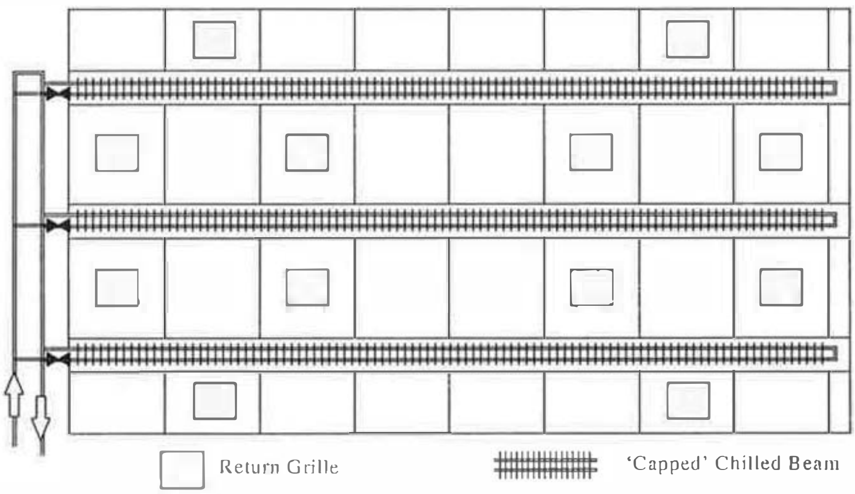
Figure 13 Room layout



For the above room layout, two different ceiling arrangements were considered:

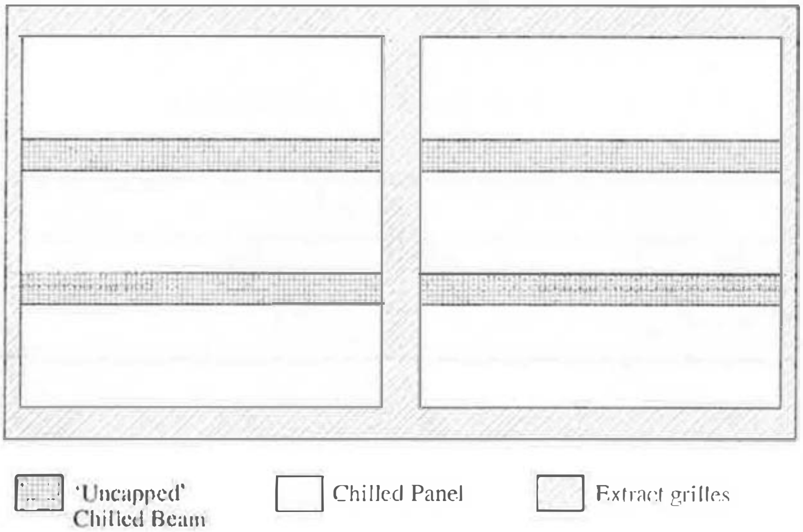
i. The first layout incorporated 'capped' chilled beams (see Table 1) in which air could not pass down from the ceiling void through the chilled beam and into the room. The beams were installed in the facility, as shown in Figure 14.

Figure 14 Ceiling layout with 'capped' chilled beams



ii. The second ceiling layout was a composite of 'uncapped' chilled beams (see Table 1) and radiant chilled panels, offering two alternative systems to be operated independently. Figure 15 shows the installation.

Figure 15 Ceiling layout with 'uncapped' chilled beams and chilled panels



2.2.4 Test Scenarios Considered

To assess the airflow and thermal environment within an office space incorporating displacement ventilation with and without chilled ceiling devices, a number of scenarios were considered in the original study[1]. Initially, the displacement ventilation flow field was characterised for an internal heat load of 20 W/m^2. Internal gains were then increased in increments of 20 W/m^2 up to a maximum load of 80 W/m^2. The maximum load of 80 W/m^2 was selected to represent the upper level of typical loads in internal zones of open-plan offices. At load settings of 40 W/m^2 and above, chilled beams or chilled panels were operated to provide the additional cooling necessary. For these scenarios, the supply water temperature was regulated and set to ensure a nominal 14°C at the entries to the beams or panels. The ventilation rate and the supply air temperature† were 3.5 ach^{-1} and 19°C respectively.

The thermal loads in the test facility were initialised according to the requirement of the test conditions by means varying the number of occupants (represented by black boxes), PCs, lights and thermal output of the photocopy machine. The typical unit heat gains from each of these sources are shown in Table 2.

Table 2 Typical thermal gains from internal sources

Category		Unit heat output, W
Lighting	fluorescent tubes (uplight)	58
Occupancy	sedentary	100
Equipment	PC	90 (approx.)
	Printer	50 (approx.)
	Photocopier	1800

† *Supply air temperature was adjusted to simulate 19°C in order to compensate for the test chamber fabric heat losses or gains.*

The information provided in this document relates to a small number of scenarios considered of in the original study[1]. These are displayed in Table 3.

Table 3 Scenarios considered

Operating condition	Internal load (W/m^2)	Cooling output (W/m^2) Ventilation	Cooling output (W/m^2) Chilled ceiling
Displacement ventilation without chilled ceilings	20	20	-
Displacement ventilation with chilled panels	60	20	40
Displacement ventilation with 'capped' chilled beams	60	20	40
Displacement ventilation with 'uncapped' chilled beams	80	20	60

To measure the thermal field and air flow within the test facility, air temperatures and air speeds were sampled in columns at heights of 0.2, 0.7, 1.1, 1.6, 2.0, and 2.5 m and at points spaced 600 mm apart on a regular square grid.

2.3 NUMERICAL MODELLING

A special-purpose computational fluid dynamics (CFD) software, FLO*VENT* ‡, was employed. This code applies the SIMPLE ('semi-implicit method for pressure linked equations') algorithm[10] to solve the finite-volume form of the governing flow equations. The popular 'two-equation' models of turbulence, the k-ε model[11], was adopted. To bridge the steep dependent variable gradients close to the solid surface, the FLOVENT code employs the standard 'wall-functions'[11].

2.3.1 Geometry and Computational Grid

The first step in the solution of the governing flow is the discretization of the solution domain into a finite number of control volumes. Discretization must conform to the boundaries in such a way as to allow accurate representation of the boundary conditions, and allow the computational control volumes to be small in the regions where the rate of change of a property with distance is high, such as at a solid surface. Domain disretization within FLO*VENT* is provided by means of a computational grid which is based on a Cartesian co-ordinate system.

Due to the symmetrical nature of the internal layout of the test chamber, the supply terminals and the chilled ceiling, only half of the facility was modelled. This allowed better utilisation of the grid distribution, an increase in the grid resolution and allowed higher concentration of cells in the regions of high thermal and velocity gradients, giving improved accuracy of the computed data.

‡ *This code was developed by BSRIA and Flomerics Limited for analysis of air movement, temperature distribution and airborne contaminant dispersion in the context of the built environment.*

2.3.2 Thermal and Flow Boundary Conditions

It is normal practice with CFD models to specify the internal surface thermal boundary conditions for the space. These may be based on the measured surface temperatures, and therefore effectively account for thermal radiation exchange within the space. However, using a single measured value of temperature on a wall may lead to errors particularly in the applications of displacement ventilation where the vertical temperature gradient in the space may affect the temperature variation along the vertical surfaces. Therefore, in this study, heat fluxes through the solid boundaries of the test chamber were accounted for based on the measured external (ie laboratory) air temperature, recommended external surface convective heat transfer coefficients[5] and the thermal conductivities of the wall materials. This practice, however, excludes the radiation heat transfer effects between the internal surfaces of the space.

In the application of chilled ceiling devices, the elimination of the radiation heat transfer may lead to errors in the predicted flow field values, therefore it is necessary to account for this. This was performed using a simple surface-to-surface radiative heat exchange model.

2.4 MEASUREMENTS AND PREDICTIONS

The measurements of air speeds and temperatures were used to verify the predicted data obtained by the computer model which was subsequently used to analyse the detailed room air movement and environmental thermal comfort.

2.4.1 Vertical Air Temperature and Speed Profiles

The predicted and measured plane-averaged air temperature and speed for the test scenarios considered (see Table 3) are displayed in Figures 17-20.

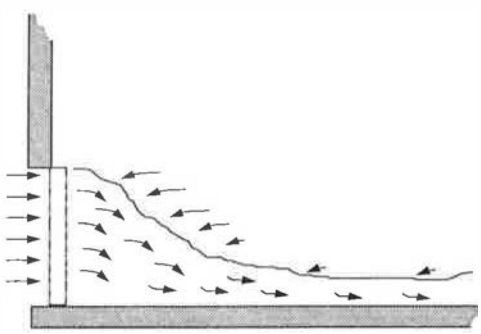
The data in these figures are calculated by averaging for each variable at each plane, and applying a weighting function based on cell size. The resolution of the computational and the measurement grid points on horizontal planes were 2730 and 170, and in vertical direction were 30 and 6, respectively.

The differences between the predicted and measured values were thought to be in part due to the modelling assumptions, and in part to experimental errors. In particular, these are:

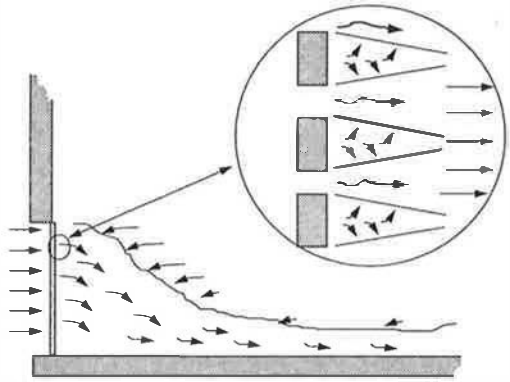
i. The computer simulations were based on the solution of the steady-state form of governing flow equations, but the physical data are related to time-dependent external environmental conditions.

ii. The reliability of predicted data depends on the theory of a turbulence model adopted, the internal surface convective heat transfer and boundary layer treatment, the numerical solution procedures and the boundary conditions employed. For example, related to the latter was the representation of the displacement air terminals, which were represented by filter mat flow resistance rather than perforated panels (see Figure 16). This geometrical approximation does not include the small-scale mixing processes at the perforations (see Figure 16-b), and entrains room air only at the boundary surface. Therefore the predicted local conditions could be different to the measured values, and the velocity and temperature along the floor could become higher and lower, respectively. Nevertheless, due to the restriction in computer power and geometrical complexity, some compromises are necessary in these situations.

Figure 16 Filter mat and perforated plate air terminals



(a) Airflow pattern from a filter mat terminal



(b) Airflow pattern from a perforated plate terminal

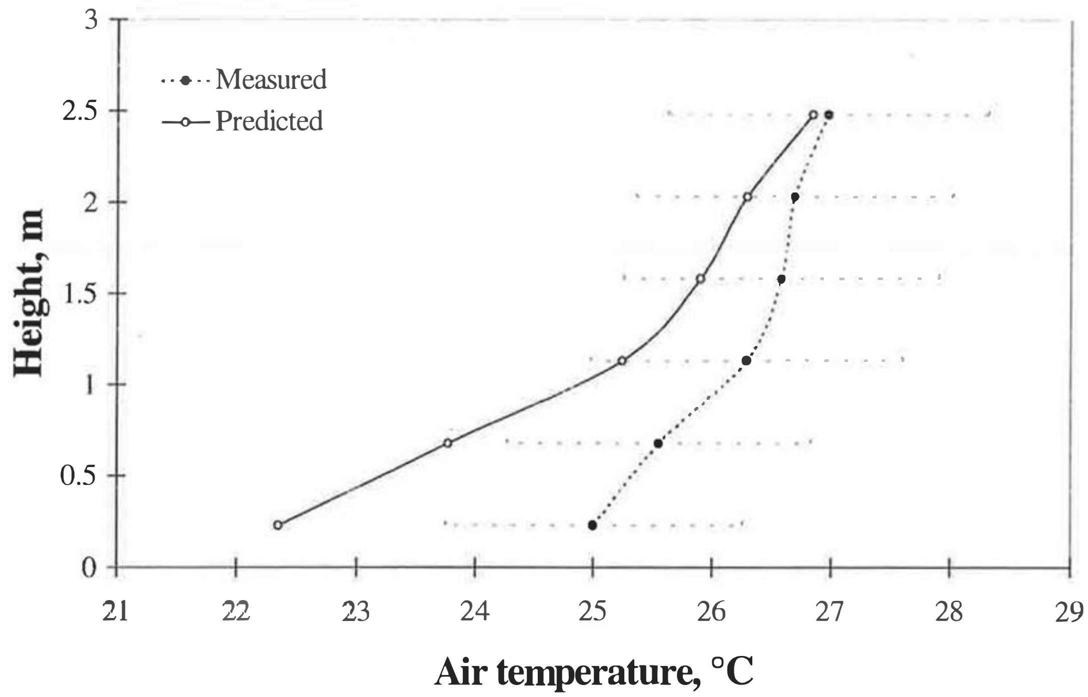
iii. In displacement ventilation the buoyancy forces and heat flux at boundaries are important and need to be included in simulations. In the numerical modelling presented here, due to unreliability of the measured external air temperature (ie laboratory air temperature), the heat fluxes at boundaries were roughly accounted for. Here, the measured external surface temperature were used as ambient air temperature. At the internal boundaries the dependent variables and heat transfer were calculated by means of empirical logarithmic wall functions[9].

iv. The influence of radiation is important for the room air movement in displacement ventilation systems. Radiation heat fluxes from the warmer upper part of the room to the cooler lower part and vice versa, increases and reduces the floor and ceiling temperatures, respectively. Although the radiation heat transfers between ceiling and floor were included using a simple surface to surface radiation heat transfer calculation between the floor and the ceiling, the effects of walls and internal heat sources were ignored.

v. All measured values were subject to experimental errors, ie precision in reading and calibration error band.

Nevertheless, acceptable agreement was achieved for the averaged air temperatures and air speed at each of the measured heights (Figures 17-20).

Figure 17 Variation of room air temperature and air speed with height - displacement ventilation (20 W/m^2)



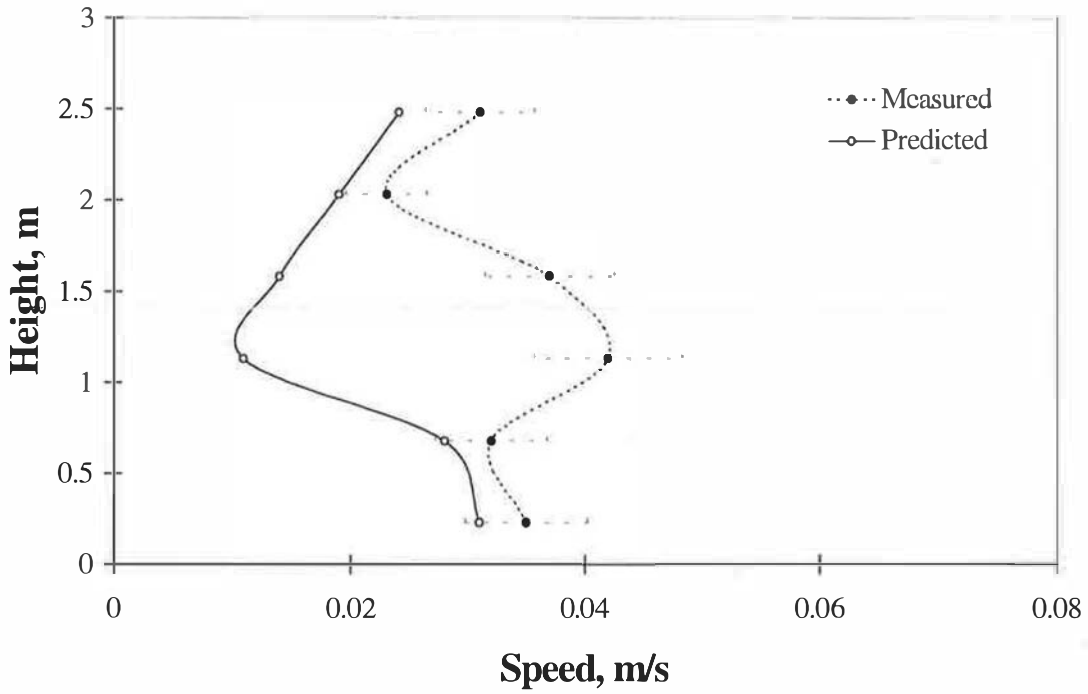
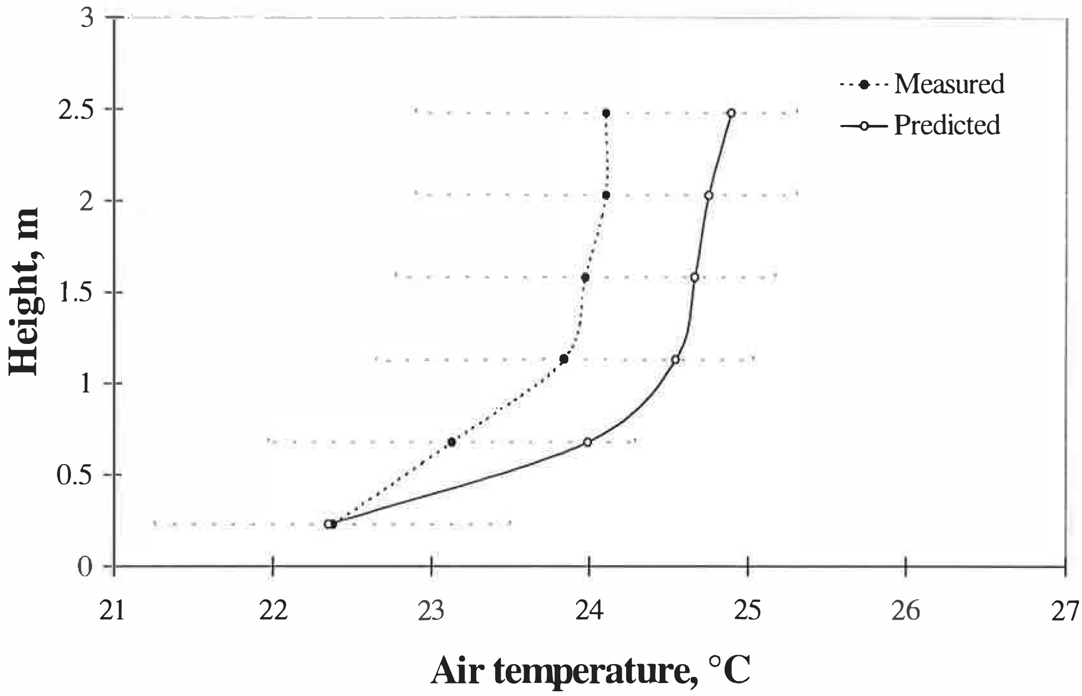


Figure 18 Variation of room air temperature and air speed with height - displacement ventilation and chilled panels (60 W/m²)



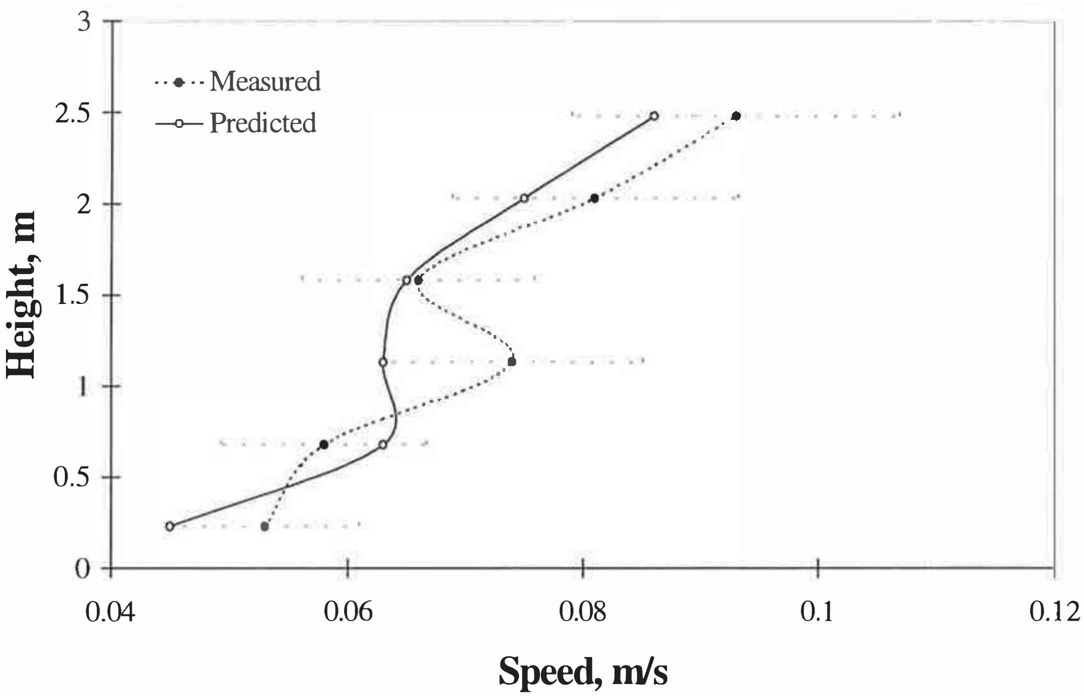
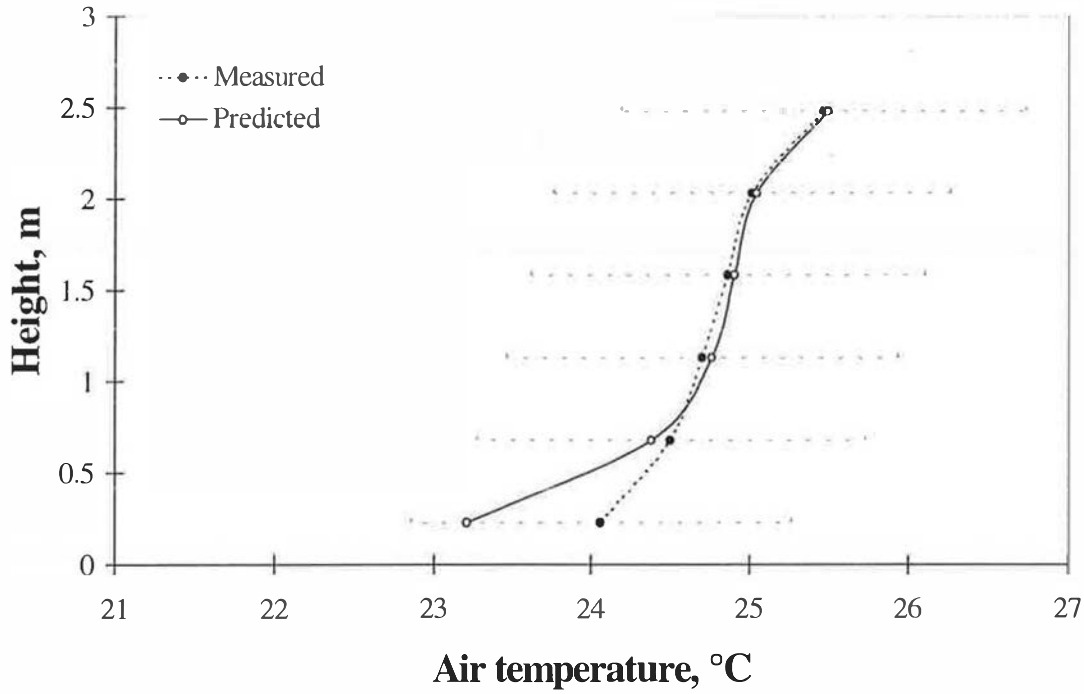


Figure 19 Variation of room air temperature and air speed with height - displacement ventilation and 'capped' chilled beams (60 W/m^2)



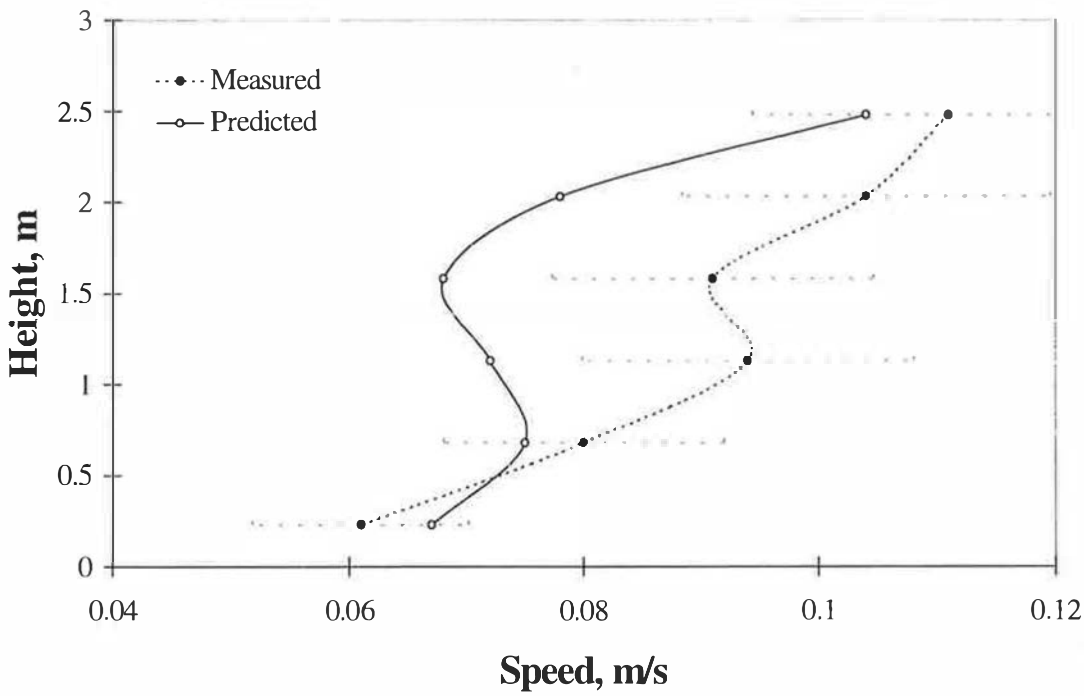
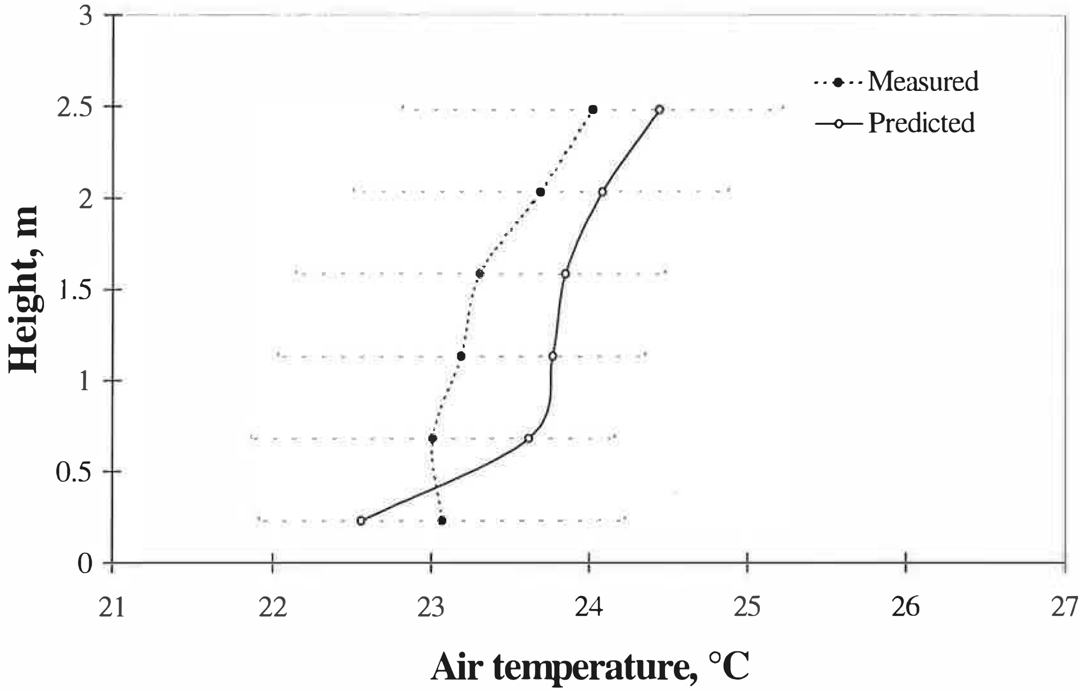
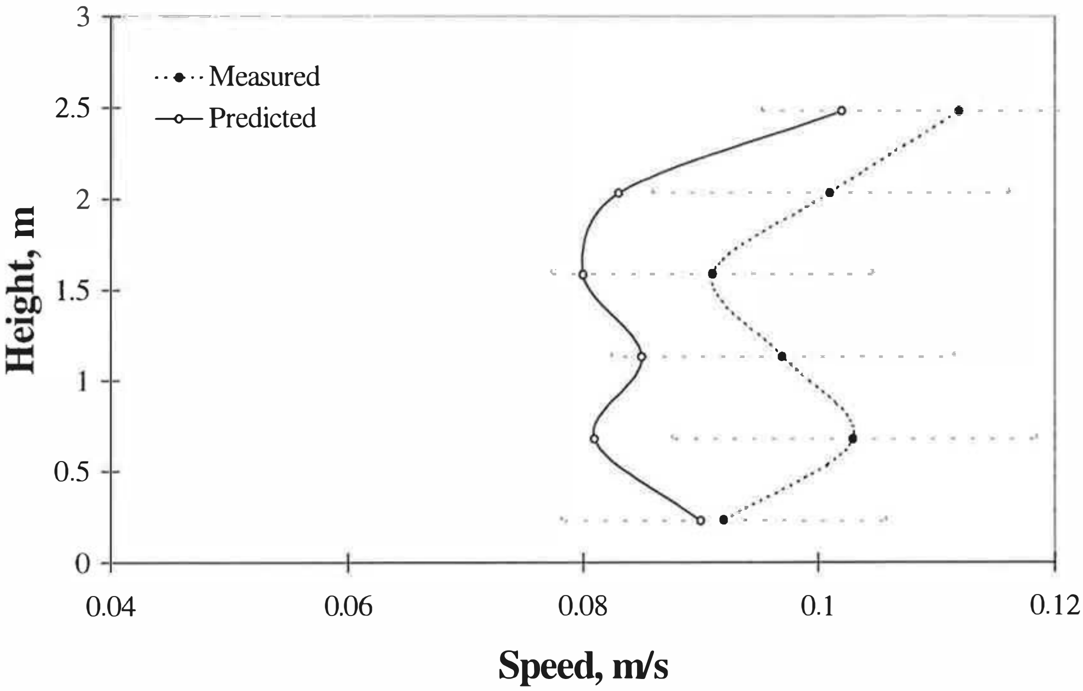


Figure 20 Variation of room air temperature and air speed with height - displacement ventilation and 'uncapped' chilled beams (80 W/m^2)





3. COMPARATIVE ENVIRONMENTAL PERFORMANCE ASSESSMENT

The information provided in Section 3 of this document is related to the work carried out involving physical and computer modelling techniques to examine the environmental performance of displacement ventilation systems with and without chilled ceilings. Due to the nature of physical modelling the conditions of the simulations were controlled within the limitations imposed by external environmental conditions. Therefore, the boundary conditions for each of the physical and numerical simulations were slightly different. Further, the ceiling layout for the different chilled ceiling products was also different. These differences in boundary conditions create some difficulties for direct environmental performance comparisons of the adopted systems. Therefore, to study the comparative effects of chilled ceiling devices, a common space was modelled with three chilled ceilings considered previously (Table 1) under given boundary conditions.

3.1 THE SPACE CONSIDERED

A smaller room than the previously used enclosure was adopted to demonstrate a comparative environmental performance of these systems under specified thermal and ventilation conditions. This represented a mid-floor office space with an external wall (45% glazing), having dimensions 4.50 m length, 4.50 m width and 3.30 m height. Within the room there is a false ceiling at 2.70 m from the floor with a flush lighting arrangement and two supply air terminals at the intersections of the internal walls. The schematic diagram of the space, the internal furniture arrangement and the ceiling layout are shown in Figures 21-24.

Figure 21 Schematic diagram of the space with corner displacement air terminals

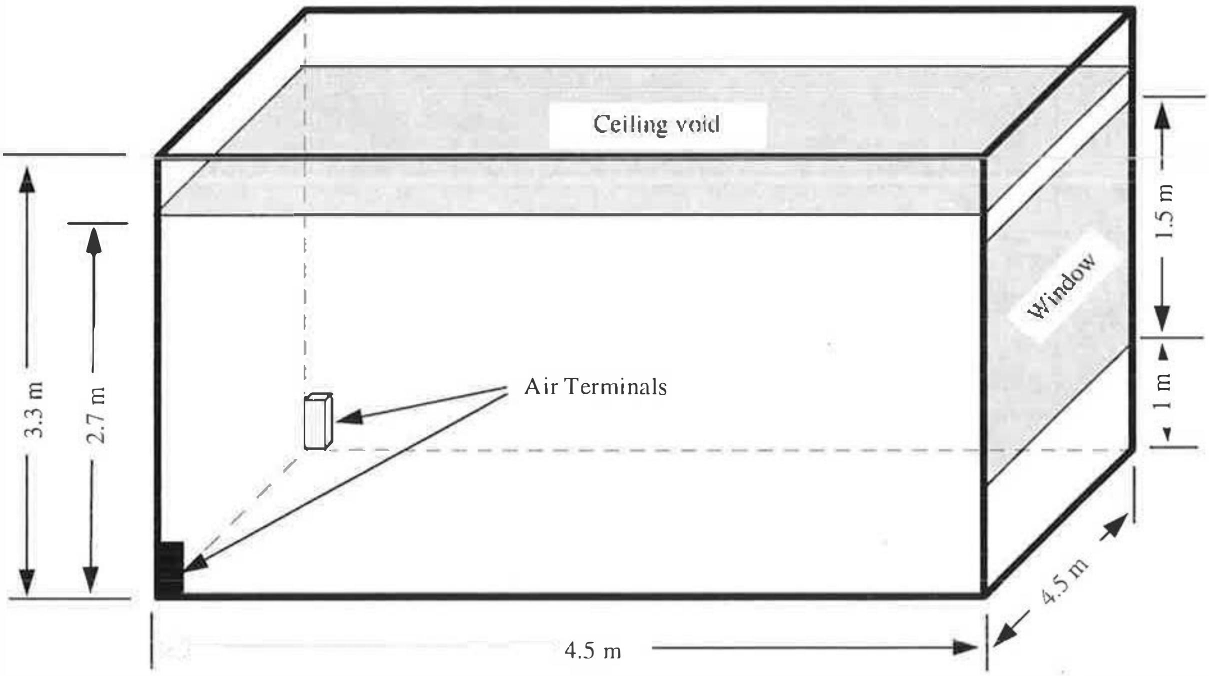


Figure 22 Room layout

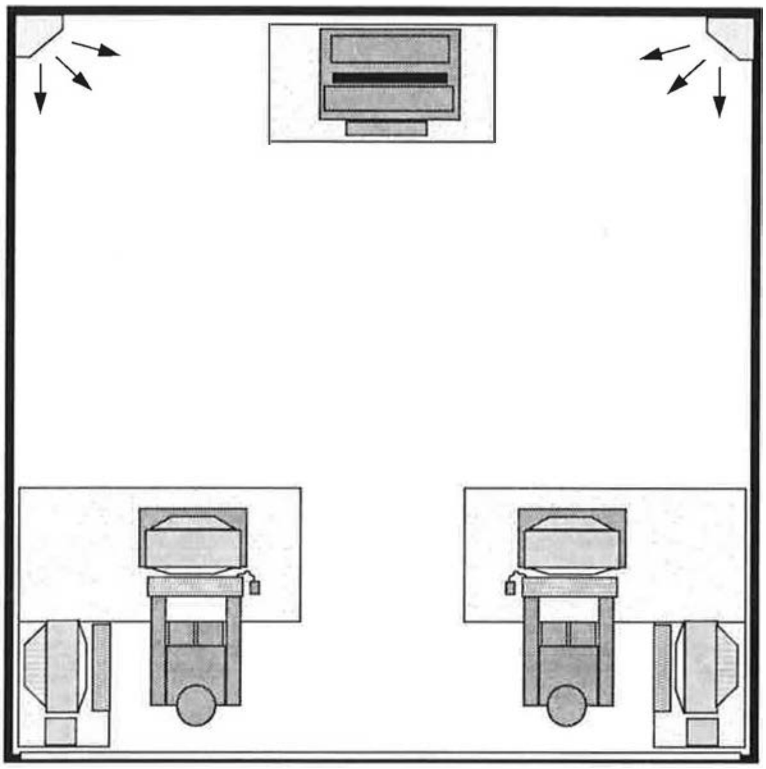


Figure 23 Ceiling layout for chilled panels

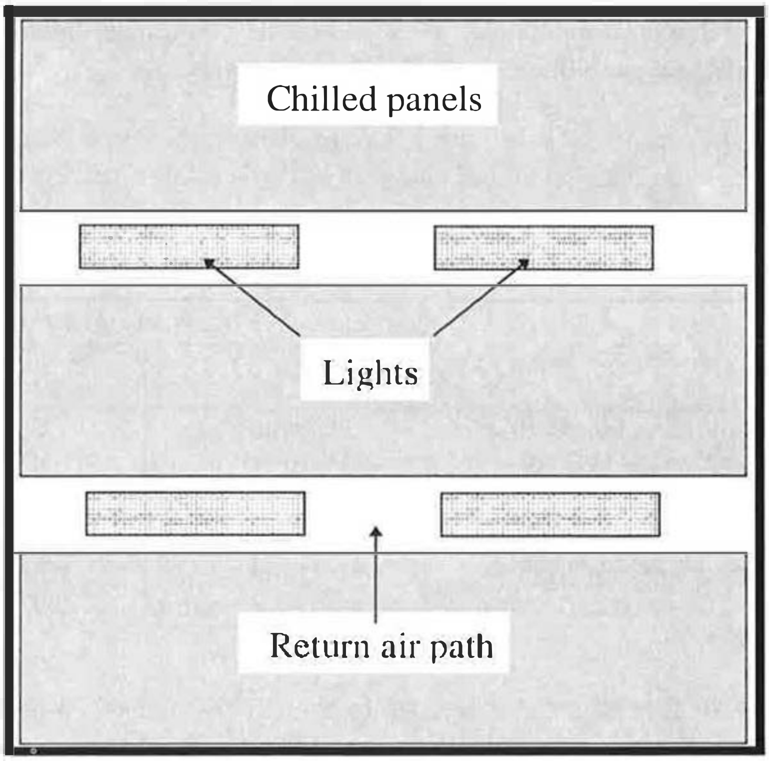
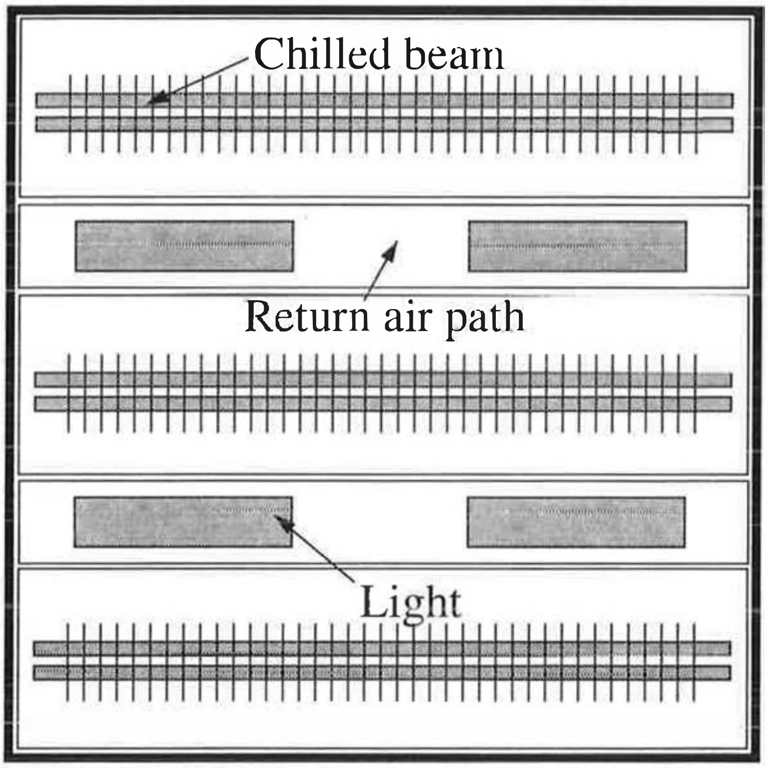


Figure 24 Ceiling layout for chilled beams



3.2 SCENARIOS CONSIDERED

The environmental performance of a displacement ventilation system with and without chilled ceiling devices was examined for a typical summer climatic condition, with an ambient air temperature of 25°C. The thermal transmittance of the external wall and window were 0.5 and 3 W/m^{2}K, respectively.

Internal heat loads of 20 W/m^2 and 60 W/m^2 were considered (see Table 4) for displacement ventilation without and with chilled ceilings, respectively.

Table 4 Internal loads

Scenario	Internal load category	Convection W	Radiation W
Displacement ventilation without chilled ceilings (cooling capacity = 20 W/m^2)	Lighting Occupancy Small power loads	30 100 90	30* 100 60
Displacement ventilation with chilled ceilings (cooling capacity = 60 W/m^2)	Lighting Occupancy Small power loads	160 100 480	160* 100 220

* *to and from the floor*

The supply air flow rate was estimated to be 3.5 ach^{-1} , based on thermal comfort design strategy (see Section 1.1.2), by assuming a vertical temperature gradient of 1.5 K/m. The supply air temperature was set at 19°C for all simulations.

3.3 PREDICTIONS

3.3.1 Flow and Thermal Field

Displacement ventilation

Figures 25-27 display the predicted velocity vectors and air temperatures within the office with displacement ventilation.

Figure 25 shows that the supply air flows in a radial pattern near the floor with the air temperatures about 22°C near the occupants. The air is then brought up to the higher levels in the convection currents generated by the occupants and small power loads, enters the ceiling void via the ceiling grilles and the light fittings, to be finally extracted from the void space (Figures 26 and 27). The distinctive features of displacement ventilation ie vertical air movement around the sources, thermal stratification with warmer air at higher levels and cooler air at lower heights, and the formation of horizontal air layers are predicted.

For the conditions considered, the warm and contaminated air layer extends from the top of the seated people to the ceiling (Figure 26 and 27). Although, the depth of this mixed region may be reduced by adopting the air quality-based calculation strategy, this will increase the ventilation rate to almost double the one considered here. However, to achieve the same thermal environment higher supply temperatures will then be required (around 21-22°C). Furthermore, it is important to appreciate that the quality of the air which would be breathed by an occupant differs from that at breathing height in the room, as it is drawn from below by the convection currents of the breathing person.

Figure 25 Predicted velocity vectors and air temperatures at a height of 25 mm from the floor

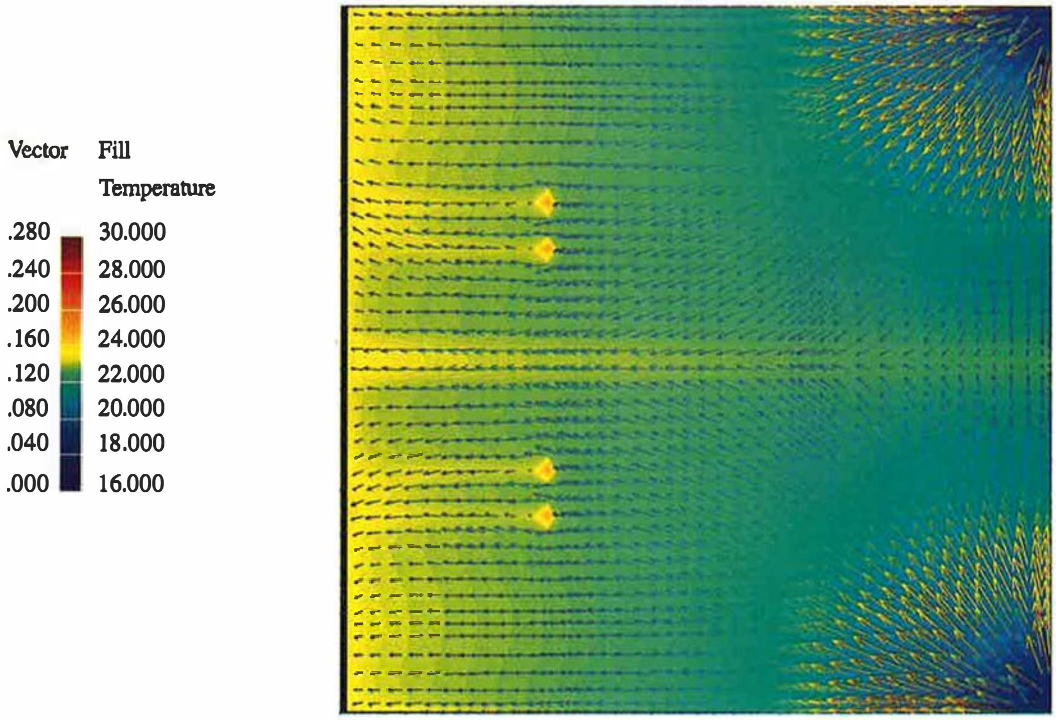


Figure 26 **Predicted velocity vectors and air temperatures at a vertical section perpendicular to the airflow from terminals**

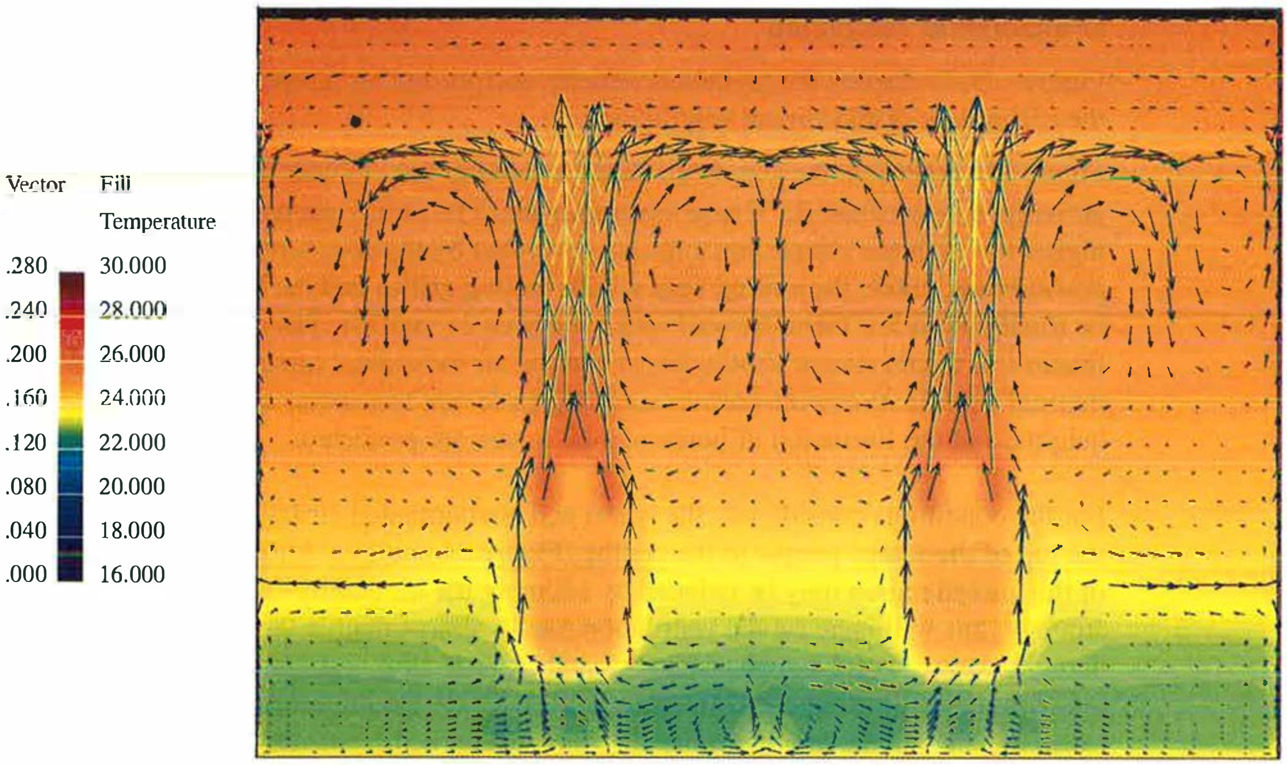
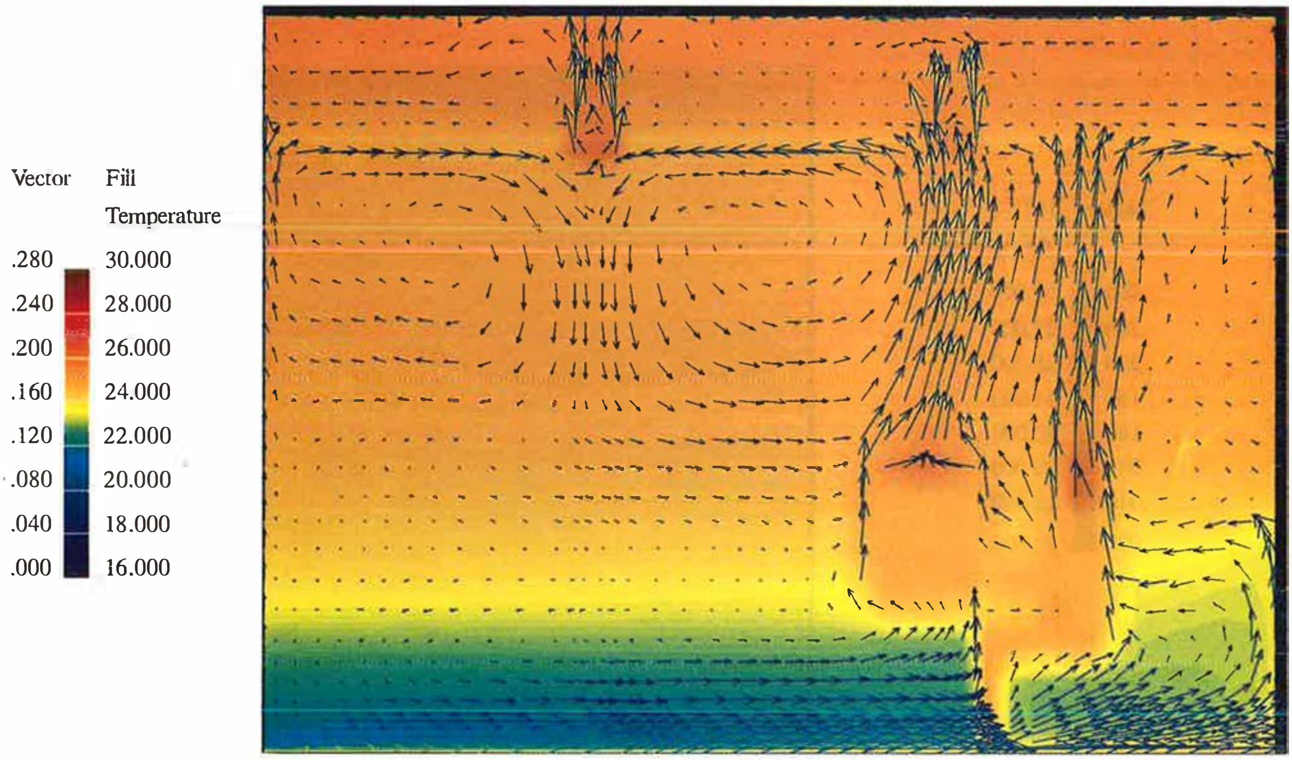


Figure 27 **Predicted velocity vectors and air temperatures at a vertical section along the airflow from terminals**



Displacement ventilation with chilled ceiling devices

The predicted airflow patterns indicate that the chilled panels created a more downward convection than the pure displacement ventilation (see Figures 27 and 28). In the vicinity of the occupants, however, the upward convection was predominant.

The flow field in the case of chilled beams gave a more mixed condition within the occupancy zone than with the chilled panels (see Figures 28, 29 and 30). Further, appreciable differences in the predicted room air movement and temperature distribution were observed between the 'capped' and 'uncapped' beams. The downward convection from the latter systems was found to be stronger than the 'capped' system. In both systems, however, downward air flows were predicted along the far wall which merge with the displacement cool air layer. This may cause higher air velocities and lower air temperatures at low levels, hence may result in localised cold discomfort. To avoid this problem the operating temperatures and cooling outputs may need to be restricted.

Figure 28 Predicted velocity vectors and air temperatures at a vertical section along the airflow from terminals - with chilled panels

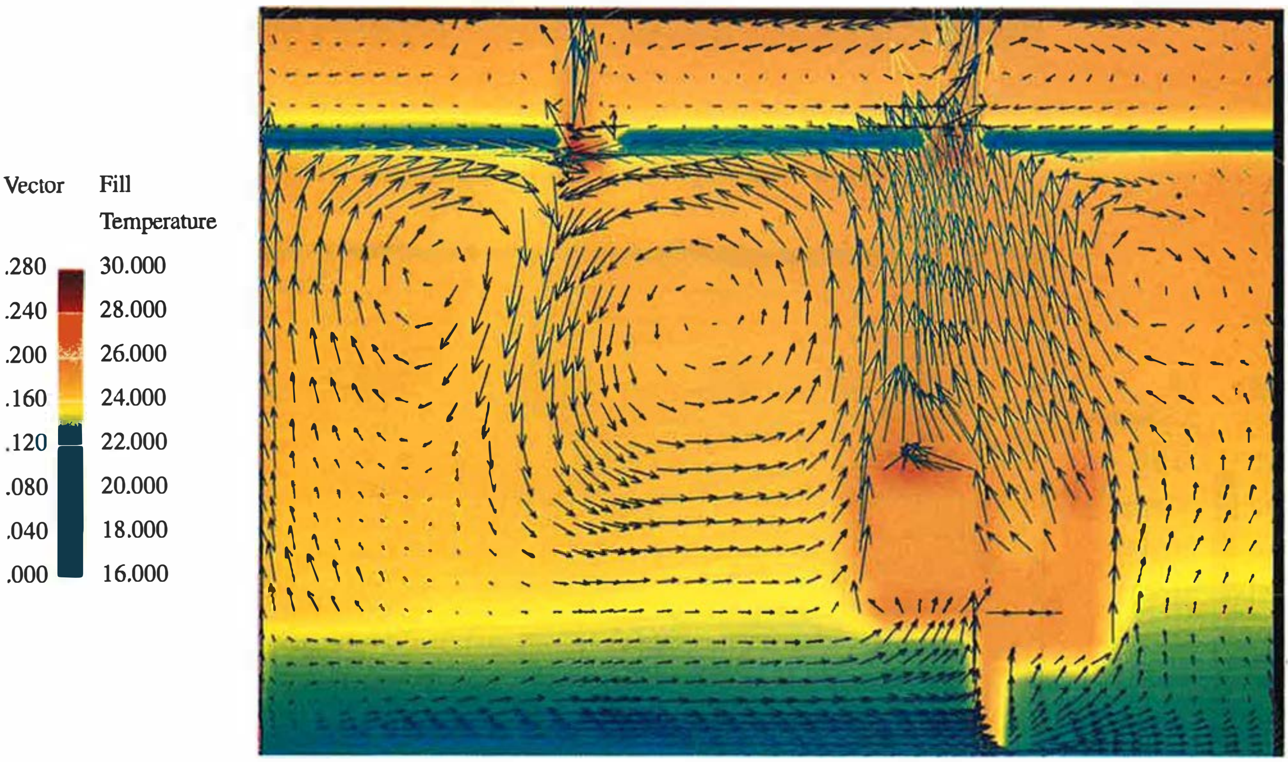


Figure 29 Predicted velocity vectors and air temperatures at a vertical section along the airflow from terminals - with 'capped' chilled beams

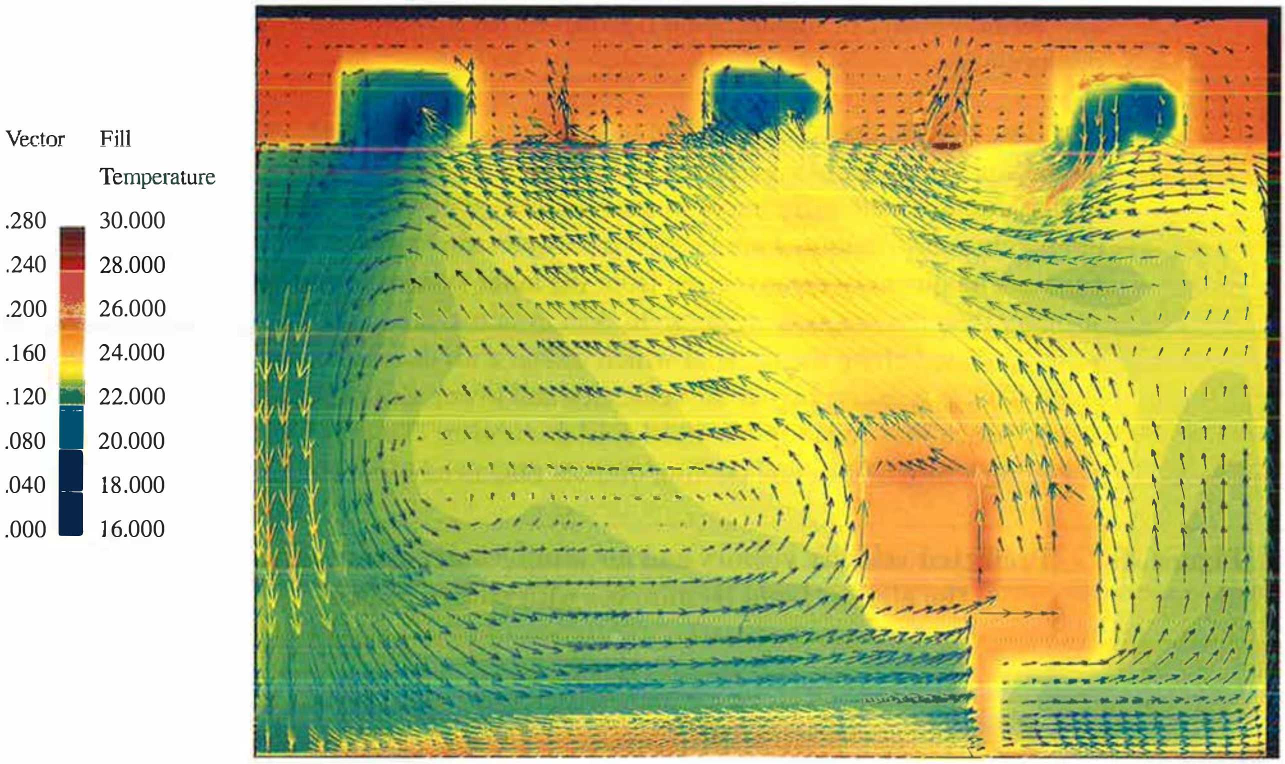
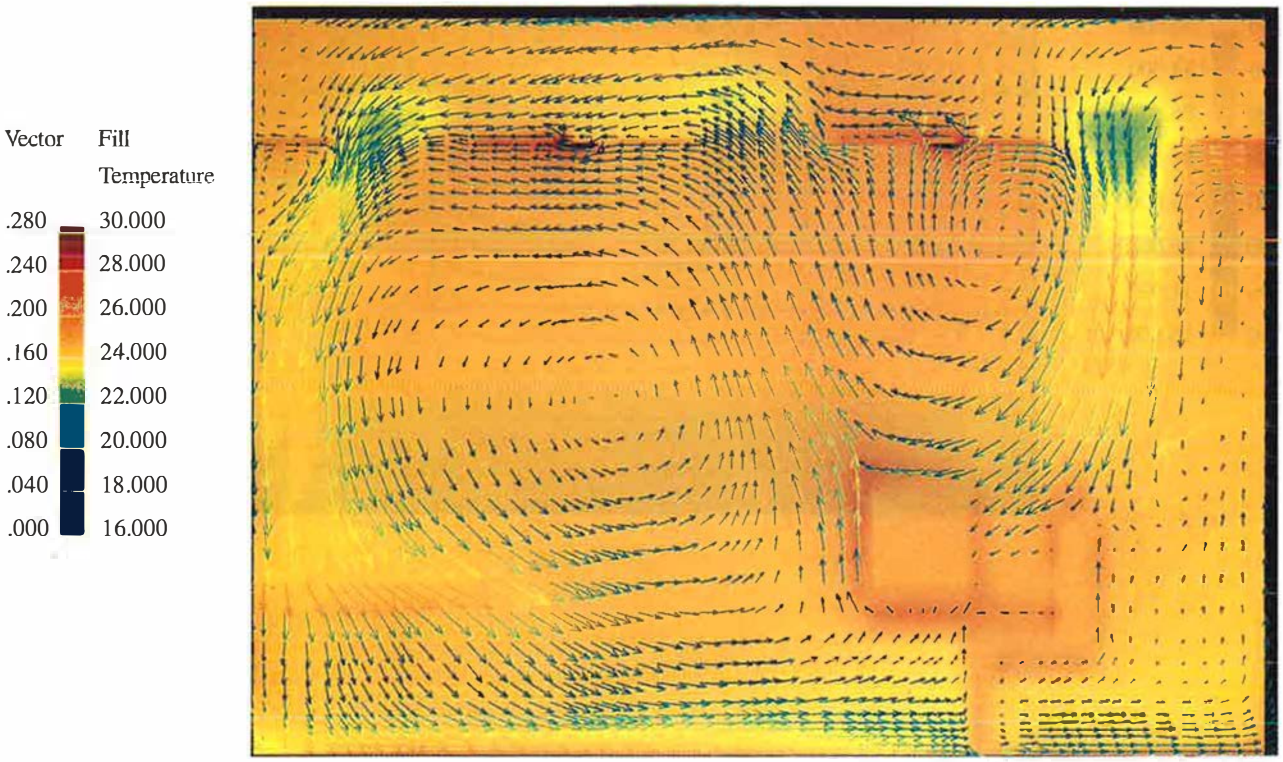


Figure 30 Predicted velocity vectors and air temperatures at a vertical section along the airflow from terminals - with 'uncapped' chilled beams



3.3.2 Vertical Air Temperature and Speed Profiles

The predicted plane-averaged air temperature and speed for displacement ventilation with and without chilled ceilings are displayed in Figure 31. The vertical temperature gradients in the occupancy zone in the case of displacement ventilation with and without chilled panels are about 1.2-1.7 K/m, whilst for displacement ventilation with chilled beams they are about 0.6 K/m. The depth of the upper mixed layer, therefore, may be judged to be above these heights where there are no significant temperature variations. Clearly, in the case of displacement ventilation and chilled beams, the downward convection was increased to the extent that the resulting flow field was virtually similar to a conventional mixed airflow system, except in the vicinity of the heat sources where upward convective plumes entrain air from the displacement cool air layer at floor level.

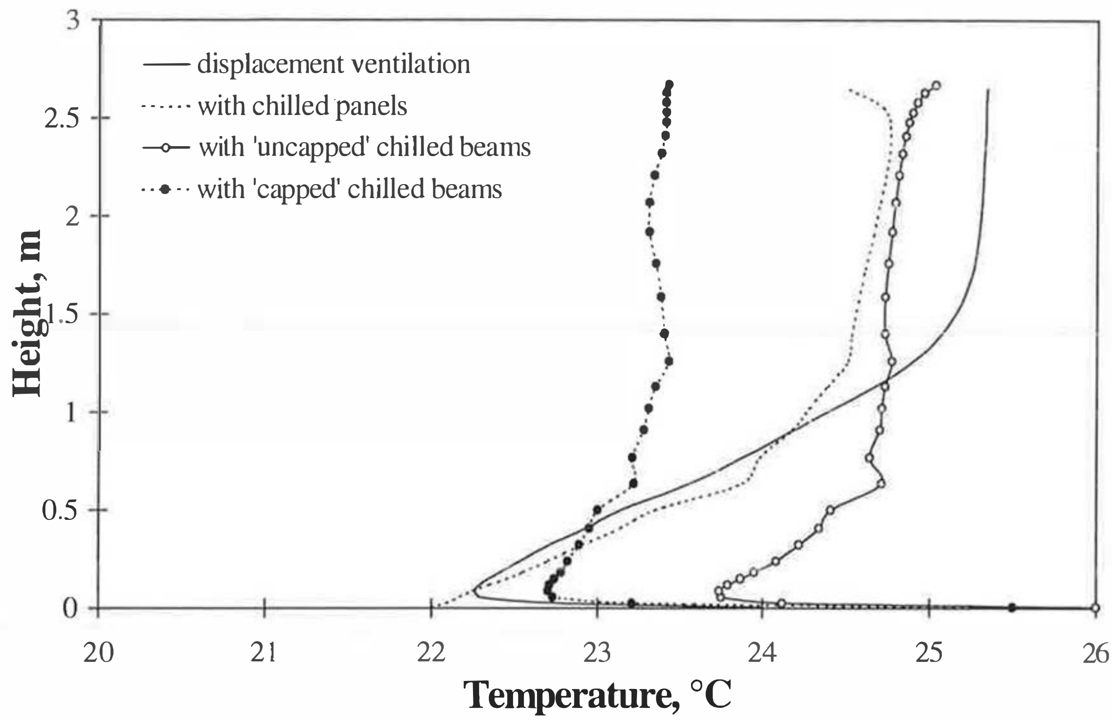
3.3.3 Thermal Comfort

The predicted mean vote and the percentage of occupants' dissatisfaction were well within the recommended ranges given by ISO 7730[5]. Indeed, in all conditions considered, thermal comfort was of a very high order, ie better than or equal to 90% occupant satisfaction. Figure 32 shows the variation of the predicted thermal comfort indices with room height.

The displacement ventilation with chilled panels and 'uncapped' chilled beams provided an acceptable thermal environment within the majority of the occupancy zone, being in the slightly warm side of the PMV values. Although the displacement ventilation with the 'capped' chilled beams also created acceptable thermal conditions, the PMV values were in the slightly cool side of the plot (Figure 32). Indeed, at low levels where the downward convective airflows close to and along the far wall merge with the displacement cool air layer (see Figure 29) the combination of lower air temperatures and higher speeds may have resulted in cold discomfort near the floor. In these region the PPD values slightly exceeded the acceptable levels (ie ±10%). This situation can be improved by reducing the operating temperatures and cooling outputs of the beams.

Air flow patterns around the people suggest that in all but the most disturbed conditions (uncapped chilled beams, Figure 30) the air at the breathing zone was drawn from low level and so is likely of higher air quality than if it were drawn from head level or above.

Figure 31 Predicted vertical air temperature and speed profiles



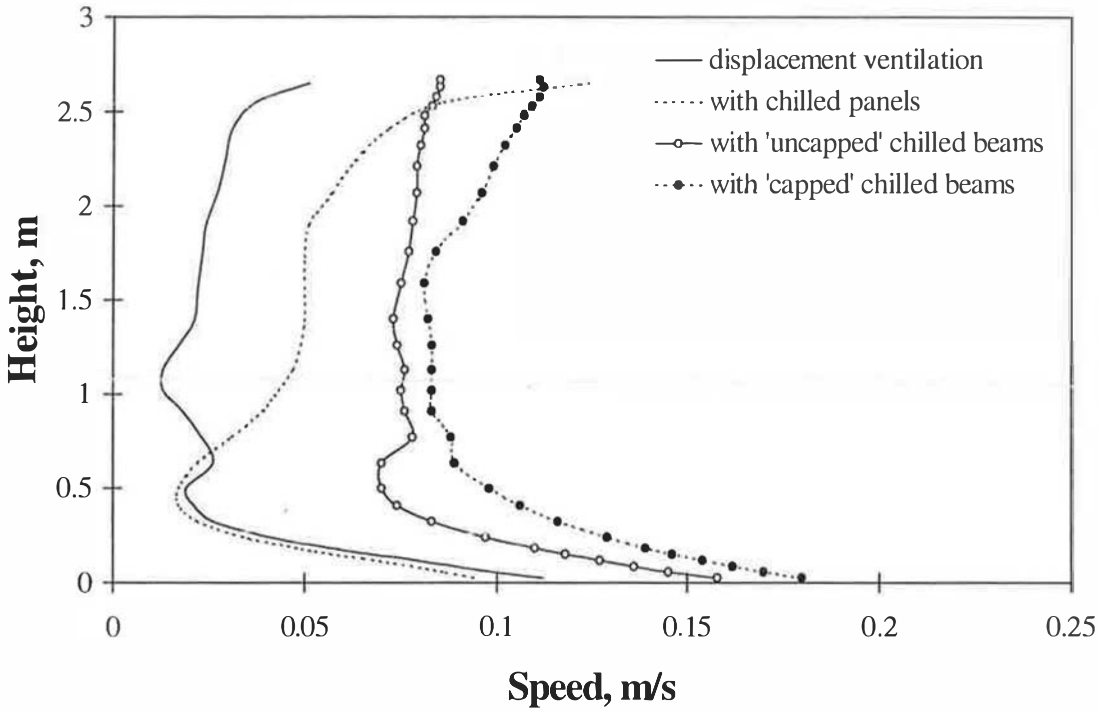
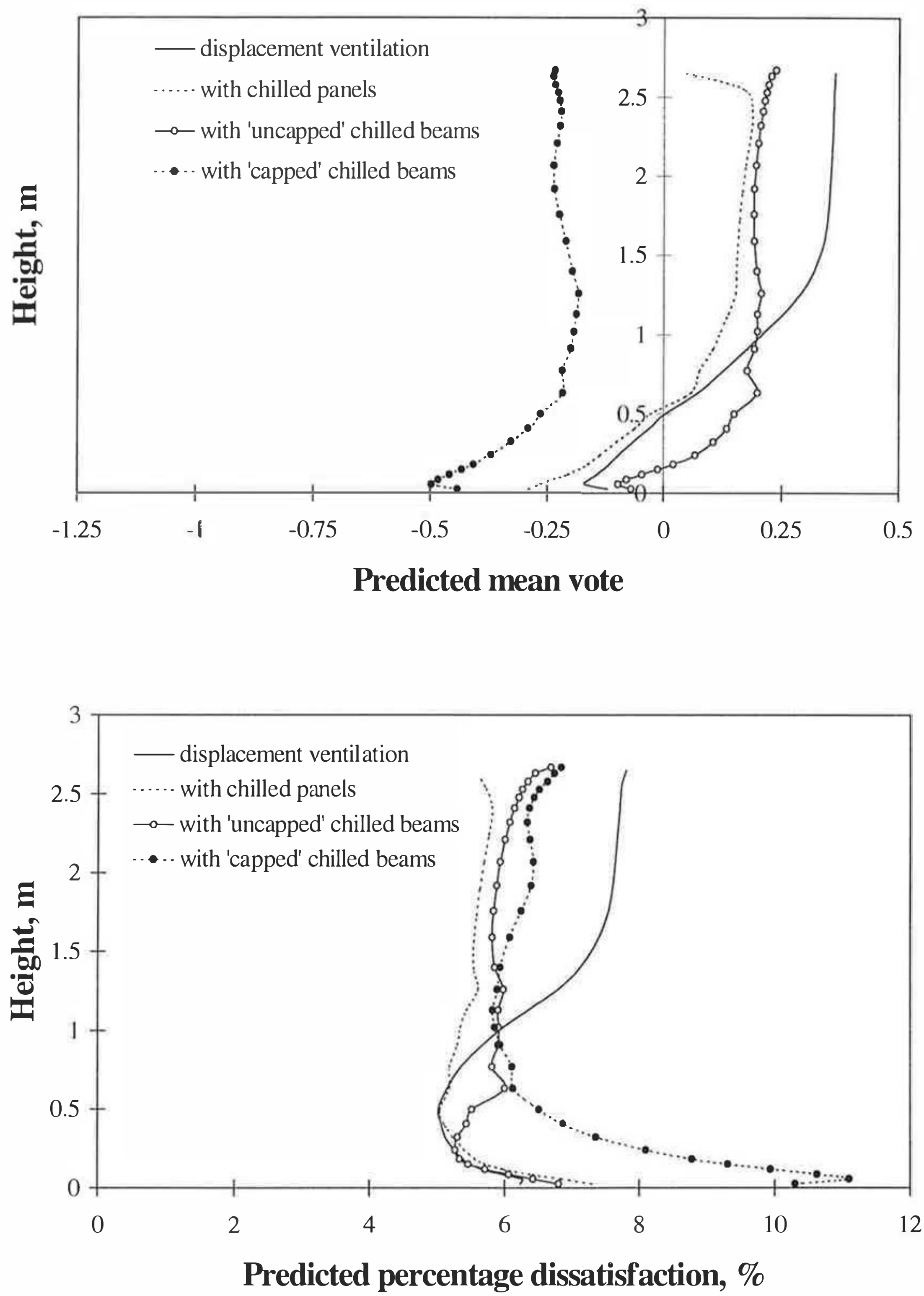


Figure 32 Predicted thermal comfort indices



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