Technical Note TN 3/93

DISPLACEMENT VENTILATION PERFORMANCE - office space application

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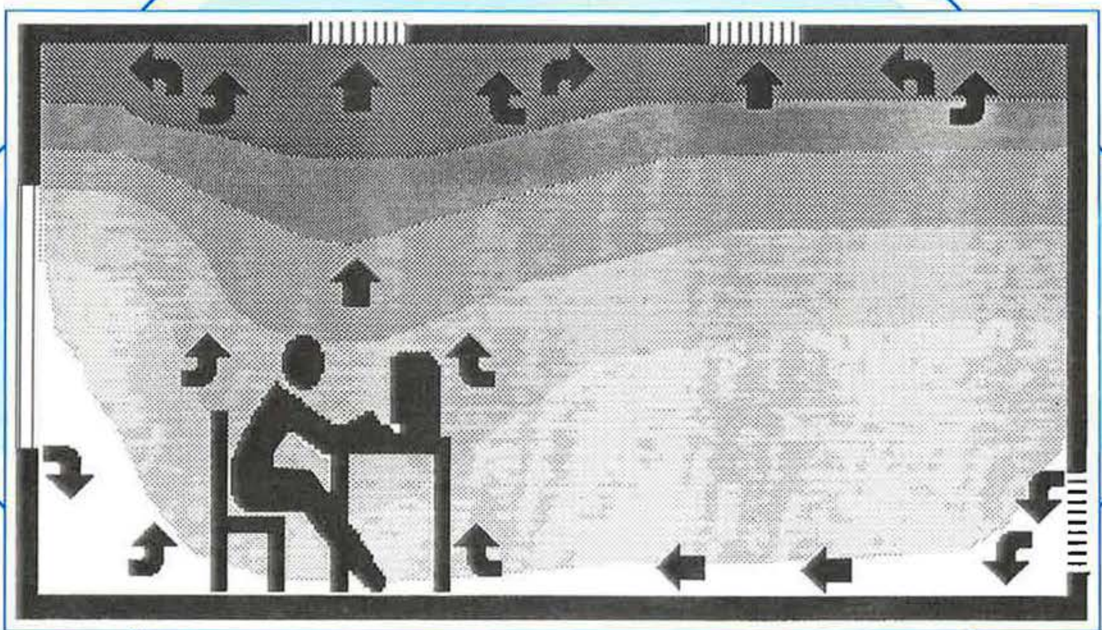

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SUMMARY

Displacement ventilation is a method that provides conditioned air to indoor environments with the view to improve air quality whilst reducing energy usage. These systems have been employed in industrial applications, notably in Scandinavia, for many years and have gained in popularity in office building spaces in recent years.

Measurements of velocity and temperature fields have been performed in three modern office spaces to analyse the performance of the incorporated displacement ventilation system(1), based on thermal comfort and temperature gradient assessment(2).

The measurements at one of the sites were used to verify the predicted data obtained by a microclimate computer model based on computational fluid dynamics (CFD) techniques. The model was subsequently used to study the effect of cold surfaces, infiltration, floor obstructions and supply air flow rates.

Although the application of displacement ventilation is more difficult in office spaces with low ceiling heights, acceptable conditions were measured in two of the three sites considered in this study. However, both measurements and predictions indicated that secondary air flows resulting from infiltration and cold surfaces can adversely affect the ventilation performance and reduce thermal comfort. In the application of displacement ventilation care should be taken to minimise these extraneous effects.

1. INTRODUCTION

In recent years, considerations of thermal comfort and environmental quality within building spaces have taken on a high profile in design and maintenance of controlled and safe environments.

A comfortable and clean indoor environment can be designed by the adoption of an effective ventilation system, both in terms of providing thermal comfort and removing contaminated air. In general, buildings may be ventilated naturally or mechanically. The latter case was traditionally achieved by diluting the contaminated indoor air with 'fresh' incoming air and removing the polluted air from a suitable location. Although this method offers a uniform, thermally comfortable environment, it produces an indoor air quality resulting from the mixture of indoor pollutants with the incoming air. The alternative ventilation method is to introduce the supply air at one part of the room and allow it to sweep in one direction across the space taking the pollutants with it for exhaust at the opposite part of the room. Displacement ventilation is in this category and is characterised mainly by buoyancy-driven air flows. In these systems, low velocity air is supplied from a low-level supply device into the occupied zone at a temperature slightly cooler than the design room air temperature[2]. The air moves along the floor in a radial pattern to create a cool layer near the floor. Convective plumes from internal heat sources entrain air from this layer into the occupied zone and cause an upward movement, with the air increasing in temperature as it rises. The warm, contaminated air forms a stratified region under the ceiling which is then exhausted at high-level. Figure 1 illustrates this concept.

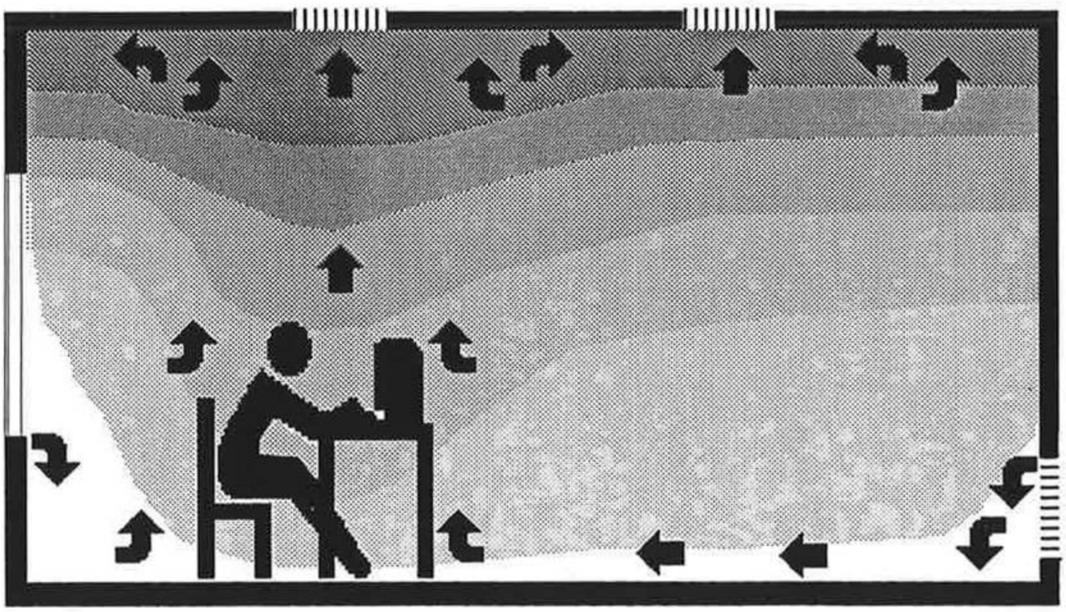


Figure 1: Buoyancy-driven displacement flow

In principle, displacement ventilation systems could potentially offer both thermal comfort and air quality required in spaces with thermal loads at a lower flow rate and higher supply air temperature than (that required to achieve an equivalent effect) with a 'mixed-flow' system. In practice however, there are a number of influential factors such as the counter-flow air movement as the result of cold surfaces and infiltration, and the existence of furniture and obstacles in the occupied zone, which could reduce the performance of these systems. In addition, displacement ventilation systems have been used mainly in industrial applications where cooling requirements are predominant, where heat sources and contaminants are at high temperatures, and where spaces are tall. Therefore, in office spaces with low ceilings, effective applications could be more difficult[2].

In an effort to provide further information on the performance of these systems in office building spaces, a study was undertaken at BSRIA which involved both detailed site investigations and numerical analyses using computational fluid dynamics (CFD) techniques[1].

The measured results were used to assess the acceptability of the room conditions in relation to thermal comfort and to verify the computational predictions. The computational technique was then used to study the impact on the indoor environment of such factors as cold downdraughts, infiltration, floor-mounted obstructions and supply air flow rates.

2. THE SPACES CONSIDERED

The information provided in this document relates to three of the sites studied in the original work[1] which incorporated displacement flow systems. These sites were:

Site 1 – an office building which has three levels. Each floor has an area of 828 m². Only the first and second floors of the building were considered in this study. The first floor is mainly open-plan with some discrete offices in two corners. On the second floor the space is divided into two regions, one largely open-plan and the other a mix of discrete offices and a lounge-cum-reception. The open-plan spaces are equipped with a displacement supply air system with a central supply/extract air handling unit located in the main plant room. Air is supplied via low-level terminals adjacent to the centreline columns. Extract air enters the ceiling void via air-handling luminaires, from where it is drawn to the extract duct situated within the void.

Site 2 – largely open space with a rectangular atrium in the centre. A mezzanine floor is constructed above the main floor and around the four sides of the atrium. The main floor has an area of some 1,330 m², of which the atrium occupies 242 m², and the mezzanine has an area of 1,088 m². The main open-plan floor and mezzanine were surveyed in this work. The site employs a system of displacement ventilation and comfort cooling, using 100% 'fresh' air. Supply ducts are routed to various parts of the building at ceiling level in the ground floor to emerge through the concrete floor of the main office to supply air to low-level terminals. Air is extracted on the main floor by arrays of spiral wound ducts which rise up through the floor to terminate just short of the ceiling. On the mezzanine floor, the extract ducts emerge near the centres of the external walls and follow the pitch of the ceiling to terminate close to the apex.

Site 3 – comprises three wings attached to a curved building. The curved section forms the main part of the building, and has a full-length atrium on the convex side and three blocks of offices on the concave side. The middle block which has three floors, each having an area of 464 m², was considered in this study. An open space exists on each level with an area of 181 m² with discrete offices to either side. 'Fresh air' from the main plantroom is fed to the floor voids, where it mixes with air drawn from the building prior to issue to the office. Extract air is removed at a height of approximately 2.0 m from floor levels.

The main thermal loads in the buildings were due to lighting, office equipment and occupants. The ventilation systems were either of the constant air volume type or of the variable air volume type. Listed in Table 1 are the thermal loads and the ventilation flow rates for the sites surveyed.

Site	Thermal Loads (W/m²)	Supply Air Flow Rates	
		Per Diffuser (l/s)	Per Unit Area (l/s/m²)
1–1	25.5	190	4.3
1–2	6.2	190	3.5
2–1	13.5	198	4.2
2–2	6.3	300	3.6
3–1	31.3	16	0.9
3–2	27.8	16	1.0
3–3	21.9	16	0.9

Table 1: Heat gains and ventilation flow rates

3. ENVIRONMENTAL PERFORMANCE ASSESSMENT - THE METHOD

To assess the performance of a displacement flow system within an office, it is necessary to obtain values of air velocities, temperatures and concentrations of pollutants within the space. This can be achieved by one or a combination of site measurements, physical modelling, and computer modelling techniques. These data may then be used to analyse environmental thermal comfort.

In the present study a combination of site measurements and computer modelling has been used to analyse the effectiveness of the ventilation system. The latter approach has also been employed in a series of parametric studies to analyse the effect of various influential parameters and the results were compared with the generalised calculation methods outlined in TM 2/90[2].

3.1 SITE MEASUREMENT

Field measurements of the air flow parameters were made in real situations using a Dantec 54N10 multi-channel flow analyser. The instrument has six probes, each consisting of air velocity and temperature transducers in close proximity, to sample conditions effectively at a point.

Measurements of air velocities and temperatures were made over 120 second periods at fixed regular heights and at locations set on a rectangular grid[1].

3.2 NUMERICAL MODELLING

Air flow and temperature distribution within an enclosure are governed by the principles of conservation of mass, momentum and thermal energy. These conservation laws may each be expressed in terms of partial differential equations, the solution of which provides the basis for a CFD model. The CFD model employed in the present study was the commercially developed code, FLOVENT[3]. The code solves finite-volume approximations to the governing equations formulated in terms of 'primitive' pressure-velocity variables. The closure of these equations is obtained through turbulent exchange coefficients for momentum, calculated via transport equations for both the turbulence kinetic energy and its dissipation rate. The transport of heat is modelled using the effective Prandtl number approach, while 'wall-functions' are employed to bridge the steep property gradients in near-wall regions. The FLOVENT code solves 'upwind' finite-volume equations for a three-dimensional, predetermined size, staggered grid in an iterative manner using the SIMPLE (semi-implicit method for pressure linked equations) algorithm[4].

3.3 THERMAL COMFORT CALCULATION

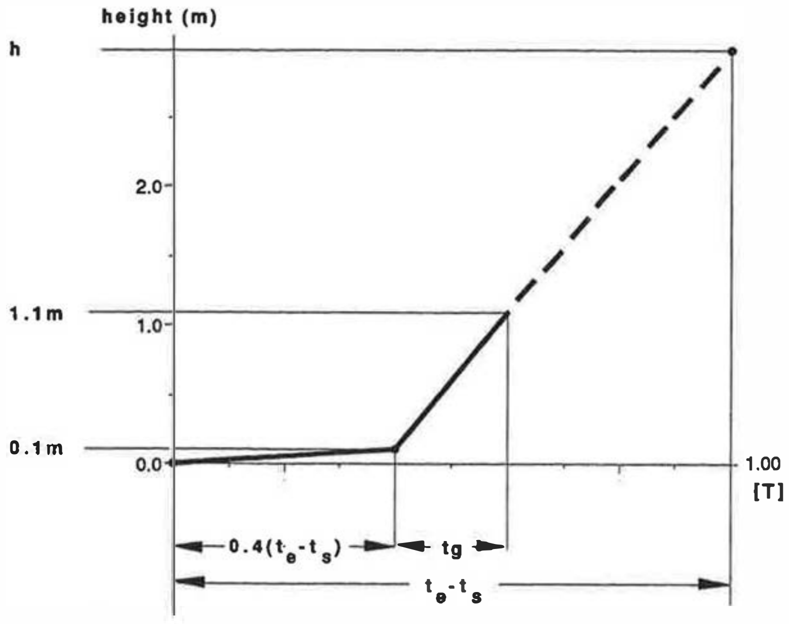
Thermal comfort assessment can be based on the perception of satisfaction or dissatisfaction that a subject experiences with the thermal environment. In this context, thermal comfort indices may be defined as two linked factors, taking into account the combination of physiological and environmental parameters, the result of which may be produced in the form of a psycho-physical scale[5]. This runs from –3 (cold) through 0 (neutral) to +3 (hot), and is known as the predicted mean vote (PMV). Values outside this range indicate unacceptable conditions. A further index, the predicted percentage of dissatisfied (PPD) gives the percentage of occupants who would be unhappy with their thermal environmental conditions.

The uniformity of the conditions in the space can be defined by the air diffusion performance index (ADPI). This takes into account the influence of an effective draught temperature which is a function of local air velocity, temperature, and mean room temperature[6]. For acceptable conditions the draught temperature must lie in the bounds of –1.7 K and +1.0 K and the local velocity must not be greater than 0.35 m/s.

3.4 AIR TEMPERATURE GRADIENT

ISO Standard 7730[5] recommends that the temperature difference between head and feet should be less than 3K for the sedentary position. This gives a gradient of 3 K/m. Other sources[2] recommend a lower value of 2 K/m for sedentary occupation and up to 3 K/m for a standing position engaged in heavy work.

The relationship between the temperature gradients in the occupied zone and the overall temperature difference have been generalised in TM 2/90[2]. The basic assumptions were that the gradient is 'linear' from a height of 0.1 m above the floor level to ceiling, that the exhaust air temperature is the same as that at the ceiling, and that the difference between the temperature of the air at 0.1 m above the floor and the supply air temperature is 0.4 times the overall exhaust-to-supply air temperature difference. These assumptions are considered valid for office space applications with room height up to 4 m. Based on these assumptions and adopting a dimensionless axis similar to that employed in TM 2/90, typical variation of room air temperature with height may be produced (Figure 2).



$[T] = (t_x - t_s)/(t_e - t_s)$
t_e = exhaust air temperature (°C)
t_s = supply air temperature (°C)
t_x = room air temperature at various heights (°C)
h = room floor-to-height (m)
t_g = air temperature gradient (K/m) = $0.6\ (t_e - t_s)/h - 0.1)$

Figure 2: Typical variation of room air temperature with height

4. MEASUREMENTS AND COMPUTATIONS

4.1 FLOW AND THERMAL FIELD

The measured data for air velocities and temperatures at one of the sites (i.e. site 1–1) are compared with the computed results of the FLOVENT program (see Table 2). Good agreement is displayed for averaged temperatures at each measured level. However, appreciable differences are observed for velocities. This is in part due to experimental errors (ie precision in reading very low velocities) and in part due to modelling assumptions[7]. The three-dimensional computer model representation of the space is presented in Figure 3. Figure 4 shows the predicted vector velocities at the height of 0.5 m from the floor. The predicted and measured air temperatures and velocities are shown in Figures 5 and 6 respectively.

Level	Average Temperature (°C)		Average Velocity (m/s)	
	Measured	Computed	Measured	Computed
0.15	21.8	22.0	0.09	0.09
0.49	22.1	22.2	0.08	0.06
0.83	22.2	22.5	0.09	0.05
1.17	22.3	22.8	0.09	0.04
1.51	22.7	23.1	0.08	0.03
1.85	22.8	23.2	0.07	0.03
mean	22.3	22.6	0.08	0.05

Table 2: Air velocity and temperature comparison

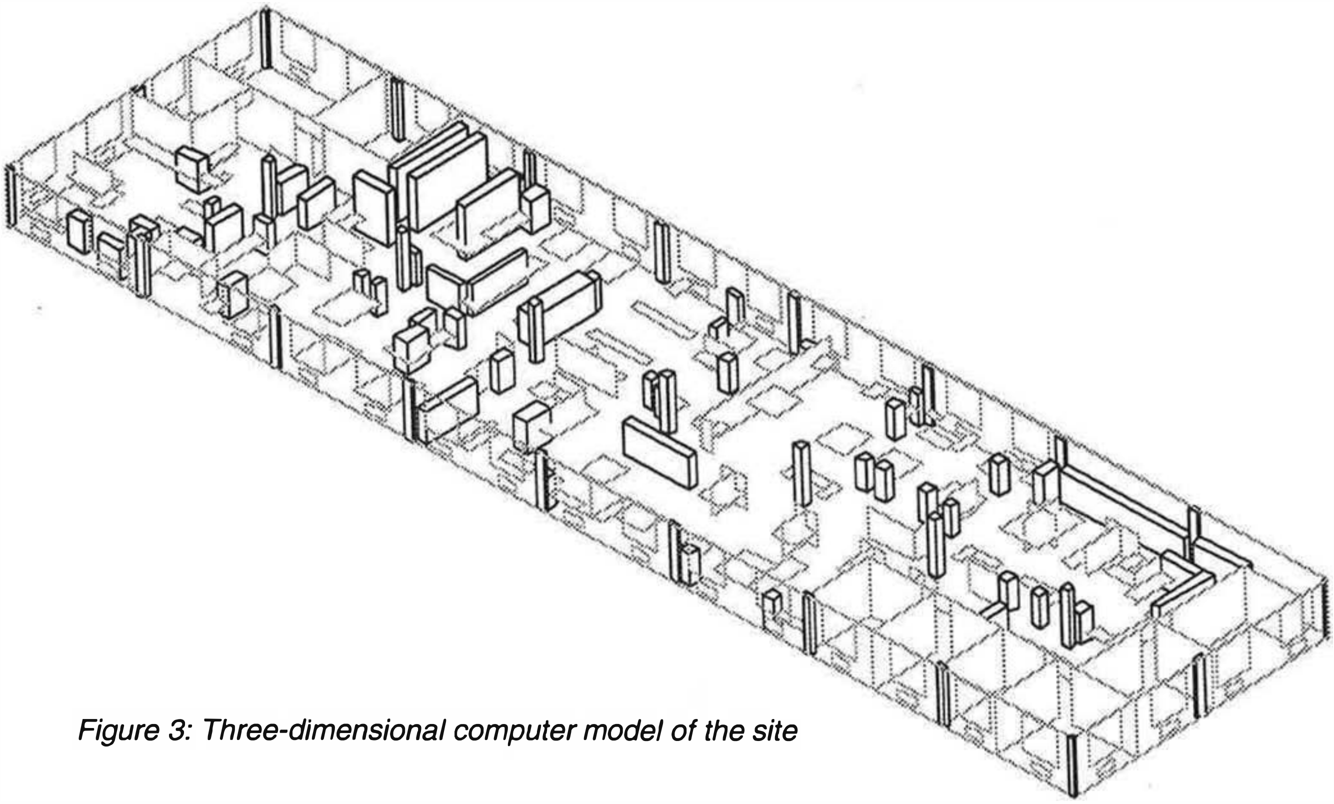


Figure 3: Three-dimensional computer model of the site

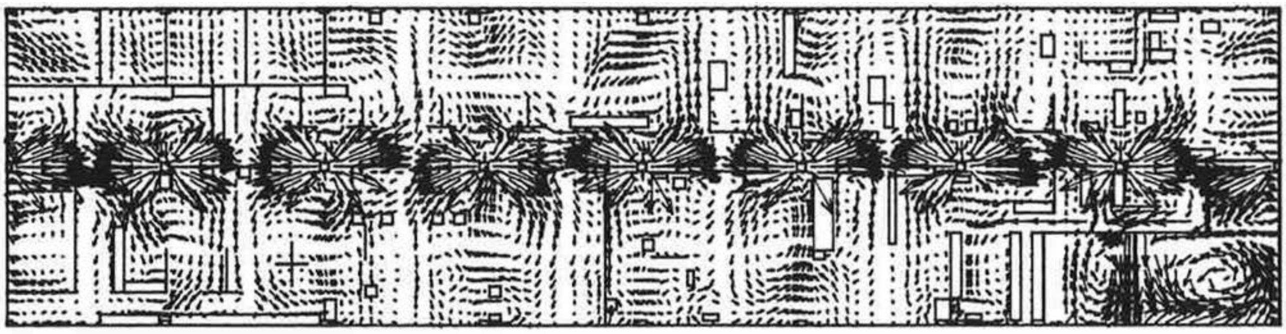
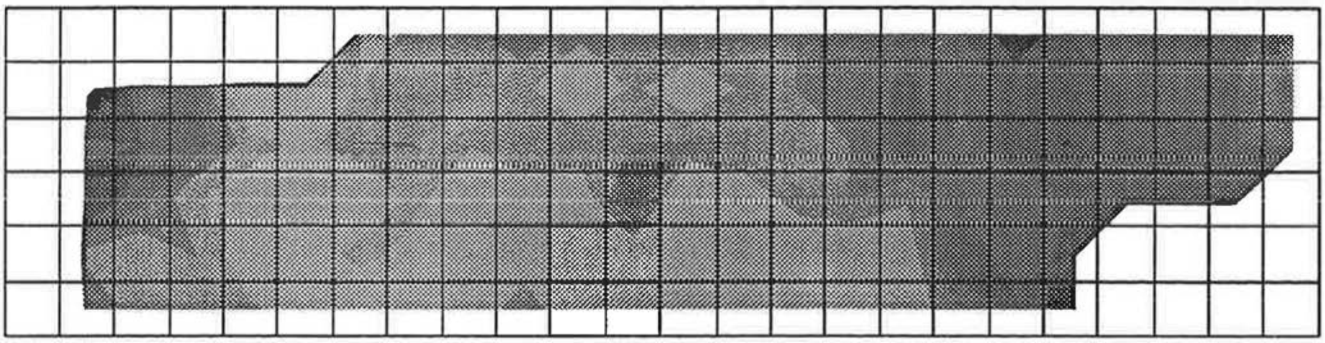
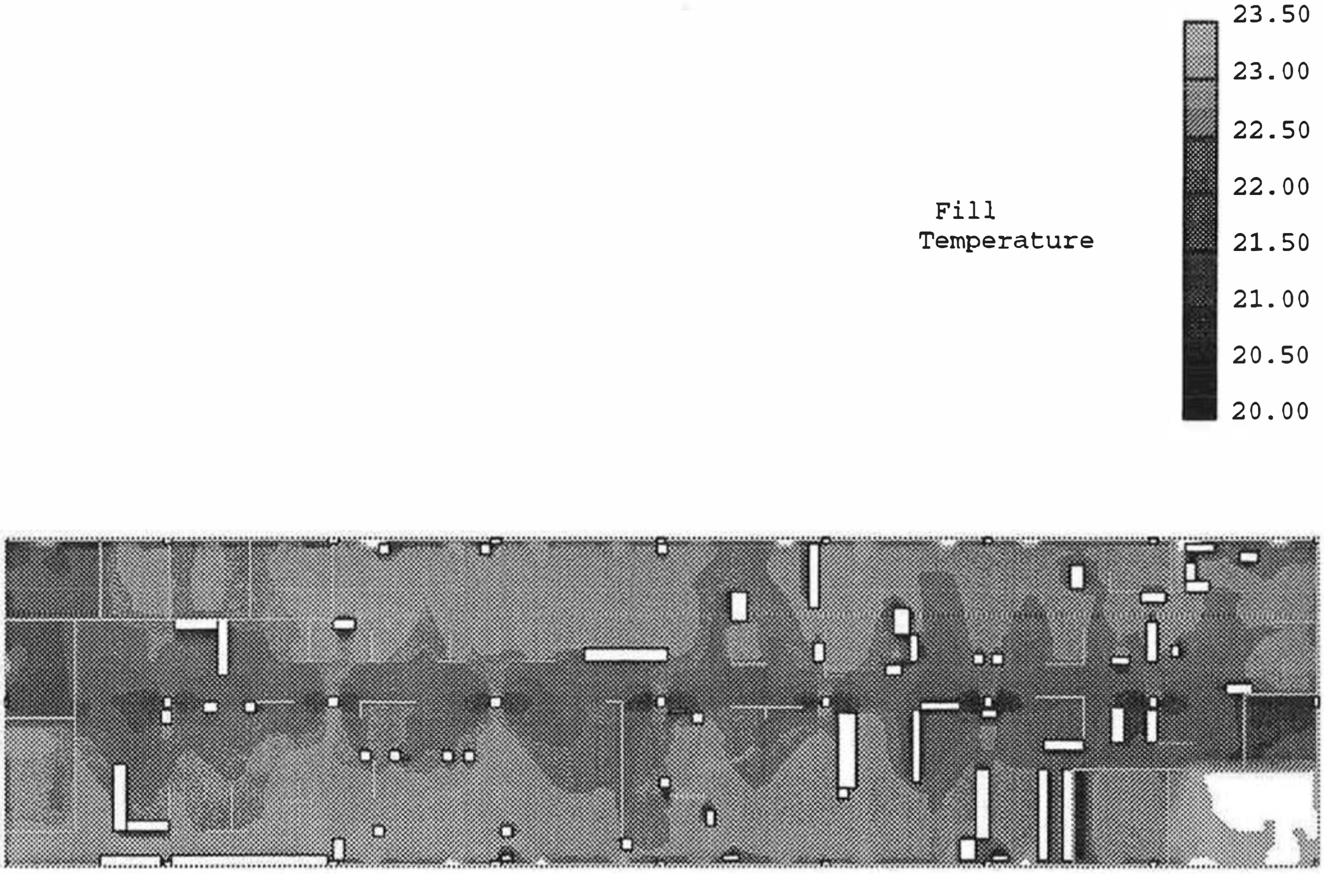


Figure 4: Predicted air velocities at a height of 0.5 m from the floor

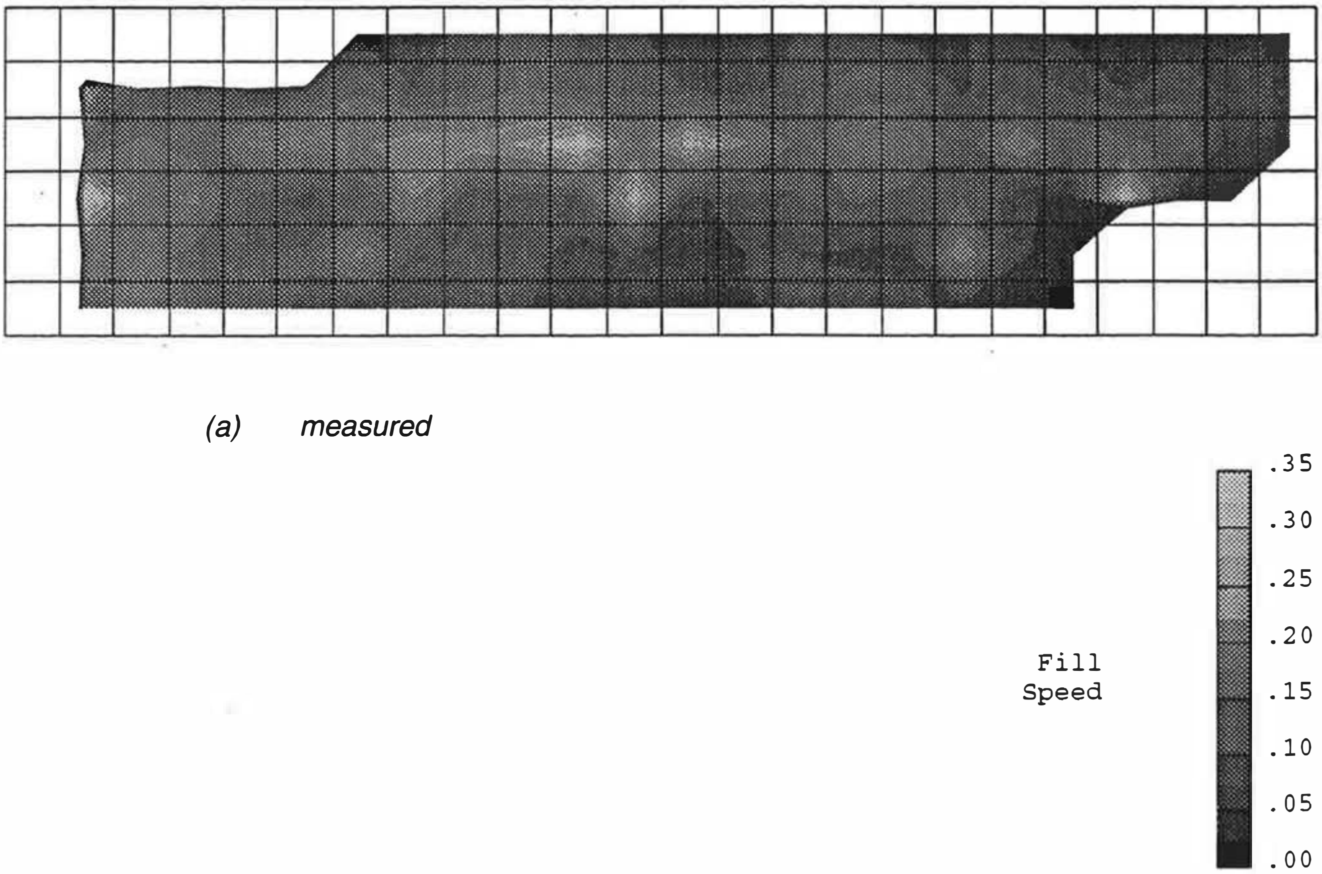


(a) measured

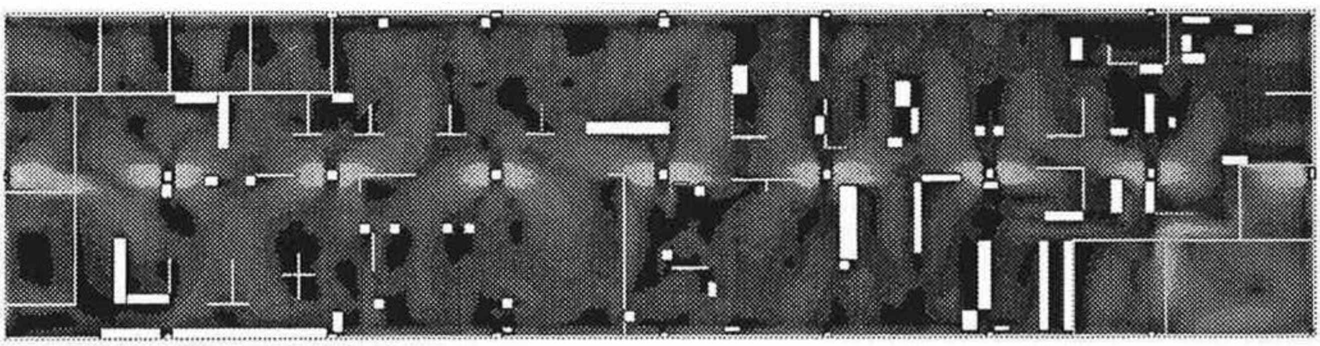


(b) predicted

Figure 5: Air temperature distribution at a height of 0.5 m from the floor:



(a) measured



(b) predicted

Figure 6: Air velocities at a height of 0.5 m from the floor

4.2 THERMAL COMFORT

Air velocities and temperatures measured at fixed heights and regular time intervals have been used to calculate thermal comfort indices and temperature gradients within the occupied zone of the sites surveyed.

For winter climatic conditions and indoor clothing levels of 0.5 clo and 0.9 clo (representing light and heavy clothing ensembles), the calculated values of PMV, PPD, ADPI, and temperature gradients for sedentary activity are presented in Table 3.

Site	0.5 clo		0.9 clo		ADPI (%)	Temp Gradient (seated)
	PMV	PPD (%)	PMV	PPD (%)		
1–1	−0.8	18	0.0	6	77	0.58
1–2	−1.1	31	−0.3	6	77	0.49
2–1	−0.8	20	0.0	6	56	2.42
2–2	−0.6	14	0.2	7	66	1.20
3–1	−0.4	10	0.3	7	81	0.38
3–2	−0.4	10	0.3	7	77	0.39
3–3	−0.4	9	0.3	8	82	0.21

Table 3: Comfort indices

4.3 AIR TEMPERATURE GRADIENT

The variations of room air temperature with height for all sites are shown in Figure 7. These values agree well with the generalised 'linear' profile[2]. Here, the extract and supply temperatures were calculated on the assumptions that the air temperature at 2.75 m above the floor (i.e. the ceiling height in site 1–1) is the same as the extract temperature, that temperature linearly increases from 1.85 m (i.e. the maximum measured level) to 2.75 m and the difference between the temperature of the air at 0.1 m above the floor and the supply air temperature is 0.4 times the overall exhaust-to-supply air temperature difference.

The variation of room air temperature with height for the measured and computed data for site 1–1 are compared in Figure 8. Good agreement is displayed, for the same assumptions mentioned above. However, the actual measured values for supply and extract air temperatures were higher and lower than these values respectively, resulting in a negative value for [T] near the floor. It is thought that this discrepancy may have arisen due to uncertainties in building air infiltration/exfiltration and thermal characteristics of the building envelope. Therefore, further parametric studies were carried out to investigate the influence of such factors.

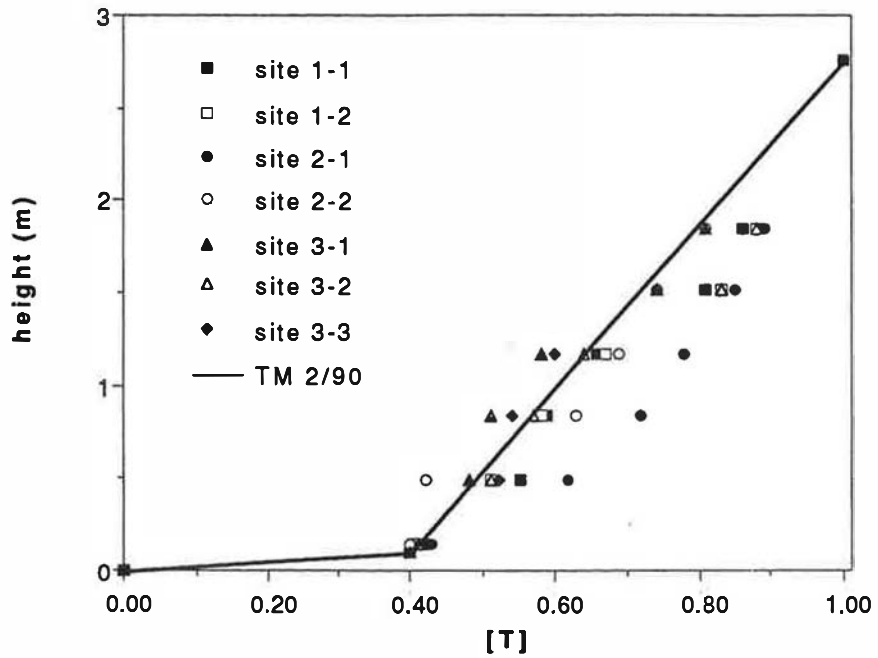


Figure 7: Variation of room air temperature with height for the measured data

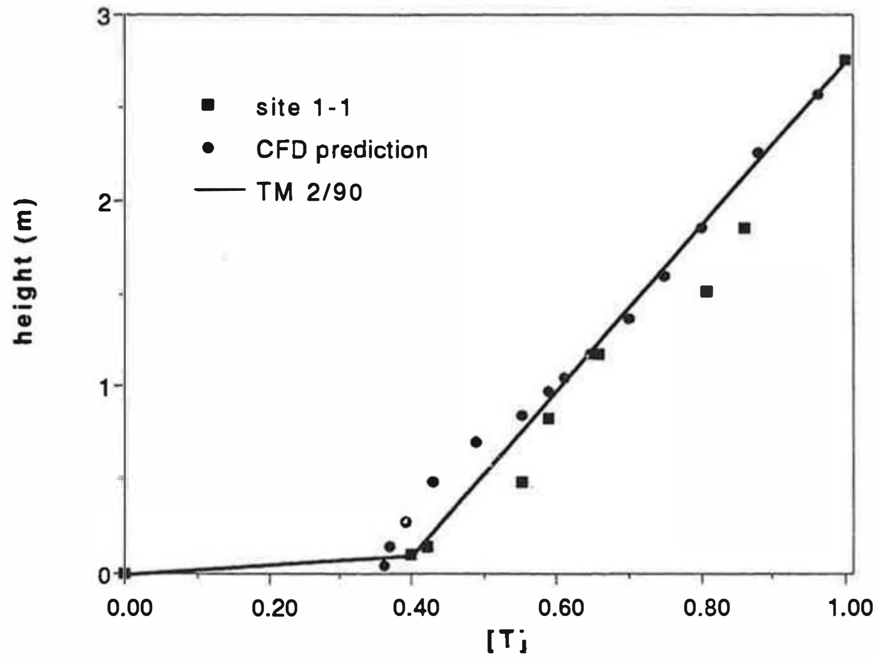


Figure 8: Variation of room air temperature with height for the measured and computed data – site 1–1

5. PARAMETRIC STUDIES

5.1 THE MODEL CONSIDERED

To study the effect of individual parameters, a simple computer model was defined. Geometrically, this was based on a central part of the space modelled in Section 4. It was set to include half an air supply unit and three extracts (Figure 9). The space was enclosed by a ceiling, floor, and far-wall, with the remaining three vertical surfaces taken to be planes of symmetry. The supply conditions were based on the thermal comfort design method, described in TM 2/90[2].

The simulation conditions were as follows:

- Supply air temperature 19.35°C
- Supply mass flow rate 0.096 kg/s
- Internal heat load 4 x 162 W

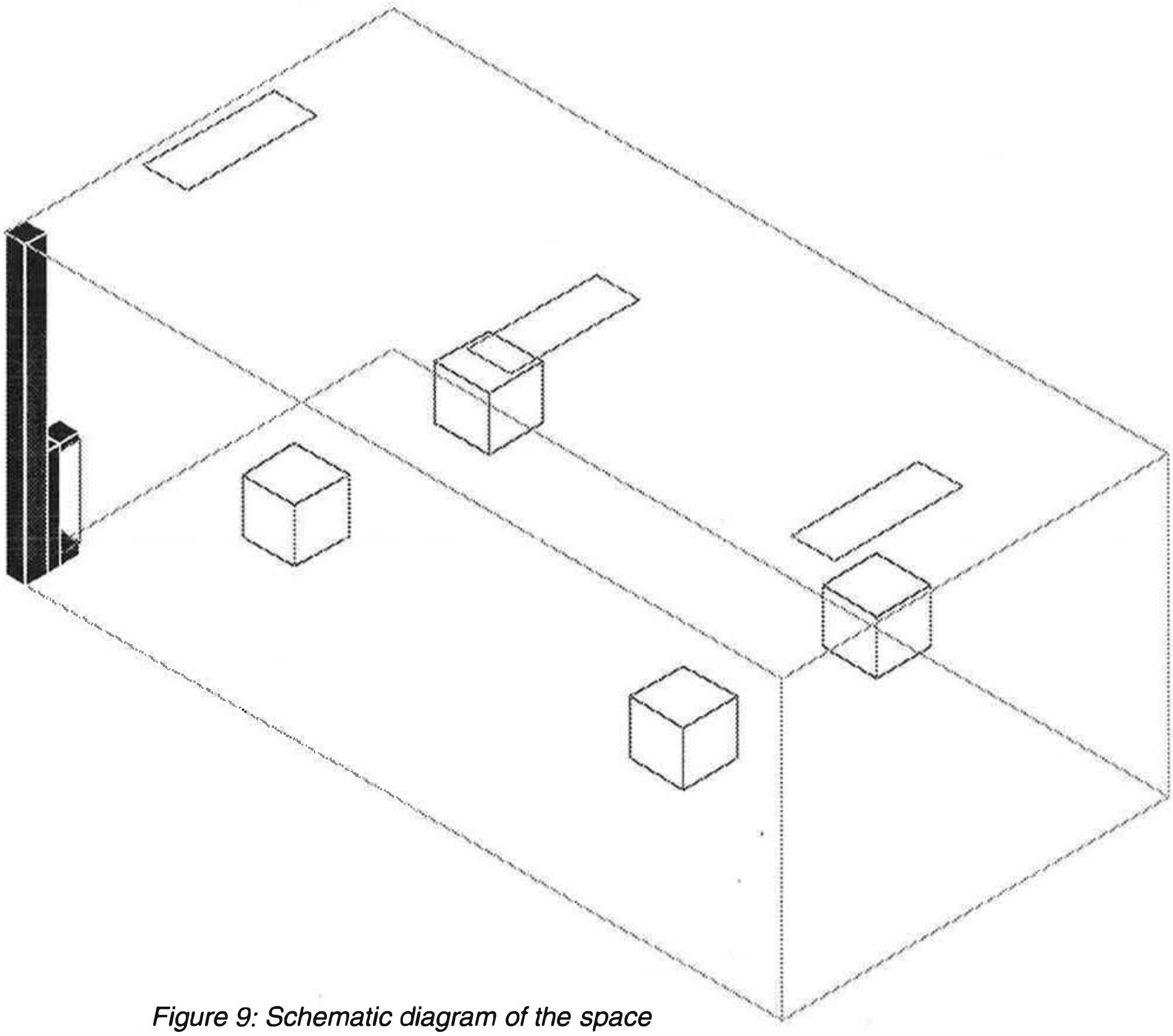


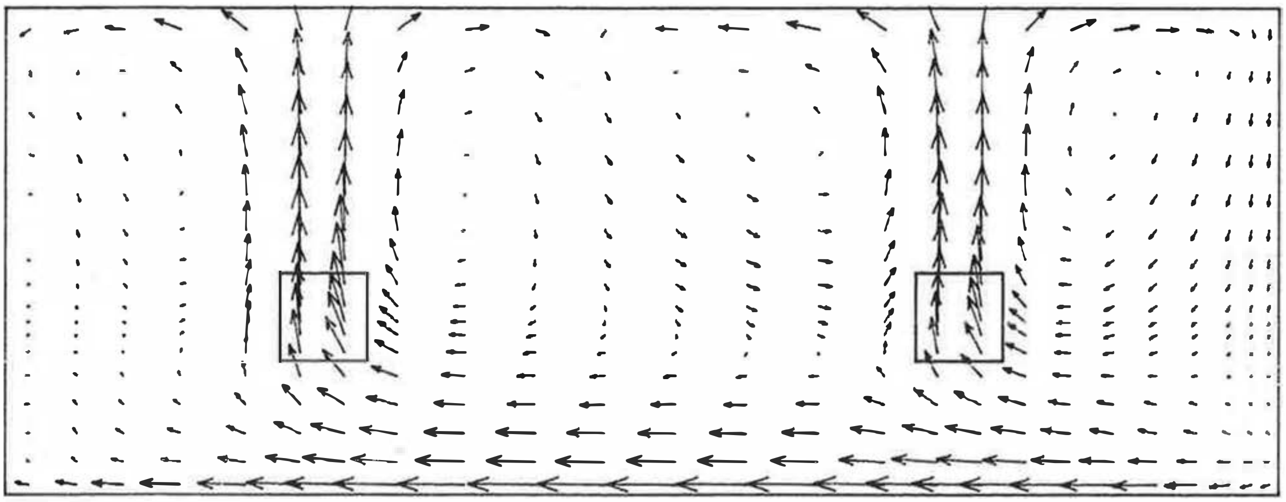
Figure 9: Schematic diagram of the space

To investigate the effects of the cold downdraughts and obstructions, the following simulations were carried out:

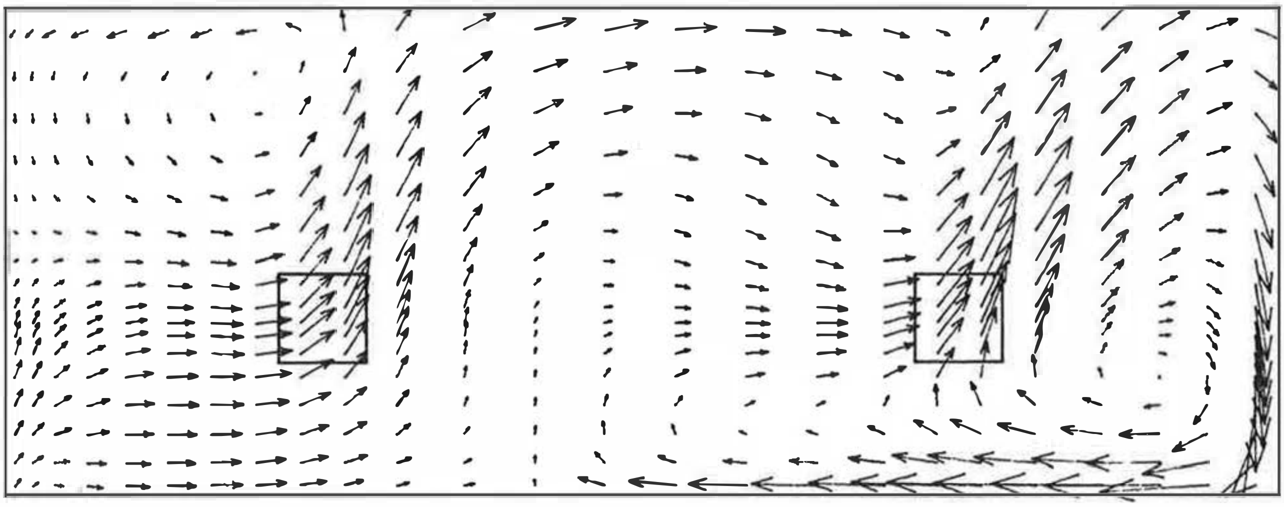
a)	No fabric effect	(isothermal walls)
b)	Cold wall and window:	external temperature = 0°C window area = 4.32 m^2 U_{wall} = 0.85 W/m^2K U_{window} = 3 W/m^2K
c)	Cold wall, window and infiltration	external temperature = 0°C window area = 4.32 m^2 U_{wall} = 0.85 W/m^2K U_{window} = 3 W/m^2K Infiltration = 0.008 kg/s at 0°C Exfiltration = 0.008 kg/s at 20°C
d)	Floor obstruction:	isothermal walls 100%, 90%, 80% and 70% free floor area (cuboid obstructions 0.75 m high)

5.2 FLOW FIELD

Velocity vector diagrams illustrating the flow patterns in an elevation passing through two of the thermal sources are shown in Figures 10 (a) and (b) (isothermal walls and cold wall plus air infiltration respectively). In Figure 10 (a), the computed flow pattern can be seen to be mainly influenced by buoyancy effects, due to the high temperatures of internal sources, confirming the satisfactory performance of the displacement ventilation system. The flow pattern in Figure 10 (b), however, indicates the strong influence of buoyancy effects due to the counteracting cold downdraughts induced by cold surfaces and infiltration. In particular, the downdraughts from these sources in the far-wall clearly negate the vigorous buoyant flow in the rest of the space. The flow pattern in this scenario is characterised by two mixed flow regions: one dominated by internal heat sources and the other by the cold downdraughts.



(a) Isothermal walls



(b) Cold wall and air infiltration

Figure 10: Flow field velocity vectors

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Ref Vector

5.3 VERTICAL TEMPERATURE GRADIENT

The effect of cold surfaces (ie external walls and windows) and infiltration/exfiltration on the room vertical temperature gradient is displayed in Figure 11. The predicted values denote that the temperature gradient in the room may be assumed to be linear as in TM 2/90[2]. However, two distinct gradients were noted: one from a point 0.1 m above the floor level to a height approximately at the centre of the heat sources (say 1.1 m from floor), and the other from this point to the ceiling. In addition, the [T] ratio at 0.1 m above the floor which was noted in TM 2/90 to be between 0.3 and 0.5, is dependent upon the thermal characteristics of the fabric and building tightness. For example, when average air temperatures near the floor are lower than the supply air temperature (as the result of cold downdraughts), the result will be a negative value for [T] (see Figure 11).

A comparison between the various levels of free floor area is shown in Figure 12, and indicates that the flow in the vicinity of the floor may be strongly influenced by obstructions (the obstructions were assumed to be solid cuboids 0.75 m high).

The influence of factors such as supply air flow rate, supply temperature and internal heat sources were also considered. However, no significant differences were observed when compared with the predicted data for the base case (for example, see Figure 13).

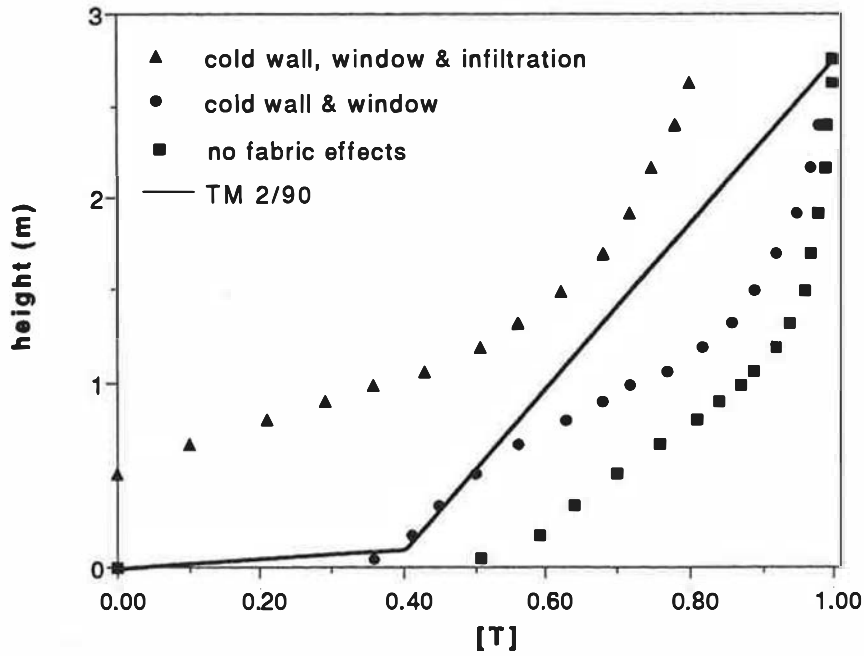

Figure 11: Variation of room air temperature with height: the effect of cold surfaces and infiltration

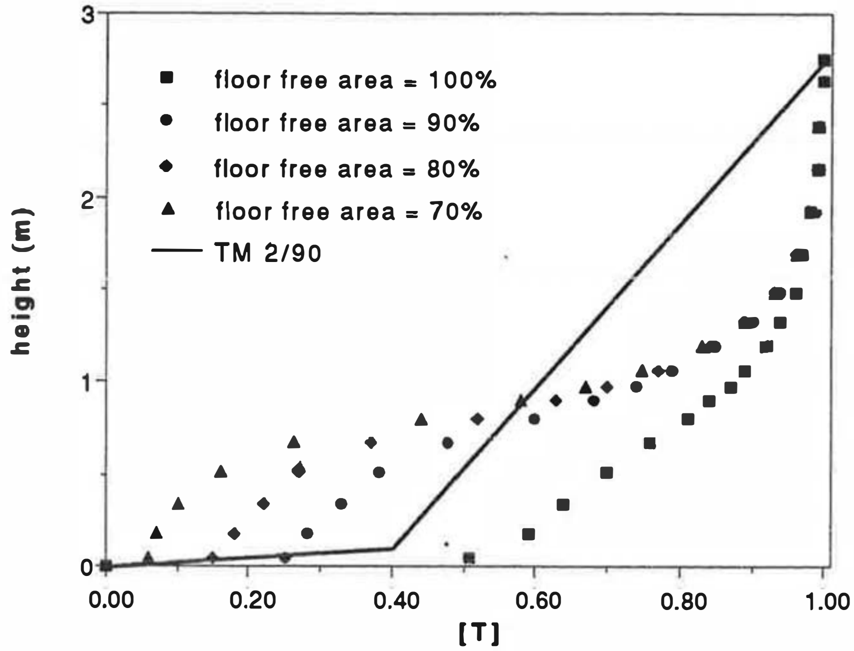

Figure 12: Variation of room air temperature with height: the effect of floor block obstructions

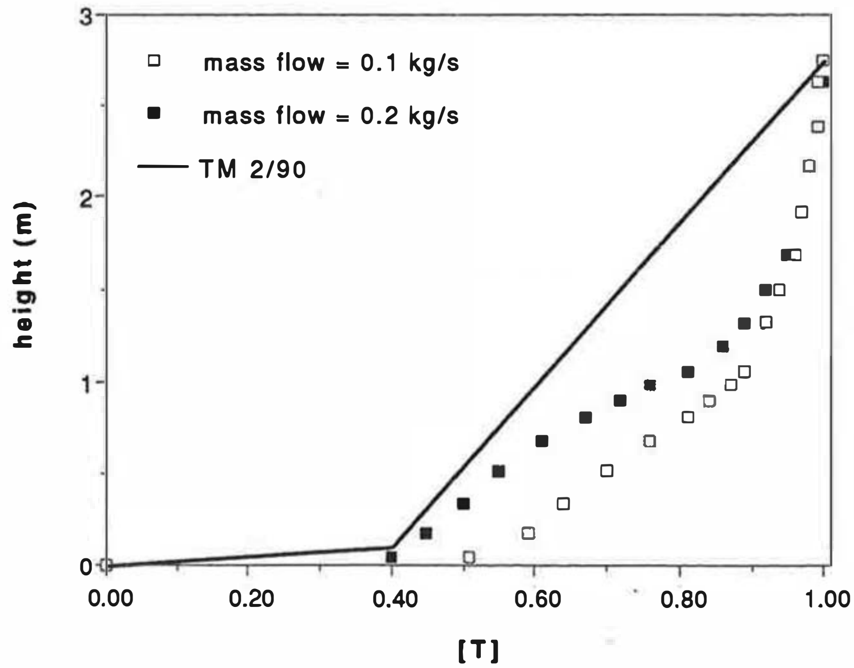

Figure 13: Variation of room air temperature with height: the effect of supply air flow rate

6. CONCLUSIONS

A combination of site measurements and computer predictions of field air velocity and temperature has been used to analyse the performance of displacement ventilation systems in office spaces. Based on thermal comfort procedure, the performance in three modern office spaces was assessed using thermal comfort indices, namely PMV, PPD and ADPI and vertical room air temperature gradient.

Good agreement was displayed for the variation of air room temperature with height when compared with the generalised 'linear' profile outlined in the earlier document[2]. Acceptable conditions were measured within two of the three sites considered in this study (i.e. sites 1 and 3). The uniformity within the spaces assessed by ADPI was found to be above 77%, which is better than typical conventional mixed systems. However, cold discomfort could be experienced by occupants wearing light indoor clothing during winter climatic conditions. This could be overcome by for example increasing the supply air temperature and/or wearing heavier clothing.

The measurements at one of the sites were used to verify the predicted data obtained by a CFD computer model. The model was subsequently used to study the effect of parameters such as cold surfaces, infiltration, floor obstructions and supply air flow rates.

The results of this study show that displacement ventilation systems are capable of providing the necessary thermal environment in office spaces. There is however a delicate balance between the interacting flow fields induced for example by secondary air motions resulting from cold surfaces and/or infiltration. These factors should be carefully considered during the design procedure and their likely impact minimised. The flow in the vicinity of the floor may be also influenced by furniture or other obstacles on the floor and hence create discomfort. These aspects can be assessed during design using microclimate predictive tools based computational fluid dynamics (CFD) techniques.

It is also evident that CFD models, such as FLOVENT used in this study, are capable of simulating the complex flow patterns generated within modern offices with high thermal loads, and can provide the necessary means by which the interactions of various factors may be evaluated to achieve satisfactory design solutions with confidence.

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