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AIR FLOW AND THERMAL COMFORT IN NATURALLY VENTILATED OFFICES

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SUMMARY

Experiments were carried out in a naturally ventilated office to measure the indoor environmental parameters such as air velocity, turbulence intensity and air temperature at several locations, each at three vertical levels. Air change rates for various indoor and outdoor climates were determined. Subjective assessments were made to evaluate the thermal comfort and indoor air quality in the office. The effect of opening windows and the door on the indoor comfort conditions was also investigated. In addition, numerical predictions of air flow and thermal comfort were performed using a computational fluid dynamics program.

Models were developed for assessing the indoor environment based on the field measurements. It was found that in real situations the occupant was more sensitive to the deviation of air temperature from the neutrality than predicted using Fanger's model. The office environment was found to be generally unsatisfactory. Recommendations are given for improving the indoor environment and reducing the heating costs.

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1. INTRODUCTION

A comfortable indoor environment is a necessity for the occupants' good health and high productivity. The indoor environment is a holistic phenomenon that involves synergy of thermal comfort, indoor air quality, other environmental factors such as the type of building and its psychological relevance for the occupants [1] and energy parameters.

There are some models available for assessing the thermal environment indoors such as the thermal comfort indices — Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfied (PPD) — developed by Fanger [2], which are based on the heat balance between the body and environment and subjective testing in an environmental chamber. However, these models may not be applicable to all the conditions encountered in practice. This is because laboratory subjects are not in their familiar working surroundings and because comfort depends not only on the quantifiable parameters used for formulating the available models but also on other factors which are difficult to quantify such as job satisfaction, stress, building characteristics and other environmental factors such as light and sound. Schiller, et al. [3], for example, found that optimum satisfaction with the thermal environment in office buildings was lower than that found under laboratory conditions and suggested that centralized, autonomous environmental systems have substantial inherent limitations in their effectiveness. Moreover, most laboratory based models are derived from measured data only to give an overall state of room environment without taking into account non-uniform reactions. For example there are differences in the sensitivity of different parts of the body to the surroundings especially at head and foot levels. Warm feet and cold head is preferable but many heating systems produce the opposite effect. Although there is some sophisticated models in which a human body is represented by up to 25 nodes [4], it is also based on the heat balance and is basically designed to calculate the local skin temperatures. Furthermore, most of the investigations on thermal comfort up till now have been carried out under steady state conditions such as those in laboratory tests or for short periods during field surveying. Results from such studies may not fully correspond to normal working conditions especially in naturally ventilated offices because the indoor thermal environment is essentially transient due to the changing climate outdoors, varying occupants' activities indoors and variations in the heating and ventilation systems output.

The objective of the present work is to evaluate the indoor environment in naturally ventilated offices for long durations with detailed measurements of the environmental parameters and to develop models for assessing the indoor environment based on the field measurements.

1. METHOD

This investigation has been carried out by means of physical measurement, subjective assessment and numerical prediction of the indoor environment in a naturally ventilated office room over a period of four months in the winter of 1991/92. The office is situated in the north wing of the third level of the FURS building at the University of Reading. It has interior dimensions of 5.4 X 2.3 X 2.6 m (length X width X ceiling height). The effective volume of the room, i.e., the volume excluding the space occupied by obstacles, is approximately 29.3 m^3. The room is built of one concrete external wall and three concrete brick walls connected to other rooms. The floor is made of prefabricated concrete (carpeted) and the ceiling comprises hardboard layers under the prefabricated concrete roof. The room is connected to the corridor via a hinged wooden door. There are two weatherstripped double-hung aluminium frame windows in the north face. The office is normally occupied by one person and is heated by two small hot water radiators in cold seasons; an extra electric heater was provided when needed for the experiments. A schematic diagram of the room is shown in Figure 1.

2.1 Physical measurements

During an experimental test the air velocity, standard deviation, turbulence intensity and air temperature were measured continuously at six floor locations using thermal anemometers (DANTEC Multi-channel Flow Analyser type 54N10). At each location measurements were taken at points 0.1 m (foot/ankle level), 0.6 m (back of a seated person) and 1.1 m (head/neck level of a seated person) above the floor in a vertical line. The plane radiant temperature, temperature of room surfaces and obstacles and indoor air humidity were measured using an indoor climate analyser (Bruel & Kjaer type 1213). Thermal comfort indices (PMV and PPD) were measured using a comfort meter (Bruel & Kjaer type 1212). A CO_2 gas analyser was used for the measurement of indoor CO_2 concentrations.

The air change rate for each test was determined using the concentration decay method with an infra-red gas analyser. A portable fan was employed to ensure a good mixing of tracer gas (isobutane) and air in the room for a few minutes after injecting the gas. The wind speed was measured with three vane cup anemometers and the wind direction with a wind anemometer mounted on the top of the building (about 5 m above the roof). The

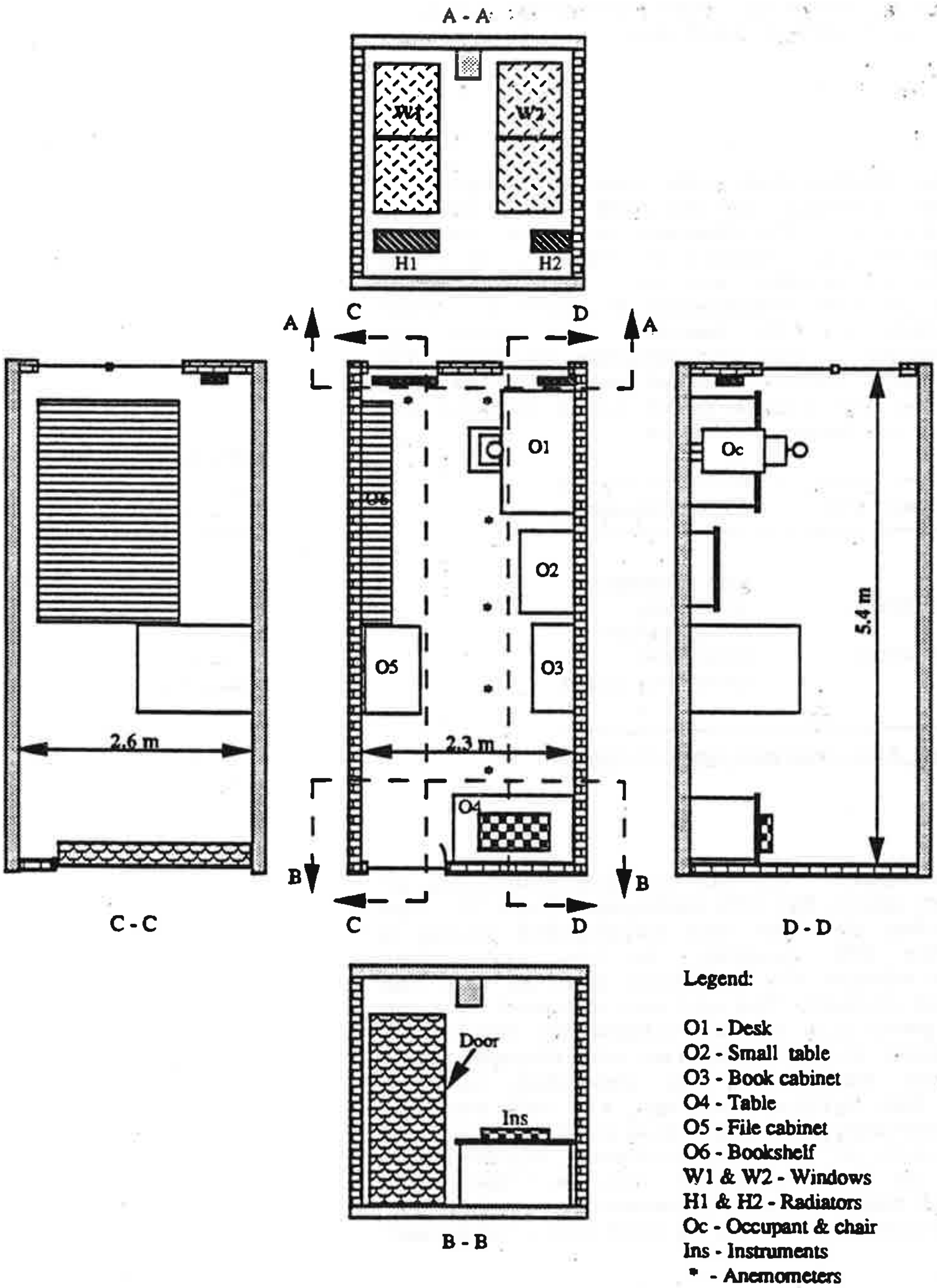

Fig. 1 Schematic of the test room

outdoor air temperature and humidity were measured using a copper-constantan thermocouple and a hand-held humidity meter respectively.

2.2 Subjective assessment

A subjective assessment was undertaken simultaneously with the physical measurements. The thermal environment was assessed according to the occupant's vote on the thermal sensation and air movement in the office under various outdoor or indoor conditions and different arrangements of window and door openings. The assessment was made based on the judgements at head and foot levels as well as for overall comfort. Besides, the indoor air quality was assessed according to the impressions of odour and freshness of air. A seven-point thermal sensation scale was used to evaluate thermal sensation and a five-point scale to rate the impressions of air movement as shown in Table 1.

Rating	Thermal sensation	Air movement
-3	cold	
-2	cool	too draughty
-1	slightly cool	draughty
0	neutral	acceptable
1	slightly warm	stagnant
2	warm	very stagnant
3	hot	

Table 1. Rating scales for thermal environment

2.3 Numerical prediction

Numerical predictions were carried out for the distributions of air flow and thermal comfort using the CFD technique [5,6,7]. The mean radiant temperature for the PMV was calculated using a radiation model within the CFD program. In the prediction presented in this paper, a window was slightly open at the top (0.1 m) whereas the door was closed. The air was assumed flowing into the room through the partially opened window and the inlet velocity was calculated based on the measured air change rate. The inlet temperature was taken as the readings of the anemometers positioned at the window opening. Air was assumed flowing out of the room through the gap between the door and floor. The initial temperatures of the room surfaces, windows and obstacles were taken to be the measured values. The heat generation sources included the occupant (assumed to be 100 W), instruments and radiators which were taken as obstacles with heat generation.

3. RESULTS AND DISCUSSION

In all 46 tests were performed. The results are discussed in three parts — environmental parameters, subjective evaluation and numerical prediction.

3.1 Environmental parameters

This includes all the measured results for air change rate and other parameters concerning the room environment.

3.1.1 Air change rate

The air change rates were determined for different arrangements of window and door openings. The air change rate for the windows and door closed is related to the wind speed and indoor-outdoor temperature difference by the following equation:

$$N^2 = 0.0393\ V_w^2 + 0.0154\ \Delta T \qquad \text{(adjusted } r = 0.98) \qquad (1)$$

where N = air change rate, h^{-1}
V_w = wind speed, m/s
ΔT = indoor-outdoor temperature difference, K.

The regression has a confidence level of almost 100%. The wind speed ranged from 0.2 to 10.0 m/s and the range of the indoor-outdoor temperature difference was between 9.7K and 20.4K.

The air change rate for a window and/or the door partly open is correlated as

$$N^2 = [a_1 + a_2|\sin(90 - \theta/2)|](V_w A)^2 + b\ \Delta T + c \qquad (2)$$

where θ = wind direction, degree from north clockwise
A = opening area of window (A_w) and/or door (A_d), m^2.

The calculation of the area A, the values of the constants a_1, a_2, b and c and the adjusted correlation coefficient r are shown in Table 2.

Window/door arrangement	A	a_1	a_2	b	c	r
Window open only	A_w	388	-435	15	3	0.94
Window & door open	$A_w A_d / \surd(A_w^2 + A_d^2)$	60059	-61481	103	0	1.00
Door open only	A_d	0	0	0	15	

Table 2. Opening area and constants for Equation (2)

The confidence level of the correlation for a window partly open is 99.5%. The confidence level for both a window and the door partly open is low (90%) due to insufficient data points (four values only) and therefore the correlation should be used with caution. The air change rates for the cases when only the door was partly open could not be satisfactorily correlated with the outdoor environmental parameters. It appears that for this arrangement of window/door opening the air change rate was influenced more by the conditions in the corridor than by the outdoor environment. The constant in Table 2 for this arrangement was calculated from the mean value of the measured air change rates in order to fit the form of the correlation. Since opening the door only or opening both the window and door in this office was not a normal practice in winter, only a few tests were performed for comparison. The window opening area for the tests performed ranges from 0.036 m^2 to 0.194 m^2. The level of door opening is between slight (0.24 m^2) and half (1.20 m^2). The air change rates for the opening areas beyond these ranges need further exploration particularly in warm seasons.

Figure 2 shows a comparison between the measured air change rates and those predicted using Equations (1) and (2). When all the windows and door were closed the air change rates ranged from about 0.44 h^{-1} for a mild and still outdoor climate to 1.94 h^{-1} for a very windy day with a mean of 0.86 h^{-1} (or 7 l/s), which is slightly lower than the minimum fresh air requirement to maintain a CO_2 concentration maximum limit of 1000 ppm. When a window and/or the door were partly open, the air change rate increased dramatically depending on the opening size, the wind direction, the wind speed and the indoor-outdoor temperature difference. The air change rates under the conditions investigated ranged from 1.51 to 5.88 h^{-1} for a window partly open, 3.40 to 10.87 h^{-1} for both the window and door partly open and 2.56 to 4.79 h^{-1} for only the door partly open.

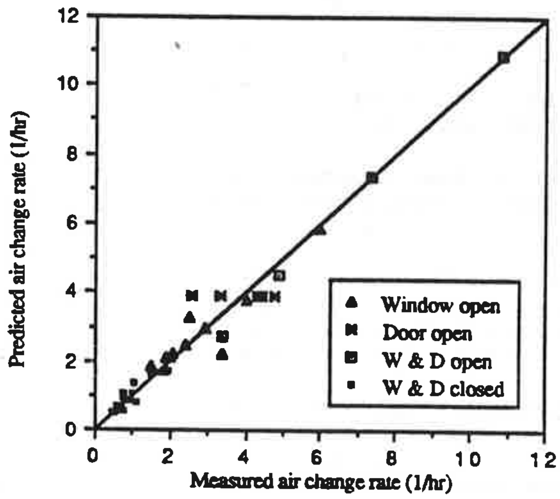

Fig. 2 Scattergram of the measured against predicted air change rate

3.1.2 Room environment

The physical data for the room environment were obtained for every test. The mean air velocity in the room was very low with an average of about 0.05 m/s when the windows and door were closed. When a window and/or the door were partly open, the velocity increased but not very much, with an average value still being less than 0.15 m/s.

The turbulence intensity for most of the tests was between low and moderate with a mean of 22.5% for the windows and door closed. When the window and/or door were open, the mean of turbulence intensity was increased to 42.0%. According to Melikov, et al. [8] the magnitude of turbulence intensity increases with the decrease in the mean air velocity. However, a regression analysis indicates that the correlation between the turbulence intensity and mean air velocity is insignificant especially for the data at foot level. The turbulence intensity appears better to be correlated to the air change rate. The best fit of the correlation is

$$Tu = 25.76\ N^{0.42} \qquad (r = 0.68) \tag{3}$$

Where Tu is the turbulence intensity in percentage.

The indoor air temperature changed from day to day during the course of measurement, ranging from 17.8°C to 26.2°C with a mean of 22.4°C because of the fluctuations of outdoor temperature ranging from -0.2°C to 13.6°C, air change rate due to opening the window or door and heat loss or gain from the room. Temperatures above 25.5°C resulted from the heat provided by the personal electric heater which was used when a window alone, or together with the door, was partially open to compensate for the ventilation heat loss. The air temperature at head level was found to be higher than that at foot level with a mean vertical temperature difference of 1.6K. A large temperature stratification was observed in some of the tests with the vertical temperature difference as high as 3.6K which is greater than the ISO limit for comfort 3K (the vertical air temperature difference between 1.1 m and 0.1 m above the floor) [9].

The room surface temperatures were usually lower than the mean air temperature especially for the north wall which was directly exposed to the cold ambient. The measured plane radiant temperature, and thereby the calculated mean radiant temperature, were also lower than the mean air temperature. In some cases where both the window and door were opened the air temperature was lower than the radiant temperature due to a large influx of cold air. The average difference between the mean air temperature and mean radiant temperature for all the tests was 0.6K.

The relative humidity in the room throughout the test period was normally within the accepted comfort limits, ranging from 40% to 55% with a mean of 46%. In some occasions it dropped to slightly below 40%, the lower limit for comfort, but no discomfort due to this was observed.

The following table summarises the distributions of the room environmental data measured with the thermal anemometers.

	Mean air velocity			Turbulence intensity			Mean air temp.		
	Head	Foot	Overall	Head	Foot	Overall	Head	Foot	Overall
Min	0.038	0.041	0.042	16.2	7.7	14.0	20.1	17.8	19.7
Max	0.113	0.136	0.115	79.1	63.9	68.2	26.2	24.0	24.9
Mean	0.059	0.064	0.060	39.4	28.7	34.7	23.1	21.4	22.4
S.d.	0.017	0.023	0.017	18.9	17.2	18.3	1.4	1.5	1.4

Table 3. Distribution of room environment

3.2 Subjective evaluation

Out of 46 tests 44 subjective measurements were collected.

3.2.1 Thermal sensation

When the windows and door were closed the mean thermal sensation was on the warm side of neutral. When the window and/or door were open, the votes were scattered widely over the thermal sensation scale, with votes for the cool side being roughly the same as those for the warm side. However, the measured PMV values, which were obtained from Fanger's comfort equation, were close to the neutral point for about 80% of the tests. This suggests that in the present investigation Fanger's equation under-estimates the thermal impressions for the cases when the windows and door were shut and under-values the swings of the impressions for these and other cases. This may be due to three main reasons. One is the assumption of steady state laboratory conditions used in the derivation of Fanger's equation. Another is the over-simplification of the metabolic rate of the occupant. The occupant rarely sat in the room for a long period, say one hour, without moving around or engaging in other activities such as teaching. The metabolic rate was however taken as constant (1.2 met) in the calculation of PMV due to the difficulty in determining its true value. The third reason is the sensitivity of PMV to clo values. In a laboratory test the clo values are consistent whereas in field tests the clothing levels vary with occupants and time as a suit may not be worn daily.

The thermal sensation was in general dependent on the room air temperature and velocity. The regression equations for the thermal sensation scale (TS) at head level, foot level and overall for the room against mean air temperature (T in °C) and velocity (V in m/s) are respectively

head $$TS = 0.5732\ T - 11.97\ \sqrt[4]{V} - 6.93 \qquad (r = 0.66) \qquad (4)$$

foot $$TS = 0.5624\ T - 7.53\ \sqrt[4]{V} - 8.28 \qquad (r = 0.63) \qquad (5)$$

overall $$TS = 0.6146\ T - 12.27\ \sqrt[4]{V} - 7.46 \qquad (r = 0.68) \qquad (6)$$

In Figure 3 the occupant's thermal sensation responses are presented as a function of mean air temperature, using a mean air velocity of 0.06 m/s for Equations (4), (5) and (6). The PMV line predicted from Fanger's equation is also presented for comparison (assuming a metabolic rate of 1.2 met and a clo value of 0.8). From the above equations or the corresponding curves in Figure 3 the neutral temperatures i.e. T for TS = 0, are found to be 22.4°C, 21.4°C and 22.0°C for the head level, foot level and overall for the room respectively. The neutral temperature predicted from Fanger's comfort equation (air temperature for PMV equal to zero) is 22.8°C. Thus Fanger's equation overpredicts the neutrality by 0.4K, 1.4K and 0.8K for the head level, foot level and overall for the room respectively due to various reasons mentioned above, which seems to confirm the findings by Schiller, et al. [3]. A more important point is that from the present investigation the correlated curves in Figure 3 are steeper than those given by Fanger's equation, suggesting the occupant is more sensitive to changes in air temperature. This fact was also observed by Fishman and Pimbert [10] whose field study showed that the steepness of the slope of the curve from the observations deviated from Fanger's equation particularly at temperatures above 24°C. In addition they also found that Fanger's comfort equation predicted the neutral temperature 0.6K higher than that from the field survey, which was attributed to the incorrect estimation of the subjects clothing. The deviation appears to be more at foot level than at head level.

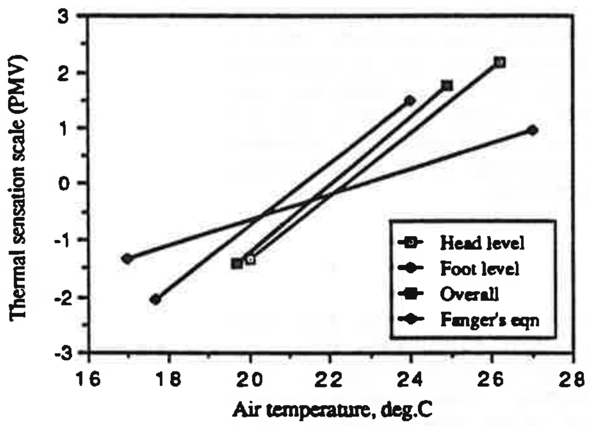

Fig. 3 Effect of air temperature on thermal sensation responses

It is also noted that the neutral temperature for the head level is 1.0K higher than that for the foot level. This seems to disagree with the common belief concerning the comfort requirement of warm feet and cold head. The reason for this disagreement may be the variation in radiation distribution. Because cold windows are 0.74 m above the floor in the north face of the room and the radiators are below the window level, the radiant temperature at head level would be lower than that at foot level whereas the measured air temperature at head level was higher than that at foot level due to stratification. The head would thus have lost more heat due to radiation but less due to convection than the feet. However, the radiation has not been incorporated into the above correlations for thermal sensation as the radiant temperature was measured in the middle of the room but not at head and foot levels. Hence, the effect of radiation on the occupant was not evaluated for local comfort prediction. Another reason for the disagreement might be the adaptation of the occupant to the neutrality, i.e. the occupant adjustment to the surrounding temperature. In this case the subject concerned is the normal sole occupant of the room and could have been accustomed to his usual environment and hence tolerated a slight vertical temperature difference. As a result, most of the thermal sensation votes gave the same rating at head and foot levels despite that there was always a positive vertical temperature difference.

If according to Fanger's definition [2] the central three categories of the thermal sensation scale (-1, 0, +1) were regarded as an indication of an acceptable state for thermal comfort whereas the votes outside these central categories were considered to represent dissatisfaction with the thermal state, the results suggest that about one third of the responses represented dissatisfaction with the thermal environment whether for head, or foot or overall impressions. Most of the dissatisfaction that occurred when the windows and door were closed was caused by overheating, which could be avoided simply by controlling the heat output from the emitters if a thermostat was available or by window opening. On the other hand, because the overall votes were on the warm side and the amount of heat supply could not be decreased in mild climates the heating costs could be reduced with the help of a thermostat or a weather compensated heating system. A great majority of the votes on the cool side occurred when a window was open either alone, or in combination with the door. In practical situations the window would be closed or the size of the window opening or a ventilator would not be so large when it was cold outside.

3.2.2 Air movement

The overall impression of the air movement in the room for the cases when the windows and door were closed was on the side of being stagnant. When a window and/or door were partly open, the impression shifted to being slightly draughty. The measurements showed that there was little air movement when the windows and door were closed. Even when a window and/or the door were partly open the mean air velocity at the measured points were still below 0.15 m/s. When draught was detected the thermal sensation was rated as cold especially at foot level. This implies that low temperature was the main cause of the draught.

The ratings of the air movement (AM) are associated with the air temperature, velocity and turbulence intensity as follows:

$$AM = 0.1462\ T - 20.31\ V - 0.0048\ Tu - 1.71 \quad (r = 0.57) \quad (7)$$

head level;

$$AM = 0.2037\ T - 6.65\ V - 0.0081\ Tu - 3.64 \quad (r = 0.44) \quad (8)$$

foot level;

$$AM = 0.1455\ T - 18.99\ V - 0.0069\ Tu - 1.69 \quad (r = 0.56) \quad (9)$$

for the room as a whole.

The above equations indicate that the draught risk increases (i.e. AM decreases) with an increase of air velocity and turbulence intensity but with a decrease in air temperature. A "comfortable" temperature for air movement, defined as "the air temperature for the rating of air movement to be acceptable", can be obtained from these equations for a given air velocity and turbulence intensity. By substituting the mean values of velocity and turbulence intensity for the test conditions (V = 0.06 m/s;

Tu = 34.7%) a comfort temperature of 21.1°C is calculated for head level, foot level and overall judgement, which is approximately equal to the neutral temperature at foot level and is 1°C lower than that at head level. The inference is that when the room environment is comfortable in terms of warmth at foot level, it is also acceptable for air movement and if the thermal sensation is comfortable at head level but slightly warm at foot level the occupant will feel slightly stuffy. Therefore sometimes a compromise between the requirements for warmth and air movement may have to be made to achieve an acceptable thermal condition.

Equations (7) and (9) also indicate that the overall impression of air movement is similar to that felt at head level, i.e. when the head feels stagnant the overall response of the air movement will be stagnation. This is also true for draught. Moreover, these two equations show that an increase in mean velocity of about 0.05 m/s can change air movement judgement, say, from being slightly stagnant to acceptable at head level or overall judgement. Since most of the votes were slightly stagnant for air movement and slightly warm for thermal sensation when the windows and door were closed, to increase the velocity from 0.05 m/s to 0.10 m/s would give a more pleasant thermal environment for the office. In these tests the feet were more sensitive to air temperature and less sensitive to air velocity than the head. The votes on stagnant air for the foot level are fewer than those for the head level, therefore less or no increment in the velocity is necessary to attain a comfortable condition. The effect of turbulence intensity on the air movement is marginal compared with air velocity or temperature.

Fanger, et al. [11] derived the following equation for the calculation of the percentage of dissatisfied due to draught, which was based on laboratory tests:

$$PD = (3.143 + 0.3696\ V\ Tu)(34 - T)(V - 0.05)^{0.6223} \qquad (10)$$

According to this model the draught risk for all but one tests was found to be negligible as the calculated percentage of dissatisfied using Equation (10) and the measured mean air velocity, turbulence intensity and mean air temperature is within the 10% draught risk criterion. The only exception was the one when a window was open at the largest setting for the tests on a cold day which led to an indoor air velocity over 0.10 m/s and temperature around 20.0°C. Again, the laboratory model fails to fully predict the comfort in practice because it under-estimates the effect of air velocity. Equation (10) indicates that the draught risk is small at a velocity close to 0.05 m/s whatever the magnitude of air temperature or turbulence intensity is. In reality at a low indoor temperature air close to the exposed parts of the warm human body would form a free convection current as a result of thermal buoyancy such that the velocity of air flowing over the head of a standing subject could reach 0.3 m/s [12]. Using the air temperature and velocity near the body, Equation (10) might show the presence of draught. However, the model equation was derived on the basis of the measurements taken at such a distance away from the body that the temperature and velocity were undisturbed by free convection currents. Therefore it may be inferred that the model is not reliable for the circumstances where both air temperature and velocity are lower than those recommended for thermal comfort. This model also fails to take into account the need for high velocities in densely occupied spaces and also where humidity may be high.

3.3. Numerical prediction

The predicted temperatures and velocities at a vertical section were compared with the measured values as shown in Figure 4. The prediction was in good agreement with the measurements for most of the points. The measurement however showed a more uniform distributions of the velocity and temperature than those predicted. This may be accounted for by the simplification of supply air. The window opening could function as an air exit as well as an inlet due to the fluctuation of the outdoor climate and hence the actual downdraught near the window may be smaller than predicted (see Figure 5).

Figure 5 shows the predicted airflow patterns and distribution of the thermal comfort index (PMV based on Fanger's equation [2]). It can be seen that the supply air jet falls downwards along the window due to the large buoyancy force (the Archimedes number to be as high as 1.09). The room air in the occupied area is thermally comfortable (between slightly cool and neutral) according to the prediction. However, it is cold in the area near the supply jet and warm close to the heat sources, one of which is illustrated in Figure 5(d). During this test the occupant rated the thermal sensation in the room as neutral.

Overall this and previous predictions [5, 6, 7] have demonstrated the capability of the CFD program for obtaining detailed distributions of indoor environment, giving accurate boundary conditions.

4. CONCLUSIONS

The present investigation suggests that further validations of the thermal models based on laboratory tests at steady state conditions are still needed in practical thermal environments where the climate conditions are transient and where the occupants invariably change their activities especially beyond the accepted comfort zone. For the cases investigated Fanger's equation for thermal comfort overpredicts the neutrality and under-predicts the comfort requirement when the air temperature deviates from neutrality. The equation for draught risk also fails to predict the response to draught.

The air change rates in the room are related to the indoor and outdoor climates by Equations (1) and (2). The turbulence intensity is a function of the air change rate as given by Equation (3). Models for evaluating the thermal sensation and air movement have also been developed for an office.

To achieve a good indoor climate and air quality, it is necessary to supply fresh air either by opening windows or by installing a suitable vent for the introduction of fresh air. The size of the vent opening should ideally be controllable, either manually or by a thermostat.

Further investigations are underway to evaluate the comfort in this and other offices in order to explore the effects of individuals, climate and room use on the comfort requirements.

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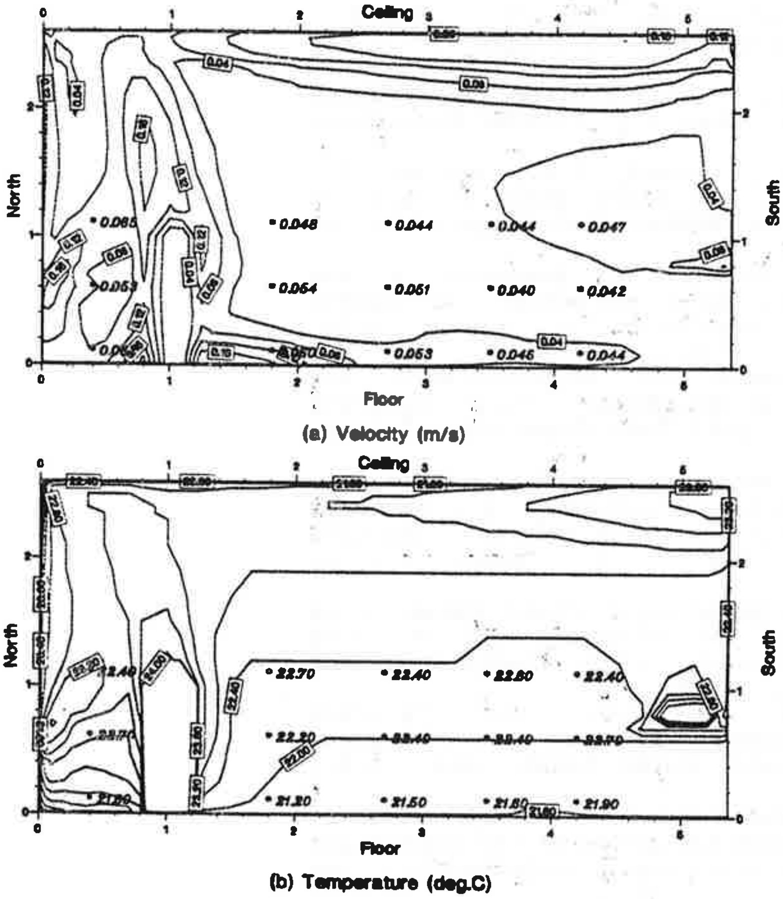

Inlet air: V = 0.24 m/s ; T = 15.0 deg.C

Fig. 4 Comparison of predicted isovels and isotherms with measured velocities and temperatures (•) on a vertical plane (0.9 m from the east wall) of the room with occupancy

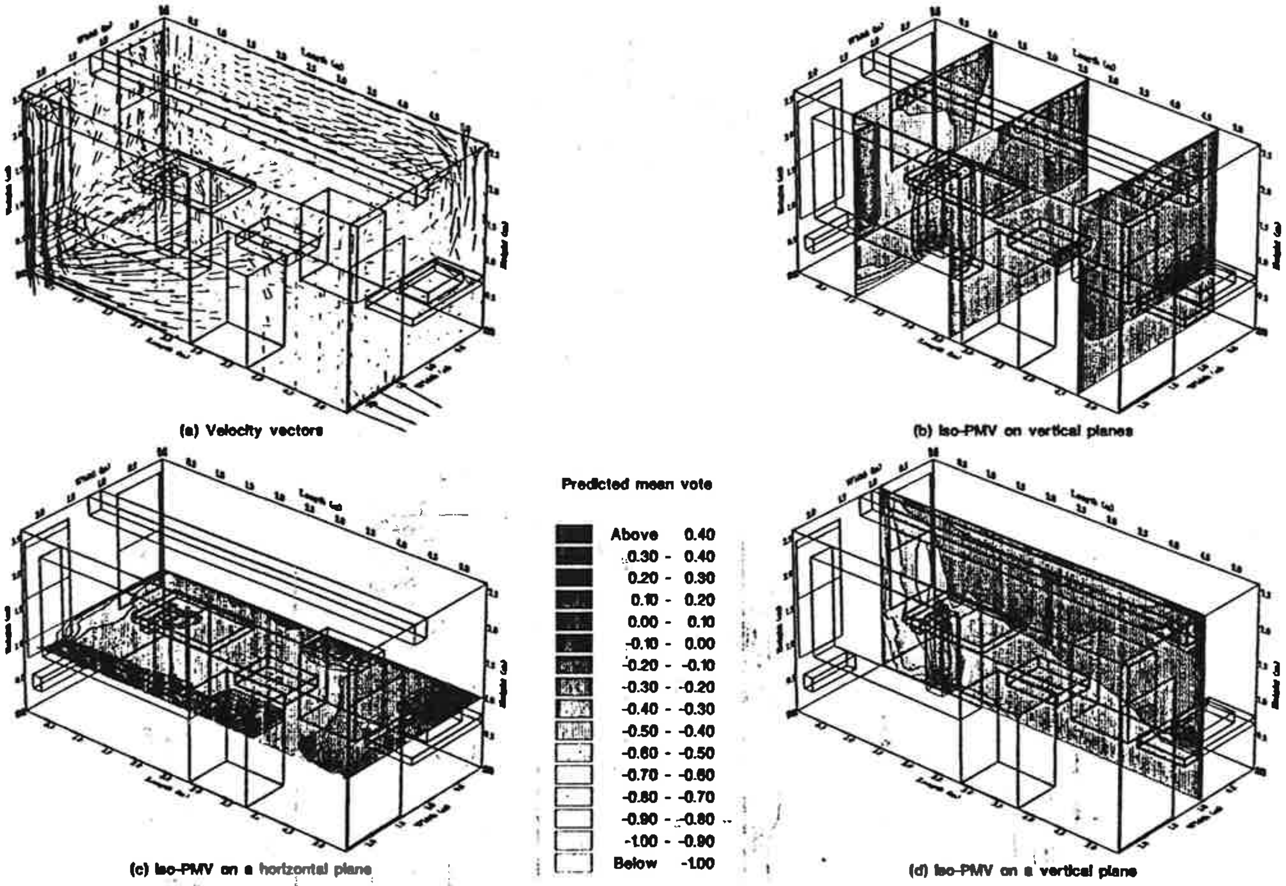

Fig. 5 Predicted velocity vectors and contours of predicted mean vote (iso-PMV) in the room with occupancy