INDOOR ENVIRONMENT IN NATURALLY VENTILATED OFFICE ROOMS

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ABSTRACT

The paper deals with the thermal environment and air quality aspects in naturally-ventilated office rooms. Measurement of environmental parameters, such as air velocity, turbulence intensity, air temperature, air change rate, carbon dioxide concentration, etc. were carried out over a period of 14 months in five naturally-ventilated office rooms. In addition, subjective assessments were made to evaluate the thermal and odour sensation of the occupants, during normal occupancy.

It is generally concluded that the thermal sensation in a work environment differs from that experienced under steady-state conditions, such as those present in laboratory-based studies. In addition, the neutral temperatures recorded were lower than those required under steady-state conditions. The potential for energy saving during the heating season for the climate is also discussed in the paper.

1. INTRODUCTION

Thermal comfort is an important factor that influences occupants' satisfaction with the room environment. The relationship between thermal comfort and environmental parameters has long been subjected to investigation. Fanger (1) has developed a set of the most comprehensive models to date for the prediction of indoor thermal comfort based on laboratory testing. However, a number of field studies (2), (3) have shown that these models could not accurately predict the occupants' thermal responses in the workplace. Besides, these laboratory based models describe the overall state of the human body but not the sensitivity of different parts of the body to the surroundings.

Air quality in offices has been a major concern in recent years, particularly due to increasing numbers of reports on the sick building syndrome. Odour intensity is one of the indicators of indoor air quality and is often associated with the level of carbon dioxide. The results of indoor CO_2 measurements are used to specify minimum fresh air requirements. However, poor air distribution in a space can result in occupants' dissatisfaction with the indoor air quality even if the ventilation rate is higher than the minimum requirement.

To provide comfort in buildings currently requires almost 40% of world non-renewable energy. Besides increasing energy expenditure, fossil fuel burning substantially contributes to global warming. A comfortable indoor environment with minimum energy use can be achieved through careful design and management of the building and the associated environmental systems as well as making use of improved building technology such as good thermal insulation, reduced air infiltration, employment of heat recovery systems, utilisation of renewable energy and the use of building energy control systems. In winter, energy consumption in offices may be reduced by lowering the indoor air temperature, hence reducing heating costs. More recent field studies (2), (3) have shown that optimum satisfaction with the thermal environment in office buildings can be achieved at a lower temperature than that obtained under laboratory based conditions, which is the basis of ISO 7730 (4).

In this paper, the thermal environment and the air quality in naturally-ventilated office rooms are studied. The thermal environment in the rooms is compared with predictions using Fanger's thermal comfort criterion. Generally, thermal comfort was achieved at a lower room temperature than that predicted by Fanger and as a result the potential of energy saving in winter is investigated in the paper.

2. METHOD

This investigation has been carried out by means of physical measurements combined with a subjective assessment of the indoor environment in five naturally ventilated office rooms (denoted as rooms A, B, C, D and E). The offices are situated in the FURS building at the University of Reading. Rooms A, B, C, and D are staff offices and room E accommodates several computer workstations. Rooms A and B which are situated in the north wing of the building are built of one concrete external wall and three concrete brick walls connected to other rooms. Each of the two rooms are connected to the north corridor via hinged wooden doors. There are two small and one large weatherstripped double-hung aluminium frame windows in the north facade of room A and room B respectively. Room C is located between south and north corridors which connects the south and north wings. The walls separating the room and the corridors are glazed while the other walls are made of concrete bricks. There is a small axial fan in the north facade near the ceiling for supplying air into the room. Room D and room E have similar structures to room A and room B respectively but are both situated in the south wing and connected to the south corridor. Rooms A, C and D are heated by hot water radiators in cold seasons. In room B there is a full-width convector under the window in addition to hot water radiators. Room E is heated by a convector of the same type as that for room B under the window in the south wall. During hot days a rotating fan was used in some of the tests. The investigation lasted for 14 months in 1991/92. Tests were conducted in winter (1991) in room A, early spring (1992) in room B, late spring in room C, summer in room D and in winter (1992) in room E.

2.1 Measurements

During an experimental test the indoor air velocity, turbulence intensity and air temperature were measured continuously using thermal anemometers (DANTEC Multi-channel Flow Analyser type 54N10). Measurements were taken at points 0.1m (foot/ankle level), 0.6m (centre of gravity of a seated person) and 1.1m (neck/head level of a seated person) above the floor. The plane radiant temperature 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 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 rooms 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 5m above the roof). The outdoor air temperature and humidity were measured using a copper-constantan thermocouple (radiation shielded) and a hand-held humidity meter respectively.

2.2 Subjective assessment

A subjective assessment was undertaken simultaneously with the physical measurements. The assessment of the thermal environment was based on the occupants' vote on the thermal sensation and air movement in the offices under various outdoor and indoor conditions and different arrangements of window and door openings. This assessment was based on judgements at head and foot levels as well as for overall comfort. 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 five-point scale to rate the impressions of comfort with regard to air movement, odour intensity and air freshness. The rating scales for these thermal environment and air quality indices are shown in Table 1.

3. RESULTS AND DISCUSSION

A summary of the results for physical measurements of room environment is presented in Table 2. These measured results are discussed together with those from the subjective evaluation as follows.

3.1. Room environment

The Physical parameters describing the room environment, except for the air change rate, were obtained for every test. Figures 1 to 3 show the distributions of mean air velocity, turbulence intensity and mean air temperature at head level, foot level and overall (mean of three heights) respectively.

The air velocity in these rooms was usually below 0.17 m/s when the windows and doors were closed and fans were not in operation. Velocities above 0.17 m/s resulted either from the provision of an additional fan used in warm days in room D or from opening windows in rooms C, D and E. It is shown in Figure 1 that the air velocity at foot level was usually higher than that at head level, especially in rooms B and D which have a ceiling level higher than that of the other three rooms.

The turbulence intensity data is presented in Figure 2. The values were between low and moderate for room A but between moderate and high in the other four rooms (Table 2).

The indoor air temperature results are plotted in Figure 3. The temperature changed from day to day during the course of measurement due to the fluctuations of outdoor temperature, air change rate and heat loss or gain from the rooms and due to opening the windows or doors. In some tests overheating was observed during mild outdoor climates. Temperatures much higher than 26°C were due to the solar heat gain from the partly opened south window of room D in the summer. It can be seen in Table 2 that the air temperature at head level is higher than that at foot level with a mean vertical temperature difference of 1.7K.

The measured plane radiant temperature, and hence the calculated mean radiant temperature, were generally lower than the mean air temperature for all rooms except for room C where there was no external wall exposed to the cold ambient. The average difference between the mean air temperature and mean radiant temperature for all the tests was within 1K.

The relative humidity in the room throughout the test period was normally within the comfort limits, ranging from 40% to 55%. On some occasions it dropped to slight below 40%.

The air change rate was determined for most of the tests in rooms A, B, D and E. However, only one test was carried out for room C because the ventilating fan was always on during occupancy and the air change rate was assumed to be constant.

3.2 Subjective evaluation

Figures 4 to 6 present the distributions of votes over the measured space during the time of measurement. The results are those for thermal sensation, air movement, odour intensity and air freshness.

3.2.1 Thermal sensation

As shown in Figure 4, the mean thermal sensation was on the warm side of the neutral point. However, the measured PMV values, which were obtained from Fanger's comfort equation for the corresponding tests were close to the neutral point for most of the test conditions. This suggests that Fanger's equation underestimates the thermal impressions and under-values the deviations of the impressions from neutrality. This may be due to three main reasons. One is the assumption of steady state conditions used in the derivation of Fanger's equation. Another is the over-simplification of the metabolic rates of the occupants. The occupants rarely sat quietly in the rooms for a long period, say more than one hour, without engaging in some activity or movement. The metabolic rate was however taken constant as 1.2 met in the calculation of PMV due to the difficulty of predicting its exact value. The third reason is the sensitivity of PMV to the thermal resistance of clothing (clo value). In laboratory experiments, the clo values are consistent whereas in field tests the clothing levels vary with individual occupants and time.

The thermal sensation was found to be dependent on the air temperature and velocity in room A. The effect of air velocity was however insignificant for other rooms probably because of insufficient data collected for each subject or too high an indoor air temperature to be compensated for by a small increment of air velocity. The regression equation for the termal sensation (TS) at head level, foot level and overall for the rooms against mean air temperature (T in ^{0}C) and, for room A, velocity (V in m/s) can be expressed as follows:

$$TS = a\,T - b^4\sqrt{V} - c \qquad (1)$$

where a, b and c are constants whose values are dependent on the room environment.

In Figure 7, the occupant's thermal sensation responses are presented as a function of mean air temperature. The PMV lines predicted from Fanger's equation are also presented for comparison (assuming a metabolic rate of 1.2 met and clo values between 0.8 and 1.0 and using the average values of the measured air velocity and radiant temperature for the corresponding rooms). Note that a PMV line is theoretically not a straight line but because the curvature is very small, then the error caused by linerising the curve is negligible in the region close to the comfort temperature. From Figure 7 a neutral temperature i.e. T for TS = 0, can be obtained. The neutral temperature can also be predicted from Fanger's comfort equation. The neutral temperatures calculated from the above equation and from Fanger's equation together with the difference in neutral temperature between them are shown in Table 2.

It can be seen from Table 2 that Fanger's equation overpredicts the neutrality for the rooms except room C. The reasons for this were mentioned above, which seems to confirm the findings of Schiller, et al. (2) and Brager (3). They found that the predicted neutral temperature was on average 2.4K higher than that measured for 304 office workers in 10 buildings. In addition, Figure 7 indicates that the data lines from the present investigation are steeper than those given by Fanger's equation, suggesting the occupants are more sensitive to changes of air temperature. This fact was also observed by Fishman and Pimbert (5) whose field study showed that the gradient of their data curve deviated from Fanger's equation particularly at temperatures above 24^{0}C. Furthermore, they found that Fanger's comfort equation predicted the neutral temperature 0.6K higher than that obtained from the field survey, which was attributed to their incorrect estimation of the subjects' clothing.

If according to Fanger's definition the central three categories of the thermal sensation scale 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 were dissatisfied with the thermal environment whether for the head, foot or the overall impression. Most of the dissatisfaction that occurred in rooms A, B and E when the windows and doors were closed was caused by overheating, which could be avoided by window opening or by controlling the heat output from the emitters and hence reducing the heating costs with the help of a thermostat or a weather compensated heating system. For room C however these measures were not adequate because the emitter was already turned off during the test period. Although in the latter period of tests, a window was opened on the roof, the room was still uncomfortably warm in most days. The roof window was effective in reducing the indoor temperature only when the outside was windy such that cooler outdoor air would be blown into the room. A possible way to decrease the indoor temperature in cold seasons is to introduce air directly from the outside of the building rather than from the corridor, using the existing ventilating fan. Due to its location a comfortable thermal environment is difficult to achieve in room D in hot sunny summer days unless it is mechanically ventilated or air conditioned.

3.2.2. Air movement

Figure 5 shows that the overall impression of the air movement was on the side of being stagnant. For room A when a window and/or the door were partly opened, the impression of air movement shifted to being slightly draughty. The main cause of the draught was considered to be low temperature as air velocity and turbulence intensity were not very high.

Figure 5 also indicates 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. In these tests the feet were more sensitive to air temperature but less sensitive to air velocity than the head.

3.2.3. Odour intensity

Odour was detectable in most cases see Figure 6. However, no satisfactory correlation between odour intensity and CO_2 level or air change rate could be established. In some cases, when the CO_2 level was low, or the air change rate was high, odour was still perceivable while in other cases when the CO_2 level was higher than 1000 ppm the odour intensity was not detectable. This seems to suggest that there were other pollution sources such as building materials or furnishings which could be more significant than the odour emission from the occupants. The judgement could also be affected by fatigue of the olfactory sense due to long term exposure to the pollutants.

3.2.4 Air freshness

Figure 6 also shows that the rating of air freshness was in general slightly stuffy. The assessment of odour intensity and air freshness shows that the air change rate is not a good indicator of indoor air quality since the air reaching the breathing zone could be contaminated as a result of poor mixing in the room.

4. ENERGY SAVING POTENTIAL

Since the neutral temperature given by Fanger's equation is higher than that found from field measurements, energy can be saved by decreasing the room temperature. The potential for energy savings is described here.

In most office buildings, heat losses in cold seasons are mainly due to conduction and ventilation. These heat losses are proportional to the temperature difference between indoors and outdoors (T_i - T_0). Thus, for a comfort temperature based on laboratory models, the amount of heat loss, q_l, is

$$q_l \propto (T_{il} - T_0)$$

and for the comfort temperature which is based on the field measurements, the heat loss, q_f, is

$$q_f \propto (T_{if} - T_0)$$

where T_{il} and T_{if} are the required room temperatures predicted from the laboratory model of Fanger and from the field measurements respectively.

The amount of energy saving is then

$$(q_l - q_f) \propto (T_{il} - T_{if})$$

Hence the ratio of energy saving based on the room temperature recommended by the current standards is

$$(q_l - q_f)/q_l = (T_{il} - T_{if})/(T_{il} - T_0) \qquad (2)$$

In the UK, the average outdoor temperature during the heating season, T_0, is about 6°C and for the whole year is 10°C. Assuming the room temperature predicted for neutrality is 22.6°C, the ratio of energy saving is obtained for several differences between predicted and measured neutral temperatures and this is given in Table 3.

It is seen that lowering the room temperature by 1K represents about 6% reduction in space heating energy for T_0 = 6°C. For a temperature reduction of 1.1K, the maximum value measured in this investigation, the potential of energy saving is 6.6%. To take full advantage of this saving, it is necessary to control the heat supply to an office by installing an individually adjustable thermostat or by using a personalised environmental system.

5. CONCLUSIONS

The results show that in naturally ventilated office rooms the air velocity at foot level is generally higher than that at head level. The turbulence intensity in the room however does not seem to correlate with the air velocity. The CO_2 concentration does not represent a good indicator of the sensation of freshness in the rooms. This was attributed to the poor mixing of outdoor air with room air in naturally ventilated rooms.

From the present investigation, it seems that the thermal models based on laboratory tests at steady state conditions can not accurately predict the thermal environment for naturally ventilated offices where the conditions are transient and where the occupants invariably change their activities. For the cases investigated, Fanger's equation for thermal comfort generally overpredicts the neutral temperature and under-predicts the comfort requirement when the air temperature deviates from neutrality.

Heating energy can also be saved by lowering the room temperature from the currently recommended values by, for example, ISO 7730(4) without compromising the thermal comfort. Besides, a lower indoor temperature can reduce the occupants' complaints about the feeling of stuffiness.

REFERENCES

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Table 1. Rating scales for thermal sensation (TS), air movement (AM), odour intensity (OI) and air freshness (AF).

RATING	TS	AM	OI	AF
-3	cold			
-2	cool	too draughty	not detectable	very fresh
-1	slightly cool	draughty	slight	fresh
0	neutral	acceptable	moderate	neutral
1	slightly warm	stagnant	strong	slightly stuffy
2	warm	very stagnant	very strong	stuffy
3	hot			

Table 3. Energy savings during the UK heating season due to a lower room temperature.

$(T_{a} - T_{a})$ (K)	$(q_1 - q_0)/q_1$ (%)
0.5	3.0
1.0	6 0
1.1	6.6
2.0	2.0

Table 2. Physical and thermal properties of room environment.

Room No.	A	B	C	D	E	ABCDE*
Dimension (m)						
Length	5.4	11.6	4.2	4.4	7.6	
Width	2.3	2.9	3.5	2.3	2.4	
Height	2.6	3.4	2.6	2.6	3.4	
Effective volume+ (m^3)	29.3	108.2	37.5	25.0	62.0	
Normal occupants	1	3	2	1	3$	
Average air change rate (h^{-1})	0.86	0.86	7.60	3.03	3.81	
Average air supply rate (l/s per person)	7.0	8.6	36.9	21.0	21.9	
Mean air velocity (m/s)						
Head level	0.059	0.071	0.098	0.063	0.066	0.071
Foot level	0.064	0.100	0.111	0.086	0.130	0.095
Overall	0.060	0.082	0.099	0.067	0.098	0.080
Turbulence intensity (%)						
Head level	39.4	59.2	43.8	38.6	47.8	44.5
Foot level	28.7	44.4	34.0	33.0	28.1	32.9
Overall	34.7	54.3	41.2	37.1	39.6	40.3
Mean air temperature (°C)						
Head level	23.1	23.8	25.7	27.1	23.1	24.5
Foot level	21.4	21.7	24.5	25.0	21.7	22.8
Overall	22.4	22.9	25.1	26.2	22.4	23.7
Difference between air temperature and radiant temperature (K)	0.6	0.7	-0.7	0.6	0.7	0.5
Relative humidity (%)	45.8	45.7	42.9	47.6	45.5	45.5
Measured neutral temperature (°C)						
Head level	22.4	22.4	23.2	22.8	21.1	21.7
Foot level	21.4	20.4	21.1	22.1	21.0	21.0
Overall	22.0	21.7	22.5	22.7	21.4	21.7
Predicted neutral temperature from Fanger's equation (°C)#	22.8	22.8	22.3	22.9	21.4	22.6
Difference between predicted and measured neutral temperature (K)	0.8	1.1	-0.2	0.2	0.0	0.9

Notes: * average of the data for rooms A, B, C, D and E
\+ the volume excluding the space occupied by obstacles
$ the occupants are not the normal office users
\# based on 0.8 clo and 1.5met for rooms A,B,C and D, 1.0 clo and 1.2met for room E, 0.83 clo and 1.4met for all rooms

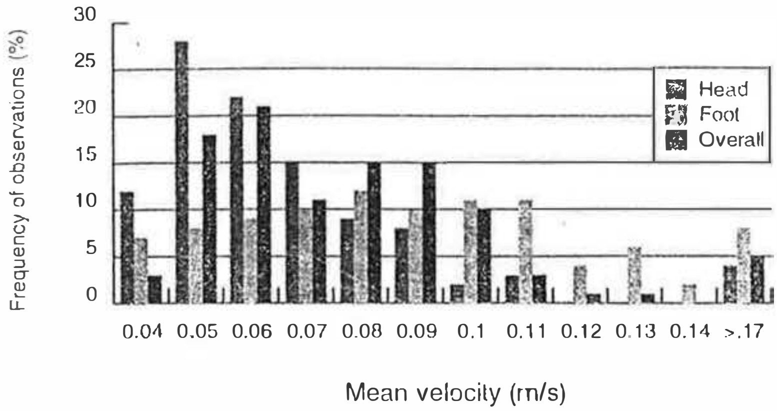

Fig 1 Frequency distribution of mean air velocity

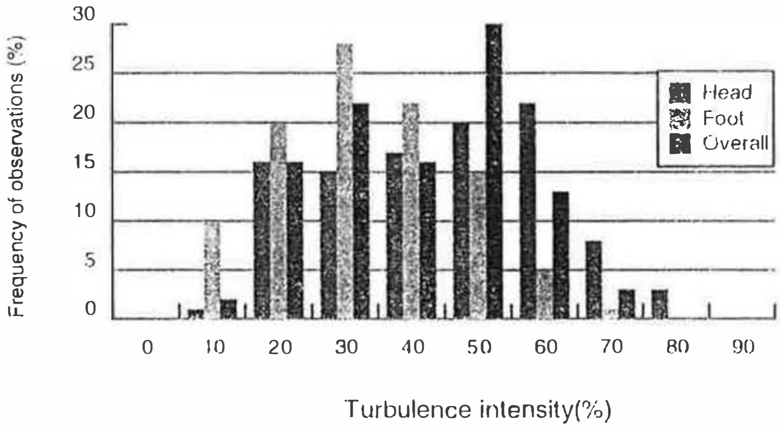

Fig 2 Frequency distribution of turbulence intensity

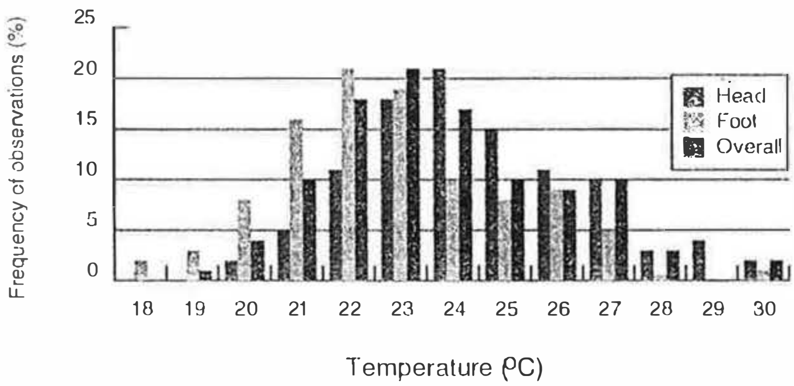

Fig 3 Frequency distribution of mean air temperature

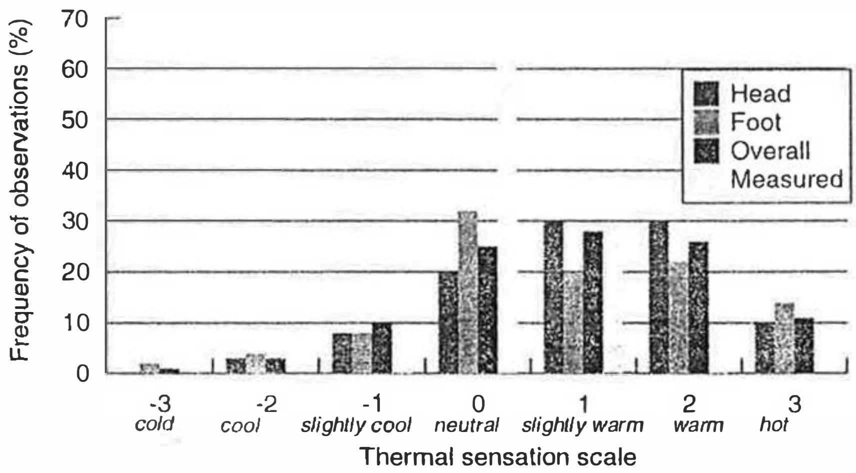

Fig 4 Frequency distribution of thermal sensation vote

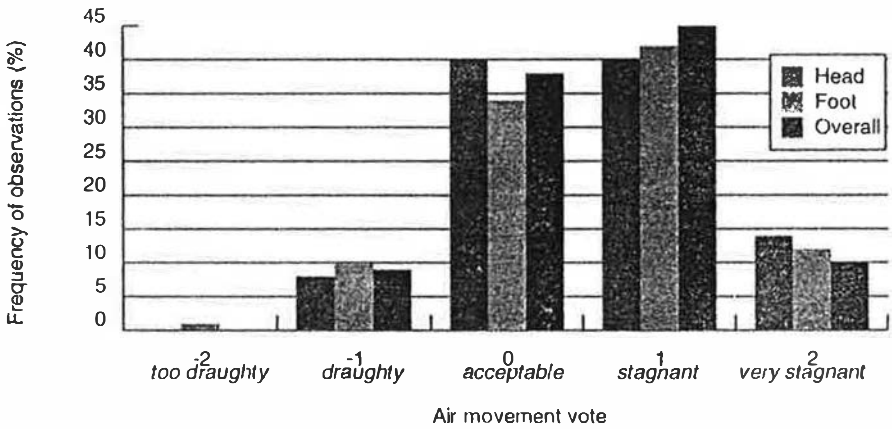

Fig 5 Frequency distribution of air movement scale

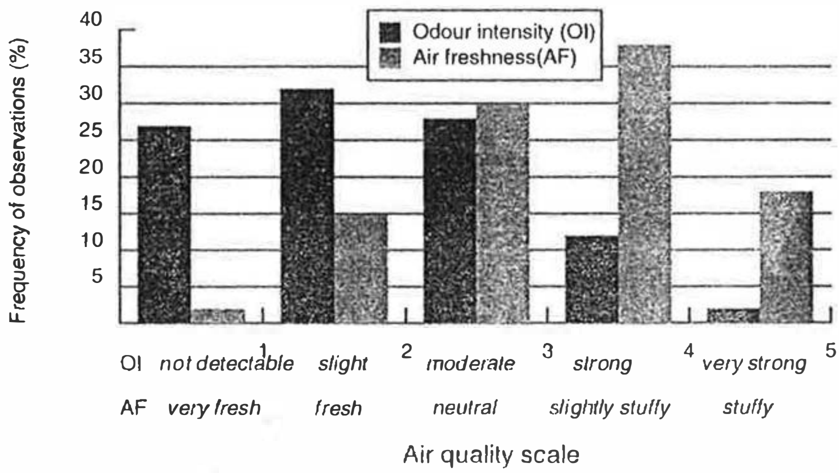

Fig 6 Frequency distribution of air quality votes

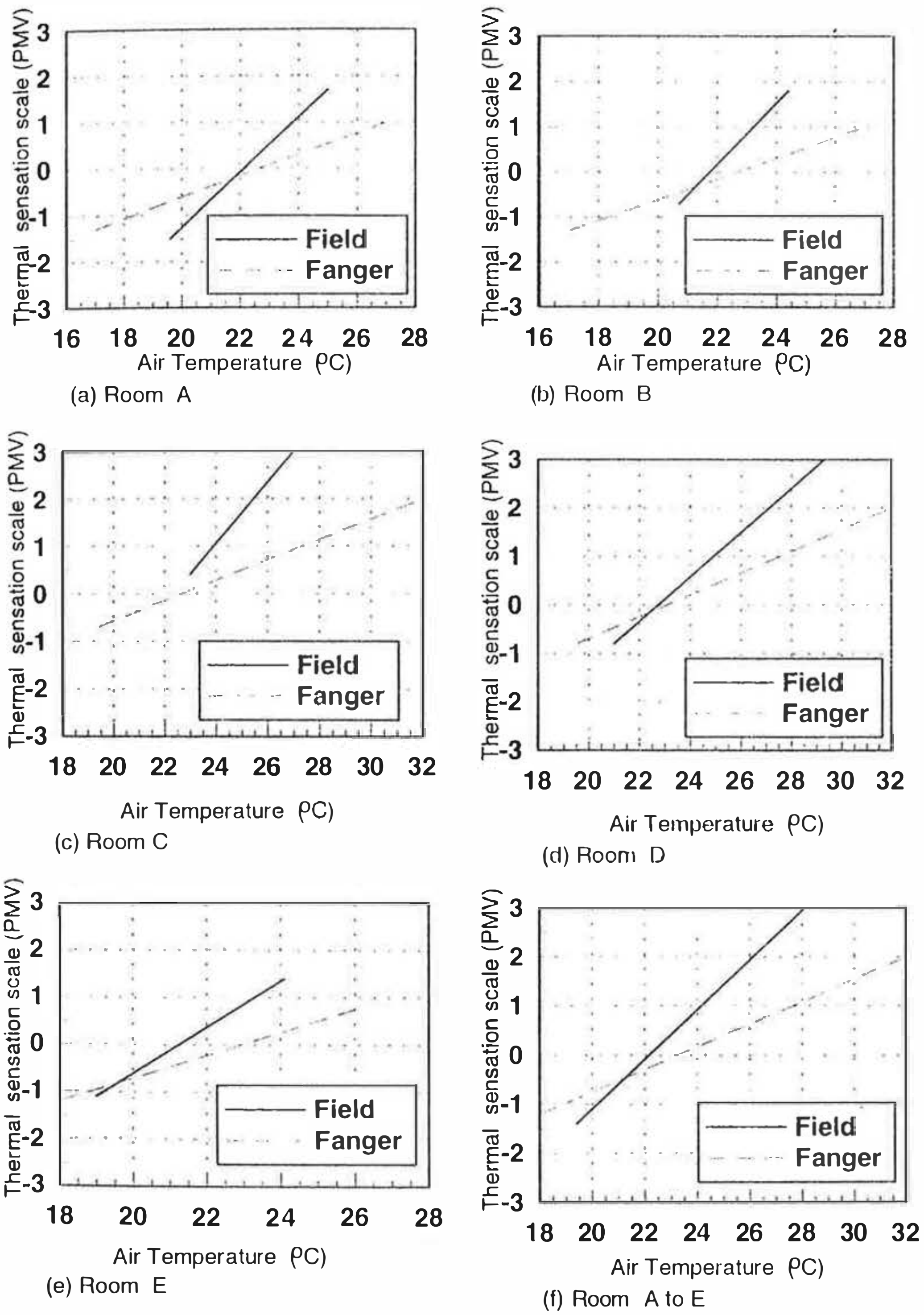

Fig 7 Effect of air temperature on thermal sensation