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Thermal Comfort Aspect of Three Different Heating Systems

Field Measurements

Key Words
Heating systems
Thermal comfort

Abstract
The performance of three heating systems with respect to thermal comfort based on data collected over two heating seasons in a three-storey building was measured. The systems were compared according to the thermal comfort as well as local thermal discomfort parameters. In terms of global thermal comfort, all systems performed similarly; however, in terms of their ability to distribute heat to a room and their performance with respect to local thermal discomfort, all three systems performed differently.

Introduction

The purpose of heating systems is to deliver heat to an environment and maintain suitable conditions for its intended use. The way different systems accomplish this purpose distinguishes them, so comparisons between them are needed. There are many criteria on which systems can be classified. For example, one can use simply the efficiency in terms of energy consumption for the heat output. This is the mechanical efficiency.

The mechanical efficiency is an important criterion but not the only one. It does not describe how the heat is distributed in a space. If the space is intended for human occupancy, the prime consideration should be the thermal comfort of occupants. Recent studies have shown that the indoor environment at offices and in homes do not always meet the standards that have been set [1, 2]. Complaints about air movement, dry air or temperatures that are too high or too low, are common [3]. Thermal comfort is a basic need if people are to feel satisfied in an indoor environment [4, 5].

The prediction of thermal comfort in a room requires an accurate understanding of the room air temperature and velocity distributions. These are influenced by the dimensions of the room, the air infiltration, the number of outside walls, the size and type of windows, the amount of insulation, and the outside temperature. An attempt to evaluate the thermal comfort aspects of heating systems, or comparing one system with another requires the measurement of all the parameters that influence the thermal satisfaction of occupants, while the geometric configuration of the building is constant. Air temperature, temperature gradients, air movements, and humidity are examples of important factors that have to be analyzed. The ASHRAE Standard set limits for acceptance, and outlines the instrumentation and procedures necessary to test for compliance [1, 2].

The purpose of this paper is to assess the thermal performance of several heating systems from detailed field measurements of various indoor parameters.

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Test Facility and Experimental Procedure

Three heating systems were installed separately on each of three floors of a building with a forced-air, modular and hydronic system on the first, second and third floors, respectively. Moreover, on the first and third floors, a modular system was added in one room in order to compare this system with the other two. Tests were concentrated in these two rooms. Test rooms selected had almost identical geometrical configurations.

Each type of heating system produces a specific thermal condition in a space which requires more than a single room air temperature measurement to describe it. Thus, air temperature, mean radiant temperature, interior and exterior room surface temperatures, and air velocity were measured. For room air temperature measurements, a grid of nine thermocouple columns at nine locations in two rooms was used (fig. 1).

In order to simplify the comparison of the systems, a standard room condition was imposed. This consisted of: (1) keeping the door closed, (2) preventing the penetration of direct solar radiation into the room, and (3) allowing for a high convective heat exchange at the window surface (using a reflective panel on the outside). A thermostat setting of 21 °C was selected. The systems were studied over periods of 3–4 days a week during the heating season of 1988–1989.

To calculate the comfort indices, and hence to study the acceptability of the heating systems throughout the heating season, a series of calculations was performed. The calculations were based on the measured comfort parameters and some assumptions concerning the occupants' activity and clothing.

Fanger [6] developed a mathematical model to predict the thermal and physiological response of a human to an environment. The basis of Fanger's model is that the internal temperature of the human body remains constant if there is a balance between the heat production by the body and the heat loss to the environment. The heat balance equation for a clothed person is:

$$M \pm W - E - RES = \pm K = \pm R \pm C,$$

where

- M = Metabolism,
- W = External work,
- E = Heat exchange by evaporation,
- RES = Heat exchange by respiration,
- K = Heat conduction through clothing,
- R = Heat exchange by radiation,
- C = Heat exchange by conduction.

Fanger then developed a procedure, based on the testing of subjects in an environmental chamber, to calculate the thermal sensation or predicted mean vote (PMV):

$$\begin{aligned} PMV = {} & (0.303\, e^{-0.036 M} + 0.028)\, [(M - W) \\ & - 3.05 \times 10^{-3} \{5{,}733 - 6.99 (M - W) - P_a\} \\ & - 0.42 \{(M - W) - 58.15\} \\ & - 1.7 \times 10^{-5} M (5{,}867 - P_a) - 0.0014\, M (34 - t_a) \\ & - 3.96 \times 10^{-8} f_{cl} \{(t_{cl} + 273)^4 - (t_r + 273)^4\} - f_{cl} h_c (t_{cl} - t_a)], \end{aligned}$$

where

$$\begin{aligned} t_{cl} = {} & 35.7 - 0.028 (M - W) - 0.155\, I_{cl} [3.96 \times 10^{-8} f_{cl} \\ & \{t_{cl} + 273)^4 - (t_r + 273)^4\} + f_{cl} h_{cl} (t_{cl} - t_a)] \end{aligned}$$

$$h_c = \begin{cases} 2.38 (t_{cl} - t_a)^{0.25} & \text{for } 2.38 (t_{cl} - t_a)^{0.25} > 12.1 \sqrt{v} \\ 12.1 \sqrt{v} & \text{for } 2.38 (t_{cl} - t_a)^{0.25} < 12.1 \sqrt{v} \end{cases}$$

$$f_{cl} = \begin{cases} 1.00 + 0.2\, I_{cl} & \text{for } I_{cl} < 0.5 \text{ clo} \\ 1.05 + 0.1\, I_{cl} & \text{for } I_{cl} > 0.5 \text{ clo}, \end{cases}$$

where

- PMV = Predicted mean vote,
- I_{cl} = Thermal resistance of clothing (1 clo = 0.155 m² °C/W),
- f_{cl} = Ratio of the surface of the area of the clothed body to the surface area of the nude body,
- t_a = Room air temperature (°C),
- t_r = Mean radiant temperature (°C),
- v = Air velocity (m/s),
- P_a = Water vapour pressure (P_a),
- t_{cl} = Surface temperature of clothing (°C),
- h_c = Convective heat transfer coefficient (W/m² °C).

PMV gives values over the range of −3 to +3 corresponding to cold and hot thermal sensation. Another index, the predicted percentage of dissatisfied (PPD), has been devised by Fanger [6] and sets as a criterion for comfort that conditions between +1 and −1 PMV do not create discomfort. PPD can be determined from the equation:

$$PPD = 100 - 95\, e^{-(0.03353\, PMV^4 + 0.2179\, PMV^2)}.$$

The lowest value of PPD is 5 % dissatisfied corresponding to a PMV of zero.

The PMV and PPD indices provide an evaluation of comfort for individual points in a room. In order to evaluate comfort for the whole room, the lowest possible percentage of dissatisfied (LPPD) is used.

The LPPD is calculated using the PMV and PPD of each location in the room. First the average PMV for the whole room is obtained, and this mean PMV will be a negative or a positive value, or zero. If the average PMV is zero, the LPPD is simply the mean of the PPD for the entire space. If the average PMV is non-zero then the LPPD is based on corrected values of the PMV for each location in the room. The corrected values of PMV are found by subtracting the average PMV from the PMV value of each point. With the corrected PMV value for each location, the corresponding corrected PPD value for each point is then found. The LPPD can then be obtained as the average of the corrected PPD values for the whole room. The difference between LPPD and 5 % (i.e. the lowest possible percentage of dissatisfied) is a 'figure of merit' or a measure of the non-uniformity of the room environment.

The LPPD, therefore, represents how the comfort at points in the room varies with respect to the average and with respect to the lowest possible percentage of dissatisfied. Fanger [6] set out a value of 6 % as a maximum value for acceptable LPPD.

The calculation of the PMV, PPD and LPPD comfort indices [6, 7] were performed using a few different cases for each system; each case combining thermal comfort parameters in a different way. It has been assumed that the occupants' activity might change. For example, a housekeeper might be busy cleaning and hence produce heat from her high metabolic rate. Data collected for room air temperature, t_a, and mean radiant temperature, t_r, are used for calculation while a range of observed values for air velocity is considered. Table 1 gives the range of air velocity and metabolic rate used in the calculation [8].

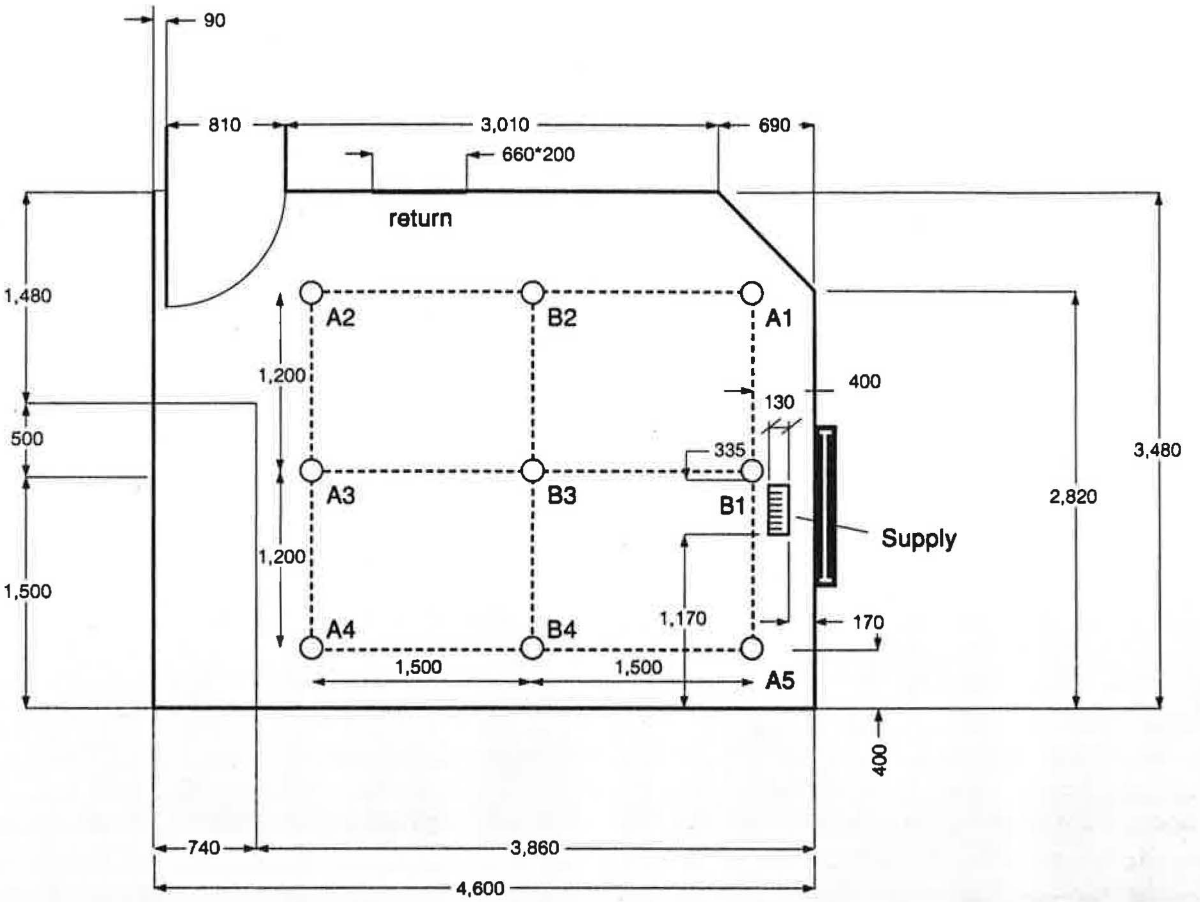

Fig. 1. Plans of thermal comfort room and thermocouple grid location, 1st floor.

Results

LPPD Index

The results for the LPPD index calculations are reported first because this index provides the evaluation of thermal comfort that is required to (1) evaluate the thermal uniformity in the rooms, (2) make this evaluation independent of the temperature maintained by the thermostat, (3) allow to compare systems with respect to each other, and (4) take into account the fact that there will always be a predicted minimum of 5% dissatisfied in any room. It is a general index that can be used before examining the PMV index results which are case dependent.

For all the cases selected and for all the periods considered, the LPPD index has remained between 5.0 and 6.0% for the forced-air, the hydronic and the modular systems. In other words, all the systems maintained the rooms in conditions such that it would be expected that a maximum of 6.0% of the persons would feel dissatisfied thermally.

If we recall that Fanger [6] qualified his recommendation of a maximum of 6.0% for LPPD as a 'severe' demand, then we can state that the systems performed very well in terms of thermal uniformity throughout the heating season.

Room Averages

The results for LPPD indices can be further inspected using the individual results for the PMV and PPD indices in order to describe the thermal uniformity of the rooms.

Table 1. Parameters selected for comfort assessment

System	Case	v, m/s	M	I_{cl}	PMV	PPD
Forced	1	0.10	1	0.7	-0.9	20
	2	0.20	1	0.7	-0.1	32
	3	0.00	1.2	0.7	-0.2	6
	4	0.20	1.2	0.7	-0.5	10
Hydronic	1	0.10	1	0.7	-1.5	50
	2	0.15	1	0.7	-1.7	60
	3	0.00	1.2	0.7	-0.8	17
	4	0.15	1.2	0.7	-0.9	24
Modular 1st floor	1	0.10	1	0.7	-1.1	30
	2	0.15	1	0.7	-1.3	39
	3	0.00	1.2	0.7	-0.5	9
	4	0.15	1.2	0.7	-0.6	13
Modular 3rd floor	1	0.10	1	0.7	-1.2	37
	2	0.15	1	0.7	-1.4	46
	3	0.00	1.2	0.7	-0.6	12
	4	0.15	1.3	0.7	-0.7	15

The results of PMV and PPD in terms of room seasonal average are given in table 1.

For all the systems, cases 3 and 4 were the ones for which the thermal comfort parameters combined in the best way as represented by the corresponding PMV and PPD values. Recall that cases 3 and 4 had the same activity level and clothing insulation values (1.2 met, 0.7 clo) but case 4 had a higher air velocity. Consequently, higher PPD values were observed for case 4 than for case 3. Cases 1 and 2 were set at the low activity level of 1.0 met and the lower PMV and higher PPD were expected.

As expected from the distribution of all cases and all the systems, PMV values were found to be below zero, with thermal sensation predicted to be between slightly cool (PMV = -1) and cold (PMV = -2) and PPD ranging from 9 to 60%. It was noted that the average room temperature observed for the hydronic system and the modular system on the third floor were lower than for the other two systems [5]. This explains why these systems attained lower PMV and PPD indices. In this sense, the results would have been better if the thermostat for these systems had been set higher.

For all the systems, if activity level and clothing insulation parameters are fixed, the change in air velocity will cause the comfort of the occupants to be modified significantly and in different proportions for each system. The range of air velocity observed for the forced-air system was higher than that for the other systems (maximum of 0.2 m/s compared to 0.15 m/s), and these higher values were assumed in the calculations. The PPD changes for the forced-air system would therefore be more important than for the other systems due to higher air velocity fluctuations.

Now, if the activity level is varied from 1.2 to 1.0 met then, for all the systems, the new conditions will not be satisfactory. For all the systems, this type of change in activity level will cause the number of persons dissatisfied to be between two times and five times higher at some point in time.

It is known that the PMV and PPD indices depend on air temperatures, indicating that the systems maintain conditions that are correct for comfort within a small range and, if the activity level or another parameter is modified, changes in thermostat settings are needed to satisfy the new conditions. This signifies that air temperatures, being fairly constant for the periods considered [5] and for each system, the environment created was suited for very specific conditions.

Room Thermal Distribution Spatial Variations

Further study on the uniformity of thermal distribution in the rooms based on the spatial and time variations of PMV and the corresponding PPD indices for each system was carried out. Using the data for the whole season allows one to identify how the rooms were maintained from a comfort point of view and decide if, for example, a specific location is always colder or warmer than another one, or if comfort conditions vary more with one heating system than with another one.

Results for PMV and PPD calculated were similar for a given system with differences less than 0.02 PMV between cases. For example, the average PMV deviation from mean to location B1 for the forced air were calculated as -0.07, -0.08, -0.06 and -0.06 for cases 1–4, respectively. Because these differences were small, it was decided to present results for case 3 only or for the conditions when all systems performed optimally.

Because the results from PMV and PPD calculations gave values close to zero PMV and 5% PPD, the thermal environment for all systems was considered neutral, so only small differences were observed from location to location. The results of calculations for room PMV, PPD, period averages and maximum PMV variations are presented in figure 2–5 for all systems. As can be seen, all four systems provided environments that were almost uniform throughout the rooms. Results for PMV deviation from mean are all between 0.2 and -0.2 PMV and PPD is less than 6%. The room variations were less than

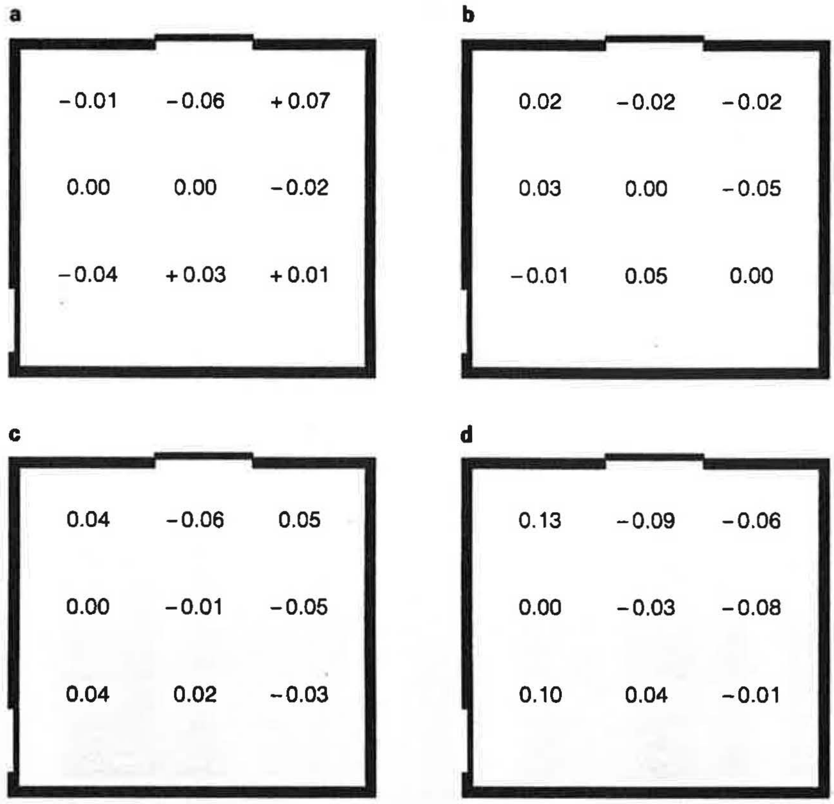

Fig. 2. Room distributions of deviation of PMV from mean value (seasonal average). **a** Forced-air system. **b** Hydronic system. **c** Modular system (1st floor). **d** Modular system (3rd floor).

0.4 PMV for all systems and maximum variations were less than 0.8 PMV for all the systems for the periods considered.

PMV Variations

The PMV at the location facing the window (fig. 2) was negative for all three systems, indicating that this zone would be expected to be slightly colder than the rest of the room. The compensation for this cool zone from the hydronic system is slightly better than for the other systems with a PMV of -0.02 compared to PMV of -0.06, -0.06 and -0.09 for the forced-air, the modular system on the first floor and the modular system on the third floor, respectively.

The room distribution of PMV for the forced-air system varied by 0.13 (0.07 to -0.06) for the different locations in the room (fig. 2). The general trend is to have the warmer locations at the lower right and in the upper right corners of the room. The centre location is at zero PMV.

For the hydronic system, the room distribution of PMV varied by 0.1 PMV (-0.05 to 0.05), with the colder points on either side of a diagonal going from the upper left side to the lower right side of the room. The PMV at the centre of the room was zero.

The room distributions of PMV for the modular systems are similar on both floors and were characterized by the cooler locations located on the right side of the room. The PMV range for the first floor was 0.11 (-0.06 to 0.05) and the PMV range for the third floor was 0.22 (-0.09 to 0.13). A higher PMV was found at the location close to the heater (+0.13) on the third floor, which was due to the fact that the modular system on the third floor was closer to the thermocouple grid than was the case on the first floor.

PPD Variation

PPD distributions for all the systems (fig. 3) show that values greater than 5.3% were not observed. Therefore, based on the measurments, the thermal acceptability of the rooms can be considered to be fairly good for all the systems. The higher PPD values at the different locations are the reflection of the PMV values shown in figure 2. For all the systems, the PPD at the centre of the room was 5%, the smallest value theoretically possible.

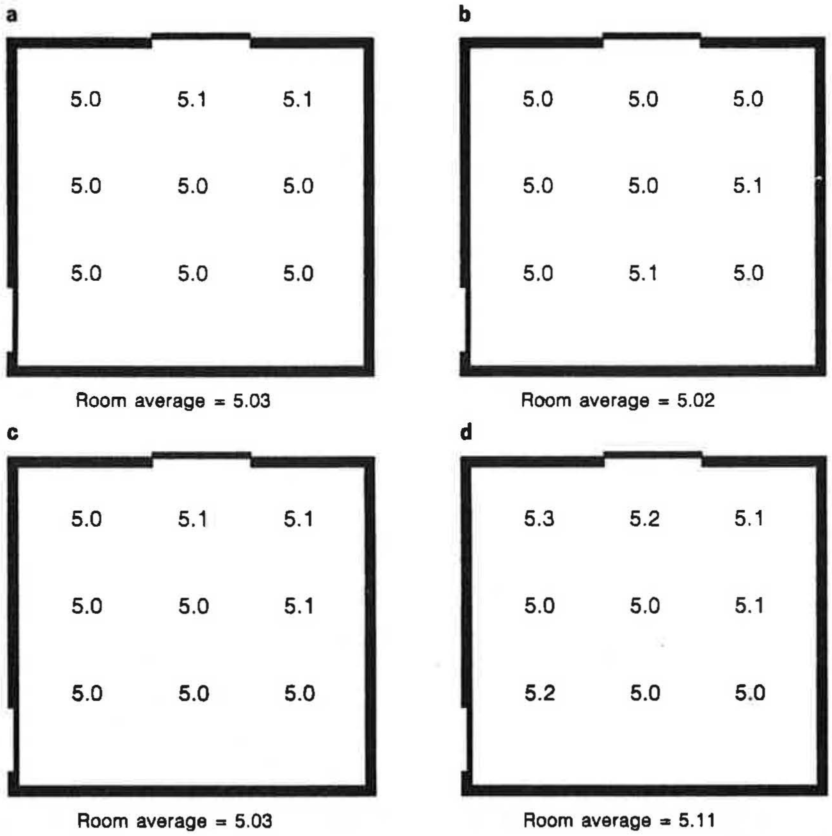

Fig. 3. Room distributions of deviation of PPD from mean value (seasonal average). **a** Forced-air system. **b** Hydronic system. **c** Modular system (1st floor). **d** Modular system (3rd floor).

Room Thermal Distribution Time Variation

The average PMV and PPD values of the room describe the general thermal comfort to be found in the room. The average and maximum variation of PMV can help to determine where and by how much the thermal sensations are expected to fluctuate in time or over a period such as the ones observed in the measurements.

For the forced air system, the average and maximum room variations of PMV are similar for all points, indicating again the effect of the forced-air system, which is to distribute heat uniformly throughout the room. The average PMV point variations observed were less than 0.25 PMV for all the periods considered (fig. 4a), and the maximum variations were less than 0.66 PMV (fig. 5a). The average room variation was 0.21 PMV. Notice how the local average and maximum variations in the rooms are close to each other.

For the hydronic system, the variations in PMV were in general slightly higher close to the heater than for the back locations. The average PMV point variations observed were less than 0.25 PMV for all the periods considered and the maximum variations were less than 0.64 PMV (fig. 4b, 5b). The room average variation was 0.19 PMV. The average variations observed close to the heater (front of window) were approximately 1.4 times higher than those in the centre of the room (0.22 PMV compared to 0.16 at the centre). For one location, the back centre location, variations observed were higher than the front location. This may result from an effect of the enclosure at the back wall, since similar variations were observed for the modular system on the third floor (fig. 4d, 5d).

The variations in PMV observed for modular systems on both floors were almost similar for the room average (0.21 PMV in both cases) but differed in terms of points measurements. The greatest difference was again for the location close to the heater where average and maximum variations were higher on the third floor (average = 0.33, maximum = 0.71) than on the first floor (average = 0.24, maximum = 0.36) (fig. 4c, d, 5c, d). This was probably due to the location of the modular system unit, which was located closer to the measurements location on the third floor.

For all the systems, the maximum variations in PMV were small over a period, especially if we consider the cor-

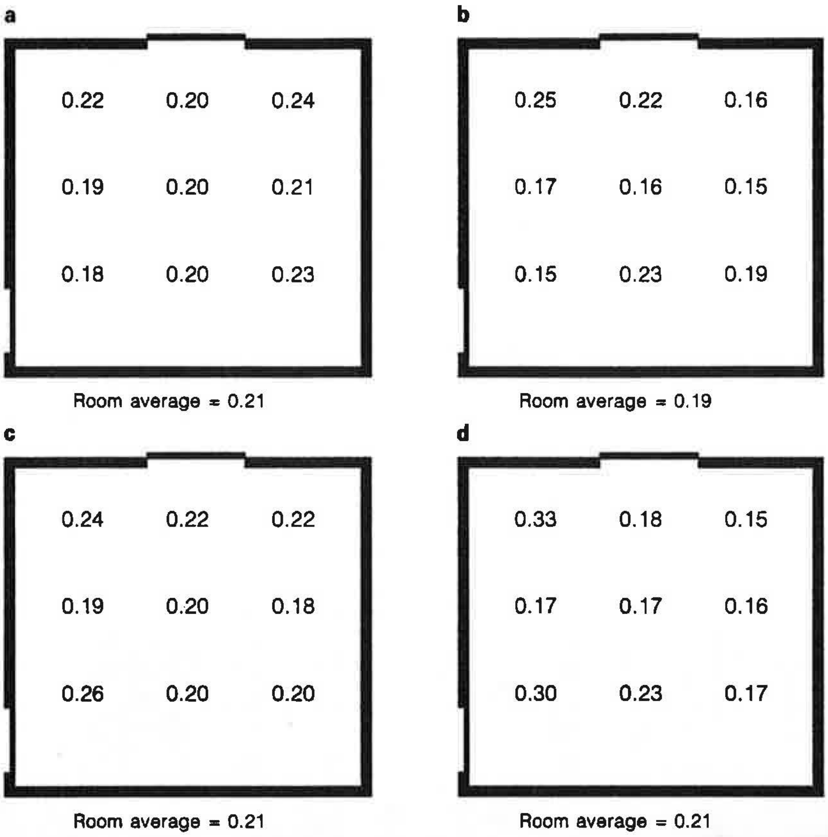

Fig. 4. Average variations of deviation of PMV from mean value per period (seasonal average). **a** Forced-air system. **b** Hydronic system. **c** Modular system (1st floor). **d** Modular system (3rd floor).

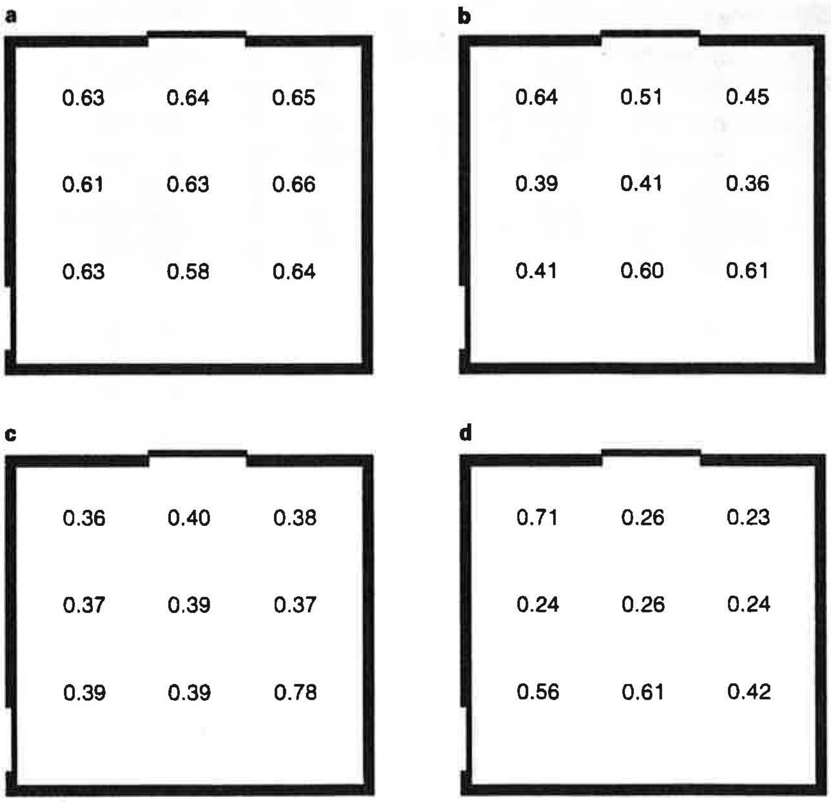

Fig. 5. Maximum variations of deviation of PMV from mean value per period (seasonal average). **a** Forced-air system. **b** Hydronic system. **c** Modular system (1st floor). **d** Modular system (3rd floor).

Table 2. List of local discomfort problems for each system

	Forced air	Hydronic	Modular
Asymmetric thermal radiation	Not a problem: surface temperatures close to air temperatures, heater surface temperatures (hydronic and modular) not high enough, window small in size and well insulated		
Draught	Not a problem: air velocity < 0.20 m/s and no major masses of cold air coming into the room		
Air velocity (standard: < 0.15 m/s)	Peak values close to 0.20 m/s average near 0.10 m/s, exceeded close to supply only	peak values < 0.10 m/s but assumed higher close to the baseboard	peaks < 0.05 m/s
Vertical temperature differences (standard: < 3 °C between 0.1 m and 1.1 m)	far from exceeded	serious problems close to the heater, correct elsewhere	close to limits but only at level higher than 1.2 m
Cold floors	Not a problem: due to boundary conditions, floor temperatures were only slightly lower than air temperatures		
Temperature drifts	Not a problem: systems were able to maintain temperatures which varied < 1.5 °C		
Cycling (standard: for peak-to-peak variations > 1.1 °C, rates < 2.2 °C/h)	peak-to-peak variations < 1.1 °C but rates of 6.3 °C/h observed for 0.3 °C peak changes at centre	problem close to baseboard where rates of 10 °C/h observed for peak changes of 1.3 °C at 0.3 m, elsewhere rates up to 3.1 °C/h but peaks < 0.8 °C	peak variations < 1.1 °C but rates up to 4.7 °C/h observed

responding changes in PPD. People in the room would not be expected to feel drastic changes in their thermal sensation. The thermal sensation would be expected to vary at most from neutral to slightly cool, taking the maximum change observed for all the systems at the neutral state (0.0 PMV). Furthermore, this maximum range was calculated based on variations occurring about a zero or neutral value and therefore includes values that range from the positive to negative. Therefore, the thermal sensations would vary at the most between slightly cold and slightly warm (−0.32 to 0.32 PMV).

Local Thermal Discomfort

The possible problems of local thermal discomfort are summarized in table 2 for: asymmetric thermal radiation, draught, temperature gradient, cold floors, temperature drifts and cycling.

The problems of cold floors, draught and radiant temperature asymmetry were not observed in this study due to the relatively good properties of the enclosure and the low heater surface temperatures.

For air velocity, only the forced-air system created peak values higher (0.23 m/s) than the standard requirements of 0.15 m/s but only for the location close to the supply with 0.1 m/s peaks elsewhere in the room.

Only for the hydronic system did vertical temperature differences exceed the standard limits. The baseboard of the hydronic system caused increases close to or higher than 3 °C/m between the 0.1 m and 0.6 m levels for locations close to heaters, however, this problem was not observed for locations away from the heater where gradients were below 3 °C/m and were less than the forced-air or the modular system values.

The major problem for all the systems was mainly the rate of temperature changes which were either exceeding or close to the standard requirements. The rates observed for the modular and the forced-air systems were higher than the limit of 2.2 °C/h but the peak-to-peak variations were less than 1.1 °C and therefore the combination of rate and time changes observed was within the standards requirement. The performance of the hydronic system was less satisfactory for the location close to the heater where rates of up to 10 °C/h were observed for peaks higher than 1.1 °C. For other locations, the hydronic system performed as well as the other systems.

Conclusion

In general, it can be concluded from measurements of PMV and PPD that all the systems can maintain thermally comfortable environments with only slight differences from the point of view of distribution and variations in these indices. The forced-air system produced a fairly uniform environment with small spatial variations throughout the room but high variations in time. The hydronic system also produced a slightly better compensation at the front of the window, with a spatial distribution less uniform than for the forced air system but with smaller time variations. In terms of spatial distribution, the modular system was found to create a zone slightly colder on the side of the room opposite to where the system was installed and created a significantly warmer zone close to the heater. Time fluctuations for the modular system were observed to be similar to those of the hydronic system.

Calculation of the LPPD, PMV and PPD indices showed that all the systems produced fairly good environments in terms of global thermal comfort with only small differences between the systems. The results did not show any major problems which could occur from the point of view of comfort if the centres of the rooms are considered. Small differences pointed to the fact that the forced-air system introduced higher variations in the environment, while the distribution created by the modular system may be problematic close to the heater. For the hydronic system, the general room distribution and ranges of variations were acceptable.

Relevance of Comfort Index

The results presented for time variations are to be viewed with respect to the assumptions which are behind the comfort indices. These indices were developed for steady-state conditions when occupants are exposed and have adapted to a given set of thermal parameters. In this sense, results based on PMV, PPD and LPPD should be valid only in those circumstances. However, in conditions such as those studied, the variations in thermal parameters in time were relatively small due mostly to the fact that the building was well insulated and tight.

The comfort standard 55–81 [1] considers acceptable to use time-weighted averages on an hourly basis in these conditions. If transient changes are considerable, then, the comfort indices should be set aside and one should focus on the variations of physical parameters individually (T, RH, v). In this case, the comfort study would become a study on 'local discomfort' rather than global comfort.

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