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Designing for Thermal Comfort in Combined Chilled Ceiling/Displacement Ventilation Environments

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ABSTRACT

This paper presents general guidance on designing for thermal comfort in combined chilled ceiling/displacement ventilation environments. Thermal comfort measurements involving 184 human subjects were carried out in a laboratory-based test room, constructed to resemble a normal office and equipped with a combined chilled ceiling and wall-mounted displacement ventilation system. Room characterization tests revealed that the chilled ceiling has a detrimental effect upon displacement flow, suppressing the stratified boundary layer at ceiling temperatures of 18°C - 21°C and destroying displacement flow all together at low ceiling temperatures (14°C - 16°C). Reduction in ceiling temperature was found to increase local air velocities at heights of 0.1 m and 1.1 m above the floor, showing further evidence of mixing, though there was an insignificant effect on local discomfort due to draft, as measured by subjective responses and by draft rating assessment. ISO Standard 7730 (1995) is shown to be valid, without modification, for predicting the thermal comfort of sedentary occupants performing office work in combined chilled ceiling/displacement ventilation environments. The vertical radiant asymmetry induced by a cooled ceiling does not significantly affect the thermal comfort of desk-seated occupants; this, together with relative humidity, is shown to require no additional comfort-related design limitations beyond those already in the literature and beyond the prevention of ceiling surface condensation.

INTRODUCTION

In the United Kingdom, as in many other industrialized countries, about 50% of total national carbon-dioxide emissions are the direct result of energy consumption in buildings. The U.K. has a temperate climate and a heating season that lasts from October to April; therefore, from the climatic viewpoint, widespread use of air conditioning is not a requirement. However, in commercial premises with significant internal gains, air conditioning has often been specified in order to achieve the required internal space conditions. Air conditioning is widely recognized as an energy-intensive solution, and attention is being paid to the adoption of lower energy techniques for the conditioning of office environments combined, where possible, with the provision of enhanced air quality and thermal comfort for occupants. One system that has been claimed to enhance the quality of air for breathing is that of displacement ventilation.

Displacement Ventilation

Displacement ventilation arrived in the U.K. from mainland Europe and consists of the provision of a fresh air supply to a space at low level, low velocity, and at a temperature lower than that of the desired zone air temperature. Density differences cause the fresh air to form a layer over the floor. The air then rises as it is warmed by heat sources (occupants, appliances) in the zone, and the convective plumes generated by these sources remove heat and contaminants that are extracted at ceiling level. The system is able to provide an environment of improved air quality, as compared with the mixing of air that occurs in conventional HVAC systems (for the same airflow rate conditions). Also, the same heat loads can be removed for a supply air temperature of typically 19°C, as compared with an air temperature of about 13°C in HVAC systems. As a result of thermal comfort limitations given in BS EN ISO Standard 7730 (ISO 1995) and currently assumed by practitioners to hold well in displacement ventilation environments (namely, that the vertical air temperature gradient should be less than 3°C/m), a displacement ventilation system is limited to removing a convective load of up to 25 Wm^{-2} of

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floor area (Sandberg and Blomqvist 1989). Office cooling loads can frequently exceed this figure, however, and so it becomes necessary to specify an additional cooling mechanism such as a chilled ceiling.

Chilled Ceilings

In a chilled ceiling system, cool water flows through pipework that is bonded to ceiling tiles, producing a typical ceiling tile surface temperature in the range 16°C - 19°C. Chilled ceilings can remove thermal loads of up to 100 W m^{-2} of floor area by the combined processes of radiation and convection and are considered to enhance the thermal comfort sensation of occupants in a manner analogous to the outdoors and beneath the open sky. When combined with displacement ventilation, the advantages offered by each system separately (improved air quality, enhanced thermal comfort) are claimed to be retained for the combined arrangement. But is this actually the case?

Chilled Ceiling/Displacement Ventilation

With regard to designing for the physical environmental conditions in offices equipped with chilled ceiling/displacement ventilation systems, guidance is now available, as surveyed by Fitzner (1996). To date, however, there is a limited amount of similar guidance with respect to designing for thermal comfort in such combination environments. Instead, it has been assumed that BS EN ISO 7730 (ISO 1995) can be applied—but how good is this assumption? It must be remembered that the ISO Standard is based on the work of Fanger, whose steady-state model for predicting thermal comfort was derived from measurements made in the early 1970s in more "conventional" environments; chilled ceilings, displacement ventilation, and the combined system produce environments that are more sophisticated than those used in the derivation of the Fanger comfort model, and so it becomes necessary to investigate the applicability of the existing standard to such new situations.

It is against this background that our three-year research program was undertaken, with the aim of answering the questions posed above and of determining the design conditions necessary for occupant thermal comfort in such combination environments, the role of the ceiling temperature being of particular interest in this context. The work was urgently needed because of the increasing numbers of such combined systems that are currently being specified in the U.K. and worldwide. It is, therefore, essential that sound advice becomes available to aid good design.

Some preliminary findings from the research program have already been presented (Taki et al. 1996; Loveday et al. 1997), and further papers that report on the detailed findings of the work are in preparation. In this paper, however, we provide a general overview of the findings for use by designers with respect to

1. the effect of the chilled ceiling temperature on displacement airflow;
2. the applicability of BS EN ISO 7730 (ISO 1995) for thermal comfort design in such combination environments;
3. the influence on thermal comfort of the vertical radiant asymmetry induced by the cooled ceiling.

The findings are based on laboratory experiments using a controlled-environment office, and general design guidance is given. The preliminary results from a field survey are also presented in support of the laboratory work.

THE TEST FACILITY

A test room has been constructed to act as an office environment, employing a chilled ceiling and displacement ventilation system (Figure 1). It is a lightweight room 5.4 m long, 3 m wide, and 2.8 m high, and its four walls are served by a

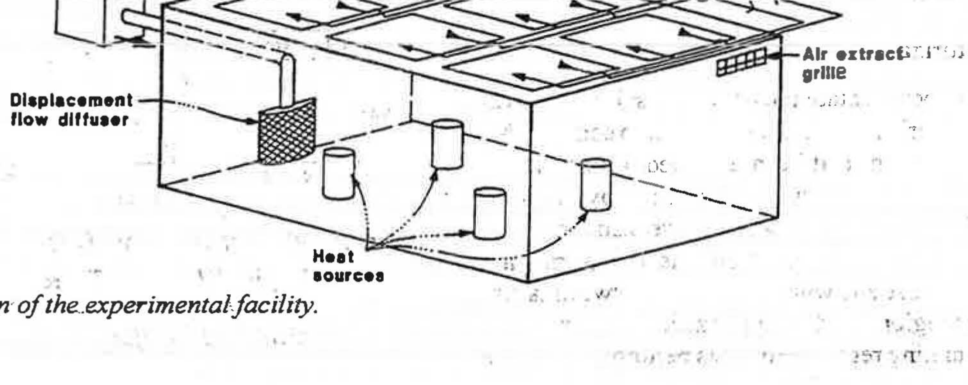

Figure 1 *Illustration of the experimental facility.*

waterflow network covered with metal panelling, which offers control of the wall surface temperatures. The chilled ceiling and displacement ventilation system is composed of commercially available units. The chilled ceiling has a 90% active area and consists of six individual circuits connected in parallel. Each circuit, in turn, is composed of four or five chilled panels connected in series, and the area of each circuit is approximately 2.5 m^{-2}. The circuits can be activated individually or collectively. Displacement ventilation is provided by a semi-cylindrical wall-mounted diffuser fitted at one end of the room; this is supplied with fresh air that can be tempered and humidified prior to entry into the space.

The room is equipped with a window that overlooks the external environment, thus preserving the impression of a normal office. However, the window is composed of seven layers of glass, providing insulation from the external environment and minimizing temperature differences between wall and glass surfaces. This effect is enhanced by extending the waterflow network to include the window itself and the piping being disguised as window framework. A set of blinds prevents the ingress of solar radiation and the formation of "sun patches" on the floor. The test room is carpeted and furnished to a normal office standard and contains four work places, each with a personal computer and desk lamp, and can either be equipped with thermal dummies to simulate human heat sources (as used in the tests on displacement flow/chilled ceiling interaction) or can be used for thermal comfort tests involving human subjects (a total of four subjects plus one experimenter).

All environmental parameters within the room are controllable: supply airflow rate, air temperature, relative humidity, mean radiant temperature, and the surface temperature of the chilled ceiling. All surface temperatures in the room are measured using type-T copper/constantan thermocouples to a resolution of ±0.2°C. The vertical air temperature profile in the room is recorded using eight radiation-shielded thermocouples (type-T) mounted on a column. All environmental parameters were logged every five seconds and average values were calculated every five minutes. The mean radiant temperature and the mean air velocity were measured at three heights (0.1 m, 0.6 m, and 1.1 m) above the floor using an indoor climate analyzer (see acknowledgments).

THE TEST ENVIRONMENT

Room Characterization

Prior to any experimentation with human subjects, it was necessary to characterize the test room in order to confirm that the required environment had been achieved. Using smoke visualization, displacement flow was confirmed by the presence of two distinct zones in the room: a lower zone in which the air remained horizontally layered, the layers moving slowly upward en masse and with little mixing between layers (the displacement region), and an upper zone in which the air was mixing (the mixing region). For tests performed with the chilled ceiling switched off, at heat loads of 25, 35, and 52 W m^{-2} of floor area, separation between the displacement and mixing regions occurred at a height of about 2.0 m above the floor. This is sufficient to maintain seated occupants within the better quality air of the displacement region. At a heat load of 62 W m^{-2}, however, separation occurred at a reduced height of about 1.5 m above the floor, and, at the same time, the vertical temperature gradient in the displacement region exceeded 3°C m^{-1} (contravening that currently assumed for human thermal comfort). For further details, refer to Taki et al. (1996). To ensure that occupants in such environments breathe only the fresh air supplied and that the temperature gradient remains less than 3°C m^{-1}, a greater supply rate of fresh air would be needed. However, since this could increase air velocity and turbulence intensity around an occupant (undesirable on thermal comfort grounds), the chilled ceiling was activated to remove the load under the same displacement flow conditions. Operation of the test facility in this way was, thus, in accord with current practice. The effect of ceiling temperature was then investigated.

Effect of Ceiling Temperature

A series of measurements was then made to determine the effect of the chilled ceiling surface temperature on displacement flow. The experimental procedure adopted was to measure the vertical air-temperature profile in the room for the following set of typical design conditions: a fixed heat load of 62 Wm^{-2} of floor area; a fixed airflow rate of 3.9 air changes per hour for the displacement ventilation supply, which was at a constant temperature of 19°C; and a range of ceiling surface temperatures, including one condition with the chilled ceiling switched off (displacement ventilation only). Inspection of the vertical profiles for air temperature (Figure 2) illustrates that as the ceiling temperature is reduced, there is suppression of the stratified boundary layer and eventual destruction of the displacement flow pattern. This, in turn, may have adverse consequences for air quality since it has been reported that vertical profiles of contamination in a displacement flow are

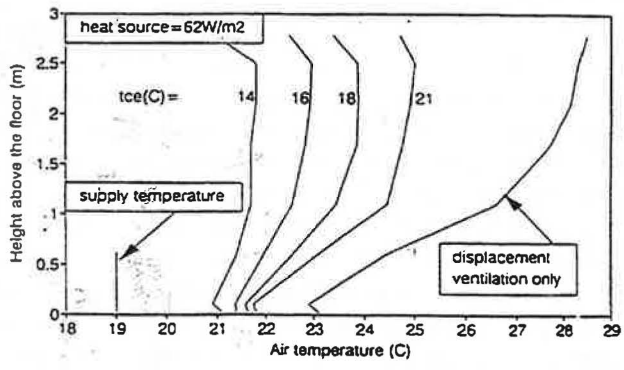

Figure 2 *Air temperature vs. height for a range of ceiling surface temperatures, t_{ce}, at a heat load of 62 W m^{-2} during room characterization experiments without test subjects.*

also severely influenced by the cooling capacity of a cooled ceiling (Fitzner 1996). The findings concerning the effect of ceiling temperature on displacement flow are in agreement with those reported by Alamdari and Eagles (1996), and it is concluded that the combination of a chilled ceiling with a displacement ventilation system could destroy displacement flow at low ceiling temperatures (14°C - 16°C). At higher ceiling temperatures (18°C - 21°C), some displacement flow is present, but the stratified boundary layer is strongly suppressed. This sets a design limitation if mixing is to be avoided (Fitzner 1996).

EXPERIMENTS WITH HUMAN SUBJECTS

Applicability of BS EN ISO 7730

This part of the research program determined the extent to which existing research findings and standards could be applied when designing for thermal comfort in combined chilled ceiling/displacement ventilation environments. A total of 200 paid volunteer subjects were recruited during the project, together with an assistant employed to deal with all aspects of test subject recruitment, preparation, care, management, measurement, and payment. An ensemble of appropriately sized typical office clothing was provided by the experimenters for each subject, consisting of, for males, long sleeve, white cotton shirt, buttoned to the neck; neck tie; dark, mixed-fiber trousers (65% polyester, 35% viscose), and for females, long sleeve, white cotton shirt buttoned to the neck (same type as for male subjects); dark, mixed-fiber knee-length skirt (65% polyester, 35% viscose, with 100% nylon lining); a pair of 15-denier nylon tights. The subjects wore their own undergarments (males—cotton underpants and cotton socks; females—bra and cotton pants) and their own office shoes (no sandals or training shoes). The clo value of both the male and female ensemble was estimated to be 0.75 clo.

Effects of Ceiling Temperature and Relative Humidity

For this set of experiments, a total of 128 subjects (64 males, 64 females), ranging in age from 21 to 60 years, took part. Subjects were admitted to the test room in groups of four (two males, two females) to carry out office tasks at their individual work stations for a period of three hours. The tasks consisted of active visual display unit (VDU) work, desktop filing, technical paperwork, and a questionnaire completion at 15-minute intervals. Subjects maintained an upright seated posture throughout the test period. They were accompanied by the assistant, who monitored and maintained their posture and activities, thus keeping their metabolic rate above resting levels to an estimated value of 70 W m^{-2} (1.2 met) (ISO 1995). Eight experimental conditions were set up: four values of chilled ceiling surface temperature (14°C, 16°C, 18°C, and 21°C) at two levels of relative humidity ("medium" and "low," corresponding to 47% RH and 26% RH, respectively). For all eight conditions, a displacement supply air temperature of 19°C at 3.9 air changes per hour was maintained. Subjects completed a questionnaire at 15-minute intervals throughout the three-hour test, the data from the last 30 minutes being used in the analysis (steady-state conditions). For calculation of PMV, values used for air temperature were those recorded at a height of 1.1 m above the floor and, similarly, values for mean radiant temperature.

Figures 3 and 4 show the comparison of predicted mean vote (PMV) from BS EN ISO 7730 (ISO 1995) and actual mean vote (AMV) over the conditions tested. Each experimental point is the average of 16 subjects' responses. For both low and medium relative humidity levels, there is excellent agreement between PMV and AMV values across the range of ceiling temperatures investigated. However, it is important to take account of the uncertainties associated with the metabolic rate and with the position at which air temperature is measured in relation to the space occupied by the human body. Recalculation of the PMV values was, therefore, carried out for the following two additional conditions. First, if air temperature measured at a height of 0.6 m is used instead of that at 1.1 m, then the PMV values would drop by less than 0.2 scale value

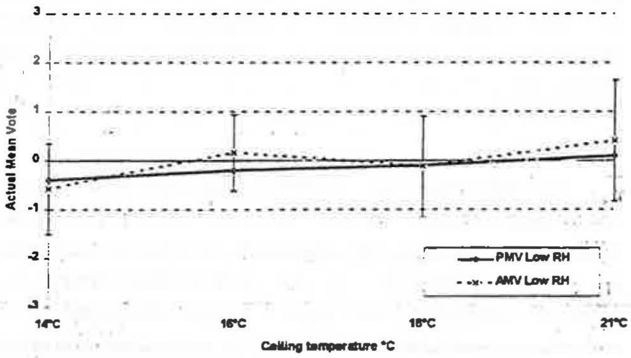

Figure 3 Comparison of AMV and PMV as a function of ceiling temperature at low relative humidity (N = 16 per ceiling temperature).

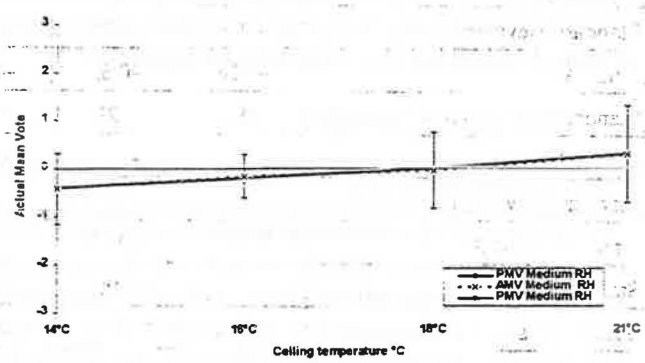

Figure 4 Comparison of AMV and PMV as a function of ceiling temperature at medium relative humidity (N = 16 per ceiling temperature).

TABLE 1
Effect of Different Metabolic Rate Values on the PMV

	PMV / met 1.2	PMV / met 1.1	Actual vote		PMV / met 1.2	PMV / met 1.1	Actual vote		PMV / met 1.2	PMV / met 1.1	Actual Vote
14°C / low RH	−0.4	−0.6	−0.58	14°C / med. RH	−0.4	−0.6	−0.42	2.5 ACH	0	−0.2	−0.18
16°C / low RH	−0.2	−0.4	−0.16	16°C / med. RH	−0.1	−0.3	−0.16	3.9 ACH	0	−0.2	−0.05
18°C / low RH	−0.1	−0.2	−0.12	18°C / med. RH	−0.1	−0.2	−0.05	6 ACH	-0.1	−0.2	−0.18
21°C / low RH	0.2	0	−0.4	21°C / med. RH	0.3	0.1	0.28	8 ACH	0.1	−0.1	0.97

for all conditions. This has an insignificant effect. on the agreement between PMV and AMV. Second, if a value for metabolic rate of 1.1 met is used (the lowest that is considered reasonable for the activities of the subjects), then the new values for PMV (Table 1) still show agreement with the AMV values reported by the subjects. Thus, it is concluded that the existing standard (ISO 1995) is valid for prediction of whole-body thermal sensation in such environments.

Following the standard technique described by Parsons (1990), Table 2 shows the corresponding local thermal sensations for different parts of the body with respect to the seven-point sensation scale, where zero corresponds to "neutral," and +1 or −1 corresponds to "slightly warm" or "slightly cool," respectively. It can be seen that mean local comfort sensations remain within +1 or −1 for all conditions, with no particular extremes of discomfort being observed. Table 3 shows a set of

TABLE 2
Summary of AMV Values for Overall and Local Comfort Sensations for the Chilled Ceiling/Displacement Ventilation Environment

Ceiling Temperature	14 °C		16 °C		18 °C		21 °C	
Relative Humidity	25%	50%	25%	50%	25%	50%	25%	50%
Overall AMV	−0.58	−0.42	−0.16	−0.16	−0.12	−0.05	0.40	0.28
Standard Deviation	1.05	1.01	0.83	1.14	0.89	1.09	1.05	1.34
Head AMV	−0.08	−0.09	0.00	0.30	0.25	0.10	0.47	0.58
Standard Deviation	0.97	0.86	0.62	0.81	0.91	1.05	0.97	1.26
Shoulders AMV	−0.40	−0.42	−0.13	−0.14	−0.04	−0.14	0.33	0.14
Standard Deviation	0.93	1.15	0.76	1.00	0.86	1.05	0.93	1.68
Trunk AMV	−0.22	0.33	−0.16	−0.11	−0.11	−0.05	0.37	0.31
Standard Deviation	0.89	3.78	0.72	1.07	0.90	1.02	0.89	1.18
Arms AMV	−0.24	−0.65	−0.09	−0.19	−0.15	−0.02	0.38	0.42
Standard Deviation	0.91	1.23	0.57	1.22	0.85	1.04	0.91	1.86
Hands AMV	−0.49	−0.48	−0.05	−0.23	−0.33	−0.13	0.55	0.56
Standard Deviation	1.06	1.11	0.72	1.19	1.01	1.75	1.06	1.08
Above Knee AMV	−0.27	−0.36	−0.26	−0.23	−0.03	0.15	0.36	0.37
Standard Deviation	0.94	0.95	0.63	1.05	1.06	0.99	0.94	0.90
Below Knee AMV	−0.60	−0.70	−0.65	−0.55	−0.44	−0.19	−0.05	−0.18
Standard Deviation	0.95	1.12	0.84	1.07	0.79	1.07	0.95	1.15
Feet AMV	−0.58	−0.79	−0.69	−0.60	−0.47	−0.03	−0.13	0.24
Standard Deviation	1.18	1.17	1.18	1.35	1.11	1.21	1.18	1.26

N = 16 per ceiling temperature/relative humidity combination

TABLE 3
Ranking of Overall and Local Comfort Sensations

Ceiling Temperature	14 °C		16 °C		18 °C		21 °C	
Relative Humidity	**25%**	**50%**	**25%**	**50%**	**25%**	**50%**	**25%**	**50%**
Overall	8	7	3.5	3.5	2	1	6	5
Head	2	3	1	6	5	4	7	8
Shoulders	7	8	2	4	1	4	6	4
Trunk	5	7	4	2.5	2.5	1	8	6
Arms	5	8	2	4	3	1	6	7
Hands	6	5	1	3	4	2	7	8
Above Knee	5	6.5	4	3	1	2	6.5	8
Below Knee	6	8	7	5	4	3	1	2
Feet	5	8	7	6	4	1	2	3
Sum of Ranks	49	60.5	31.5	37	26.5	19	49.5	51
Rank of Ranks	5	8	3	4	2	1	6	7

1 = nearest to neutrality
8 = furthest from neutrality

rankings for all conditions, where a ranking of "1" corresponds to an AMV nearest to neutrality and a ranking of "8" corresponds to an AMV furthest from neutrality (moduli only considered). If rankings of 6, 7, and 8 are then shaded, as shown in Table 3, it reveals that the preferred operating envelope is likely to lie in the range of ceiling temperatures between 16°C and 18°C for values of relative humidity up to "medium," with the most suitable condition being at 18°C ceiling temperature and medium relative humidity. These findings are for a displacement ventilation supply air temperature of 19°C at 3.9 air changes per hour, which are very typical design conditions. With respect to relative humidity, a Kruskall-Wallis analysis of variance test (Siegel 1956) (a nonparametric test of differences between sample groups) showed there to be no significant relationship between the level of relative humidity and the thermal comfort sensations reported by the test subjects.

Effect of Air Change Rate

The effect of air change rate was investigated in another set of experiments involving a total of 64 subjects (32 males, 32 females). Subjects followed the same procedure as above for the following experimental conditions: four values of displacement air supply flow rate (2.5, 3.9, 6.0, 8.0 air changes per hour), each at a fixed supply air temperature of 19°C, and a chilled ceiling temperature fixed at 18°C. The values for air changes per hour that were selected for testing represent a wider range than might normally be encountered in practice so as to ensure a thorough investigation. In all cases, the same wall-mounted diffuser was employed, the highest air change rate tested remaining well within the manufacturer's recommended flow rate for the diffuser. Figure 5 shows the comparison of PMV and AMV. From these findings, it is concluded that Fanger's model in the form of BS EN ISO Standard 7730 (ISO 1995) is valid, without modification, for predicting the whole-body thermal comfort sensation of sedentary occupants performing office-type tasks in combined chilled ceiling/displacement ventilation environments. Figures 6 and 7, respectively, show the effect of ceiling temperature (which, in turn, influenced the vertical temperature profile) on local air velocities in the test environment recorded at two locations – near to the diffuser (1 m distant from diffuser face) and at the opposite end of the room to the diffuser (4.2 m distant from diffuser face). The depth of the diffuser was 0.35 m. At each location, velocities were measured at 0.1 m, 0.6 m, and 1.1 m above the floor. The data are the mean of "low" and "medium" relative humidities and relate to a fixed airflow rate of 3.9 ACH at a supply temperature of 19°C. In general, velocities

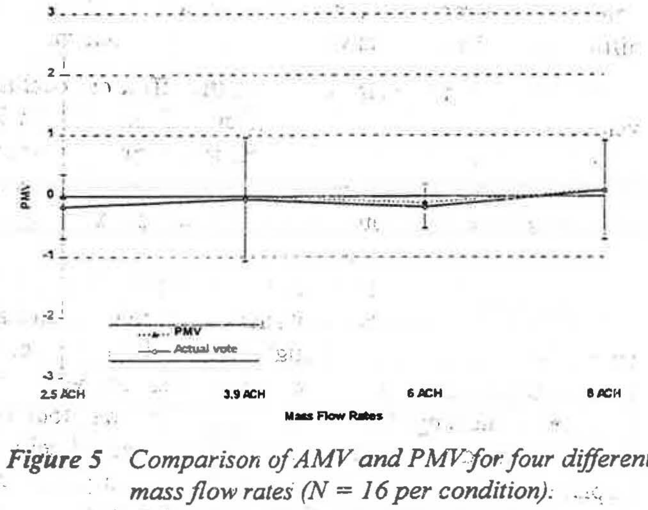

Figure 5 Comparison of AMV and PMV for four different mass flow rates (N = 16 per condition).

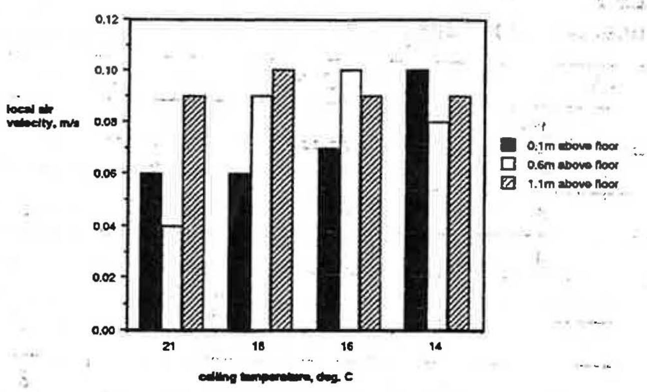

Figure 6 *Local air velocity vs. ceiling temperature: diffuser end 0.1 m, 0.6 m, and 1.1 m above the floor.*

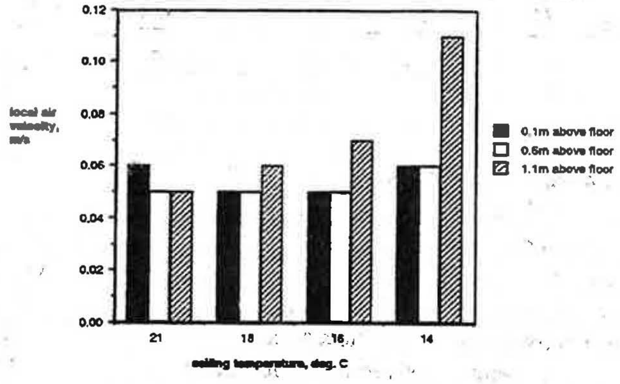

Figure 7 *Local air velocity vs. ceiling temperature: opposite end to diffuser 0.1 m, 0.6 m, and 1.1 m above the floor.*

are higher nearer to the diffuser, as would be expected. Reduction of the chilled ceiling temperature is seen to cause an increase in local air velocity at ankle level near to the diffuser and at head level at the opposite end of the room to the diffuser. This shows evidence of increased mixing of the air in the space at the lower ceiling temperatures and might suggest the formation of a whole-room cellular convection flow induced by the diffuser, though more data would be needed to confirm this.

Increases in air change rate had little effect on local air velocities, thus being indicative of effective diffuser design. In these tests, the ceiling temperature and the supply air temperature remain fixed at 18°C and 19°C, respectively. However, the highest air change rate tested (8.0 ACH) produced good mixing throughout the space, as evidenced by the vertical temperature profile observed at this condition. This could adversely affect air quality, but it should be noted that such an air change rate would be very high for a displacement ventilation system. In all cases examined, local air velocities remained in the range 0.04 to 0.11 m s^{-1}. There was found to be little or no draft risk associated with the chilled ceiling/displacement ventilation environment, with nearly all draft rating values remaining less than 10%. Mean local comfort sensations reported by the test subjects (Table 2) remained acceptable throughout the range of ceiling temperatures examined. This led to the conclusion that there is little risk of discomfort due to draft in chilled ceiling/displacement ventilation environments equipped with well-designed wall-mounted diffusers.

Effect of Radiant Asymmetry

Another set of experiments was undertaken to determine the effect on thermal comfort of vertical radiant asymmetry induced by the cooled ceiling, and whether this should lead to a design limit being imposed on ceiling temperatures. Only female subjects participated in these experiments because our preceding work had shown that the females were generally more sensitive than the males to the thermal environment under investigation. Eight subjects, tested individually and wearing the previously described clothing ensemble, carried out sedentary office-type work at a single work station, thermal dummies taking the place of the other human subjects. Ceiling temperatures of 22°C, 18°C, 14°C, and 12°C were investigated over a three-hour period, the subject being exposed to each condition in series. The PMV for each condition was maintained at a calculated value of "neutral" (as estimated from BS EN ISO 7730) by making small adjustments to all four wall-surface temperatures in the test room (thus, affecting room mean radiant temperature). Therefore, any departure from neutrality of the subject's AMV could be attributed to vertical radiant asymmetry only. Figure 8 shows the results, indicating that for the typical range of ceiling temperatures that would be experienced in such environments, increasing radiant asymmetry has almost no effect on the AMV of the desk-seated test subjects, obviating the need to set ceiling temperature design limits beyond those of normal practice (dew-point temperature). This finding is in agreement with that obtained by Fanger et al. (1985) for environments concerning comfort limits for asymmetric thermal radiation. It is concluded that the existing comfort criteria regarding radiant asymmetry can be applied without correction in chilled ceiling/displacement ventilation design.

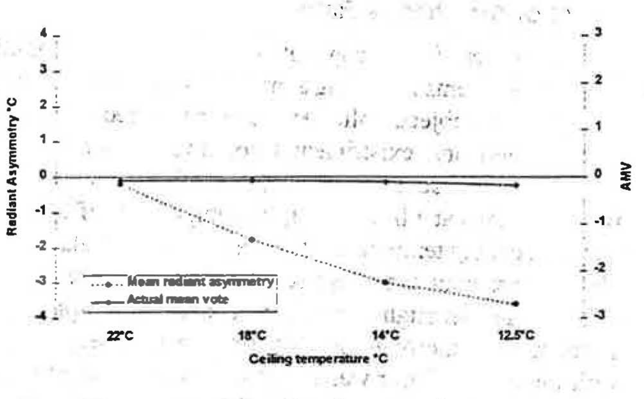

Figure 8 *Effect of vertical radiant asymmetry on AMV for ceiling temperatures.*

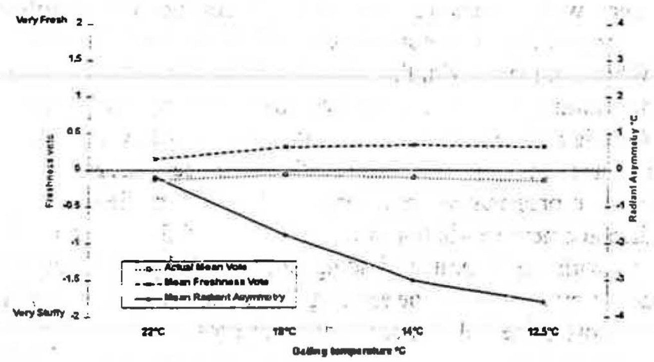

Figure 9 *Comparison of freshness with AMV and vertical radiant asymmetry (N=8).*

Freshness Sensation. As part of the radiant asymmetry investigation, subjects were asked to report their sensation of "freshness" on a five-point scale ranging from "very stuffy" to "very fresh." The results are shown in Figure 9. It appears that there may be a slight trend for the "freshness" sensation to increase with increasing radiant asymmetry. However, other factors, such as air movement and air temperature at the nose/mouth, can affect the sensation of "freshness" (Chrenko 1974; Fanger 1997). Table 4 shows the experimental conditions; though temperature at the nose/mouth level varied little during this experiment (no more than 0.3°C), radiant asymmetry and air movement did vary. While chilled ceilings may have an effect on the "freshness" sensation, it is recommended that further work be conducted to fully investigate, confirm, and quantify this effect. The work should be undertaken with confounding influences kept under control and with a larger number of test subjects.

TABLE 4
Experimental Conditions for the Radiant Asymmetry Experiments

Ceiling Temperature, °C	Temperature Difference Between Head (1.1m) and Ankle (0.1m), °C	Temperature at Nose/Mouth Level, °C
22	1.7	23.8
18	1.6	23.8
14	0.9	23.5
12	0.7	23.5

Effect of Vertical Thermal Gradients. In the design of chilled ceiling/displacement ventilation environments, practitioners currently assume that the vertical temperature gradient between 0.1 m above the floor (ankle level) and 1.1 m above the floor (head level for a seated occupant) should remain less than 3°C m^{-1} (ISO 1995) as a requirement to maintain thermal comfort. The original work upon which this limit is based was conducted by Olesen et al. (1979), who exposed 16 sedentary subjects (8 males, 8 females) for three hours to a range of vertical temperature gradients (0.4°C m^{-1}, 2.5°C m^{-1}, 5.0°C m^{-1}, and 7.5°C m^{-1}). The subjects were tested in a room in which the lower walls and floor could be cooled, while the upper walls and ceiling could be heated; no displacement ventilation was employed nor was the ceiling chilled. It was concluded that the vertical air temperature difference between positions 0.1 m and 1.1 m above the floor should not exceed 3°C to 4°C in order to maintain acceptable thermal comfort. However, it was pointed out that some of the discomfort expressed in the experiments had been caused by a difference in radiant temperature rather than a difference in air temperature (higher air and radiant temperatures at head level than at ankle level). It was then stated that if the radiant temperature of the upper and lower half of the room had been equal, the subjects would probably have tolerated a higher vertical air temperature difference.

In a chilled ceiling/displacement ventilation environment, the difference in mean radiant temperature between ankle and head level for the seated occupant is likely to differ from those encountered by Olesen et al. (1979). In our experiments, this difference had a maximum value of 1.2°C and a mean value of 0.7°C. These may be compared with corresponding values of 1.9°C and 1.15°C, respectively, determined for the experimental conditions of Olesen et al. (1979), and based on 0.7 times the vertical air temperature difference. Thus, the validity of the 3°C m^{-1} limitation on the vertical air temperature gradient could be questioned for the combined environment of a chilled ceiling and displacement ventilation. Throughout the experiments conducted in our study, the vertical air temperature gradient did not exceed 2.65°C m^{-1} in accord with current design practice. Table 2 shows there to be no evidence of unacceptable local discomfort.

No experiment to specifically investigate the effect of vertical temperature gradients on thermal comfort was conducted by the authors of this paper as part of the present study. However, such a study has been undertaken by Wyon and Sandberg (1996) using an experimental facility and test conditions very similar to our own (chilled ceiling, displacement ventilation at 4.0 ACH). Here, 207 subjects were exposed in groups of 2, 3, or 4 to nine experimental conditions (three levels of estimated whole-body heat loss and three levels of vertical thermal gradient: zero, 2, and 4°C m^{-1}). Subjects were allowed to wear their own clothing (clo values not stated) and were exposed to the conditions for one hour only. Values for the displacement air supply temperature are not given. It was found that local and whole body discomfort sensations were unaffected by thermal gradient, and it is suggested that thermal gradients due to displacement ventilation up to at least 4°C m^{-1} are likely to be acceptable, provided that air quality is satisfactory and that individual control of whole body heat loss is provided for sensitive individuals. These findings are important, as some degree of relaxation of the 3°C m^{-1} limit for chilled ceilings/displacement ventilation environments would have significant implications for the design of such combined systems and for the load removal capacity by displacement ventilation. However, since the paper also reports that discomfort due to dry eyes increased significantly above a thermal gradient of 2°C m^{-1}, further investigation is desirable before such a relaxation is recommended.

FIELD SURVEY

Oseland (1995) has reported on significant discrepancies occurring between PMV and AMV values obtained in offices as compared with climate chamber studies, attributing the difference to contextual and adaptation effects. Therefore, to complement our controlled environment investigations, a thermal comfort field survey was carried out in an open plan office consisting of about 25 - 30 staff. The office was situated in the U.K. and employed a chilled ceiling with fresh air supplied from floor diffusers. Subjects were asked to complete a questionnaire at the same time as the environmental variables (air temperature, mean radiant temperature, air velocity, and relative humidity) were being recorded at his/her location. As far as possible, data were taken in the same way, and using the same equipment, as employed in the laboratory-based studies, and an identical questionnaire was used. Details of clothing were noted for each subject.

Analysis of vertical temperature profile measurements showed that classical displacement ventilation was largely absent in this office and that the supply air temperature was, on average, about 22°C. In addition, a perimeter heating system was installed. Periodic measurements of ceiling surface temperatures indicated that the chilled ceiling was being cycled on and off in response to an imposed control signal. It was, therefore, concluded that the environment being surveyed differed significantly from that produced by a classical chilled ceiling/displacement ventilation system.

Comfort predictions from BS EN ISO 7730 (ISO 1995) agreed well with subjective responses (Figure 10). In general, the occupants reported their thermal sensation as being "too warm." This is due to the operation of the system as already described and, thus, is not representative of classical chilled ceiling/displacement ventilation environments. At this stage, it is possible to say only that the existing standard gives a reasonable prediction of thermal comfort for this environment, which can be regarded as being composed of a chilled ceiling in cyclic operation and a "floor ventilation" system. When compared with the results of similar surveys of fifteen air-conditioned and naturally ventilated offices (de Dear and Auliciems 1985), the agreement between AMV and PMV appears to be of a similar order of magnitude. A larger survey of a representative number of classical chilled ceiling/displacement ventilation offices is required for full comparison with the laboratory findings and for fully assessing the accuracy with which the existing standard can predict thermal comfort in the field for such environments.

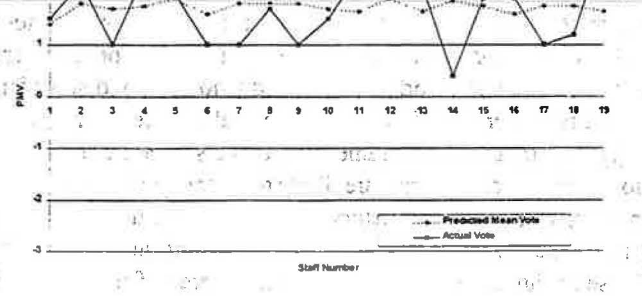

Figure 10 Comparison of the PMV and AMV for the field survey office.

CONCLUSIONS

Experiments have been carried out involving environmental parameters and human thermal comfort responses in a combined chilled ceiling/displacement ventilation environment. From tests over a range of near-typical design conditions, the following conclusions may be drawn.

1. The combination of a chilled ceiling with a displacement ventilation system could cause destruction of the displacement airflow pattern at low ceiling temperatures (14°C - 16°C). At higher ceiling temperatures (18°C - 21°C), some displacement flow is present, but the stratified boundary layer can be strongly suppressed with adverse consequences for air quality in the breathing zone. Achievement of true displacement ventilation in combination with a chilled ceiling, therefore, requires careful engineering design. For further guidance, refer to Fitzner (1996).

2. The Fanger comfort model in the form of BS EN ISO 7730 (ISO 1995) may be used, without modification, for predicting the overall thermal comfort of sedentary occupants performing office-type tasks in chilled ceiling/displacement ventilation environments over the range of typical design conditions encountered in practice. Variation of the ceiling temperature beyond practical design limits has shown that a ceiling temperature at or near 18°C is likely to be particularly suitable for overall and local thermal neutrality, though the range 16°C - 18°C would be quite acceptable for overall thermal comfort design.

3. Reduction in the temperature of the chilled ceiling causes an increase in local air velocities at 0.1 m and 1.1 m above the floor (evidence of mixing); for a typical ceiling temperature of 18°C, air change rates at about 8.0 ACH also cause mixing. Nevertheless, there remains little local discomfort due to drafts.

4. The vertical radiant asymmetry experienced within a typical chilled ceiling/displacement ventilation environment does not significantly affect the thermal comfort of desk-seated occupants. This, together with relative humidity, requires no additional design limitations beyond those already in the literature and beyond the prevention of ceiling surface condensation ("office rain").

The preceding advice is available in the form of a prototype software-based design tool. The guidance on designing for thermal comfort as furnished by this study may now take its place alongside that already available for selecting physical environmental conditions in offices equipped with combined chilled ceiling/displacement ventilation systems (Fitzner 1996).

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