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An experimental study of air flow and temperature distribution in a room with displacement ventilation and a chilled ceiling

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ABSTRACT

Displacement ventilation and chilled ceiling panel systems are potentially more energy efficient than conventional air conditioning systems and are characterized by the presence of vertical temperature gradients and significant radiant asymmetry. The characteristics of this type of system have been studied by making temperature and air flow measurements in a test chamber over a range of operating parameters typical of office applications. Results from the displacement ventilation study are consistent with other studies and show that normalized temperature profiles are independent of internal heat gain. Linear temperature gradients in the lower part of the room were found, in all cases, to be driven by convection from the adjacent walls. Significant mixing, indicated by reduced temperature gradients, was evident in the upper part of the room in the chilled ceiling results at higher levels of heat gain. Visualization experiments, velocity measurements and related numerical studies indicated that with greater heat gains the plumes have sufficient momentum to drive flow across the ceiling surface and down the walls. The significance of forced, as opposed to natural convection, is also suggested by relatively low Richardson Number (*Ri*) values found near the ceiling. Furthermore, in cases with moderately high internal gains, comparison of the temperature gradients indicated that the effect of ceiling surface temperature on the degree of mixing and the magnitude of the temperature gradient were of secondary importance. These findings are in contrast to the view that it is natural convection at the ceiling that causes enhanced mixing.

1. Introduction

Displacement ventilation has been used in Northern Europe to provide cooling and ventilation in office spaces for more than two decades [1–3]. Although displacement ventilation has potential to provide lower draught risk, higher air quality and lower energy consumption when compared to conventional all-air cooling systems it has the disadvantage of only being able to deal with moderate cooling loads. Using displacement ventilation and chilled ceiling technologies [4–6] together so that ventilation is provided by displacement ventilation and larger sensible cooling loads can be dealt with by a chilled ceiling [7–9] is now regarded as a conventional design option in commercial office projects [10–12].

In displacement ventilation systems, air is introduced into the room near the floor at very low velocity and at a temperature that is only a few degrees below the effective temperature experienced by the occupants. The Reynolds number at the diffuser is at least one order of magnitude lower ($\sim$3800 in these experiments), and the Archimedes number significantly higher (typically 0.2–2), than in conventional jet-driven air distribution systems. As a result, the cool supply air descends toward, and spreads across, the floor in a gravity current [13,14] and is slightly heated as it passes over the floor surface. Buoyant plumes are developed around any warm objects in the room, such as people and electrical equipment. Each plume increases in width as it rises and entrains more air from the surrounding cooler air. This warm (and contaminated) air rises until it reaches an upper zone bounded by the ceiling [15]. Accordingly, the room air distribution can be conceived of as a lower cooler zone, where conditions are close to that of the supply, and an upper zone of warm contaminated air that has been transported by the plumes. In steady state, the height of the boundary between these two zones (sometimes referred to as the ‘stationary front’) is defined by the height at which the flow rate in the plumes equals the supply flow rate [16].

The effect of the transport of heat to a layer of warm air near the ceiling is to produce a temperature gradient over most of the height of the room. There is a relationship between the air temperature

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gradient and thermal discomfort. Standards such as ASHRAE Standard 55–2010 [17] suggest a maximum overall head-to-foot difference of 3 K for acceptable thermal comfort. This could be interpreted as an allowable gradient of 2–3 K/m depending on if objects are considered seated or standing. To stay within these guidelines accordingly imposes a limit on the magnitude of the cooling load that can be met by displacement ventilation systems; values between 20 and 40 W/m^2 are suggested in the literature. It is this limitation that is addressed by combining displacement ventilation with chilled ceilings in order to achieve the capacities demanded in many office applications. In this study, heat gain intensities up to 72 W/m^2 have been studied experimentally and this is representative of design levels for displacement ventilation combined with chilled ceilings although not quite as high as the maximum design values suggested elsewhere [8,18].

Although there are a significant number of studies of displacement ventilation reported, there are relatively few that have been concerned with this form of ventilation combined with chilled ceilings. A limited set of vertical temperature distribution data for rooms using chilled ceiling panels and displacement ventilation were reported by Skåret [15], Behne [19], Krühne [20], Külpmann [21] and Alamdari [8]. These early studies show reduced temperature gradients compared to displacement ventilation alone, and in some cases, only small deviations from the average room temperature. Behne carried out more extensive measurements later [18] and we comment on this work below.

A number of fluid flow features found in these systems, such as inlet and plume flows, are common to purely displacement ventilation but also certain forms of natural ventilation [22] and are of some relevance to this study. Magnier et al. [14] have studied inlet conditions and those near the floor in some detail and verified the similarity to gravity current conditions. Theoretical studies of natural ventilation have been concerned with plume growth and the development of the upper warm layer dividing the room at the horizontal stationary front in both steady state [23,24] and transient conditions [25]. Although such studies, and related physical modeling studies, consider the development of the upper warm layer adjacent the ceiling they do not consider ceiling heat transfer conditions. Theoretical models have been developed for situations with a chilled ceiling [26,27] but make assumptions about heat transfer coefficients and turbulent conditions at the ceiling.

The characteristics of plumes, particularly those around the human body, are of general interest in ventilation flows and the development of breathing thermal manikins has enabled detailed measurements to be made. Such measurements have been reported recently by Zukowska et al. [28,29] and Rim et al. [30]. They point out the sensitivity of the plume to features such as hair, clothing and chair shape, the slightly unsymmetric nature of plumes from seated persons and also the influence of the breathing process. Similar detailed study of plumes from office equipment have not been reported to date.

A few studies have been made of convection heat transfer conditions in rooms with displacement ventilation and chilled ceilings. The work reported by Novoselac et al. [31] was mostly concerned with floor heat transfer in rooms with displacement ventilation alone—the floor gravity current being rather different to that in other forms of mechanical ventilation. The studies by Jeong et al. [32] examined existing correlations for ceilings and the study by Karadağ [33] was based on numerical convection heat transfer results and theoretical radiant heat transfer models rather than experimental measurements. Andrés-Chicote et al. made an experimental study of convection from a chilled ceiling in a full size test room but without any plume heat sources. Causone et al. [34] also carried out full scale experiments in an unventilated test chamber. They comment on some systematic differences in measured ceiling convection coefficients when compared to other work and suggest this may be due to different reference temperature definitions but they also point out some systematic differences between cases with heated walls as the heat source and cases with plume heat sources. The physical modeling study by Thomas et al. [35], which employed an unventilated heated water tank with a small plume, did not quantify convection heat transfer coefficients but did highlight the existence of Rayleigh-Bérnard type convection cells close to the ceiling surface under steady state conditions. These cells seem to have been confined to a relatively shallow layer next to the ceiling surface and not induced large scale mixing in the upper part of the room. The authors suggest further experiments to investigate to what extent the Rayleigh-Bérnard cells at the surface would be influenced by other features or turbulent structures in the flow.

A concern that has often been commented upon is that the air quality (ventilation effectiveness) benefits of displacement ventilation may be negated, in part, by additional recirculation induced by any chilled ceiling. Krühne [20] reported some detrimental effects when the chilled ceiling carried some of the thermal load but these experiments were with relatively low heat gains ($\sim$20 W/m^2). Behne [18,19] has carried out a more comprehensive study using tracer gas and temperature profile measurements. It was shown that temperature gradients were small and ventilation effectiveness close to unity in the upper part of the room when the chilled ceiling operated and that conditions in the lower part of the room also varied depending on how much load was apportioned to the chilled ceiling. Three reasons were suggested for the increased recirculation and reduced air quality:

(i) natural convective cooling at the ceiling surface causing the air to fall and mix with the air below;
(ii) radiant heat exchange causing cooling of the upper walls and so inducing negatively buoyant wall jets;
(iii) buoyant flow in the rising plumes being of greater volume flow rate than that supplied by the displacement ventilation system.

Behne [18] acknowledges that these possible effects are interrelated and so “it is hard to tell which influences or causes the other”. It was suggested that a good design metric was the proportion of the load shared by the chilled ceiling system—higher values indicating smaller temperature gradients but more mixing and so poorer air quality—and this suggests lower ceiling temperatures are responsible for the increased recirculation. This idea was endorsed in [10] and design charts proposed on this basis. However, the cause and effect relationship between the proportion of load carried by the ceiling and the degree of mixing is not clear. In particular, the division of the load between the air and radiant systems depends not just on ceiling temperature but also on the air flow rate. The air flow rate, as well as the number and strength of the plumes in the room, are directly related to the position of the stationary front. The stationary front, and hence size of the upper mixed air layer, is determined according to the requirement for conservation of mass and does not depend directly on heat transfer processes [23]. The reason for the impaired air quality, relative to pure displacement ventilation, is often stated (e.g. [8].) to be the first of those given by Behne; natural convection and negative buoyancy driven by the cool ceiling surface. We address this question later.

The experiments reported here were carried out as part of a project to develop a nodal model of displacement ventilation and chilled ceiling systems for use in annual energy simulation [36]. The nodal model in question required detailed measurements of air and surface temperatures over a range of operating conditions for the