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EVALUATION OF DISPLACEMENT VENTILATION FOR HIGH-CEILING AREAS

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ABSTRACT

A study is being conducted to assess the performance of displacement ventilation in high-ceiling areas such as commercial and industrial manufacturing facilities. These areas, which can range from 5 to 20 meters in height, often feature high internal heat loads and contaminants associated with heat sources. Very little performance data exists for displacement ventilation installations in high-ceiling areas, particularly any which account for the influence of wall temperature.

In this ongoing study, several experiments are being conducted in a room equipped with both a high ceiling (6.5 meters) and a displacement ventilation system. In addition to traditional person, computer, and lighting loads found in offices, the study also evaluates system performance with higher intensity manufacturing-inspired loads. The performance of the system has been evaluated by use of air temperature, tracer gas, and velocity measurements as well as wall, floor, and ceiling temperature measurements. Results of the measured data will be used to validate a CFD program previously validated for small offices and classrooms with a ceiling height of 2.4 meters. These collected data will also help to extend the applicability of current displacement ventilation guidelines to buildings with higher ceilings and higher intensity internal heat loads.

Initial results of two cases are presented here – one, a cubicle-style office and the other, an open-plan office without partitions. The displacement system performed well in both cases. Measurements show that the walls in this high space induce large convective air flows and contribute greatly to the radiant heat load in the occupied zone.

KEYWORDS

Displacement Ventilation, Design Guidelines, Office Ventilation, High Ceiling, Wall Temperatures.

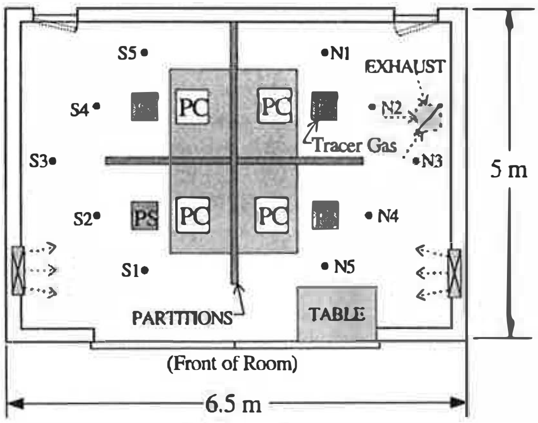
INTRODUCTION

Displacement ventilation has been in use for several decad‹'s with its first uses in industrial settings - many involving welding. Such applications are well suited for displacement as hot contaminant sources are commonplace, ilings are often high, and draught sensitivity of occupants is less of a concern. The popularity of displacement ventilation has grown most recently for comfort cooling applications, where an increased ventilating efficiency and higher air supply temperatures have proved advantageous over mixing ventilation in many installations. A great deal of attention has since been directed toward the development of design guidelines for displacement systems in a variety of comfort cooling applications, such as offices, classrooms, and restaurants. These spaces, however, typically feature relatively low ceiling heights in the range of 2.4 to 3.3 m. Several researchers including Skistad (1994) have concluded that displacement ventilation is more suitable for high spaces such as concert halls and workshops. Currently, very little performance data exist to test the validity of design guidelines or CFD simulations of high spaces.

This study is evaluating the performance of displacement ventilation in a room equipped with a high ceiling (6.5 m) under a variety of internal loads. Several cases have been tested thus far including some which replicate light manufacturing areas, computer rooms, and common office layouts. Due to space limitations, only the results from two office scenarios are presented here. Comparisons will be made to design rules for low ceiling spaces developed earlier at MIT (Chen, Glicksman 1998). To obtain the following results, extensive wall, floor, and ceiling temperatures were recorded in addition to air temperature, tracer gas concentration, and air velocity measurements at selected positions throughout the space.

TEST ROOM DESCRIPTION

The "test" room shown in Figure 1 below is a conference room belonging to MIT's Department of Architecture. In a recent renovation, it was outfitted with a variable-air-volume displacement system which for test purposes was set to provide a fixed 3 air changes per hour.



The test room shown on the left is an interior space with no outside walls or windows. The supply air is delivered from two opposing diffusers, while a single exhaust grille is located overhead to the right at a height of 5.0 m. The room is 6.5 m high. Fluorescent lighting and sound-absorbent ceiling panels are located 3.8 m above the floor. Tracer gas was injected 1.1 m above the floor directly atop the person simulator shown.

Figure 1: Test Room Layout and Description

The floor of the room is carpeted, while the walls are common gypsum board, painted white. Sound-absorbent panels (painted perforated sheet metal backed with fiberglass insulation) are suspended from the room's ceiling and comprise roughly 60% of the room's area at the 3.8 m elevation. Ductwork, building utilities, and the room's exhaust reside in the space between these panels and the ceiling.

TEST CASES

The following two popular office design scenarios were tested: the first we'll refer to as the *cubicle-style office* (CSO), the other will be titled the *open-plan office* (OPO). In both cases, individual offices were modeled by a seated occupant (PS), and a personal computer simulator (PC) located atop a 1.22 m x 0.86 m desk. For ease of comparison between these two cases, the loads, geometry, and ten data collection points were kept as similar as possible. The only physical difference between these two tests were the installation of partition walls between the four individual office areas, as shown in Figure 1. These partitions were 1.68 m in height and did not allow airflow to pass beneath them. Table 1 below lists the loads used for each office case. It includes occupant loads (Q_o) modeled with (4) 75 W person simulators, equipment loads (Q_e) modeled with (4) 90 W PC simulators, a lighting load (Q_l) of 732 W generated by overhead fluorescent lighting, and a transmitted solar radiation load (Q_{slr}) of 0 W. In each test case the load from the building envelope (Q_{wl}) was found subsequent to each test through convective and radiant heat transfer calculations based on wall, floor and ceiling temperatures.

TABLE 1
LOADS USED FOR BOTH CASES

Case	n ach	Height m	Area m^2	Q_o / A W/m^2	Q_e / A W/m^2	Q_l / A W/m^2	Q_{slr} / A W/m^2	Q_{wl} / A W/m^2	Q_{tot} / A W/m^2
CSO	3.17	6.5	31.6	9.5	16.1*	23.2	0	18.3	67.1
OPO	3.17	6.5	31.6	9.5	11.4	23.2	0	17.8	61.9

* 4.7 W/m^2 was added here to account for radiative heat absorption of the cubicle partitions

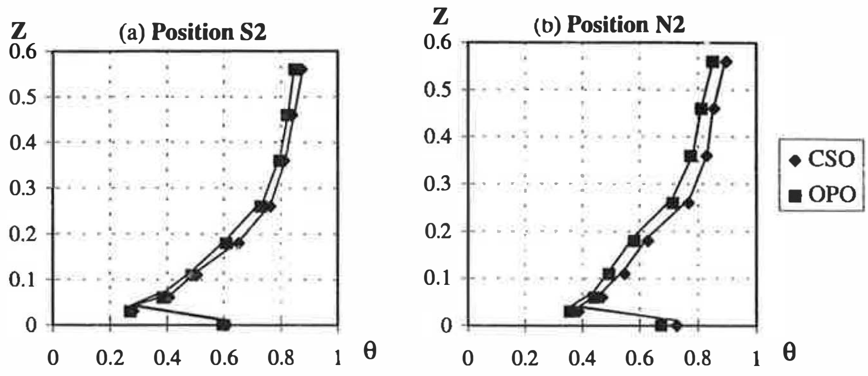
MEASUREMENTS

Table 2 highlights general measurement results. Air temperature measurements were obtained by use of 0.6mm dia. copper-constantan thermocouples with a ± 0.2 °C accuracy which were coated with an aluminum flake-based paint. This highly reflective coating helped to minimize temperature measurement disparities caused by any nearby radiant heat sources. Air velocity measurements, used here to evaluate draft risk, were taken with Sensor HT-426 Thermoanemometer Transducers. All surface temperature measurements were taken with an Exergen Model D501 Infrared Thermometer. Contaminant concentrations were obtained by the injection of a 1% SF_6 tracer gas at 863 standard cubic centimeters per minute (sccm). The room was slightly pressurized during both of these tests.

TABLE 2
GENERAL MEASUREMENT RESULTS

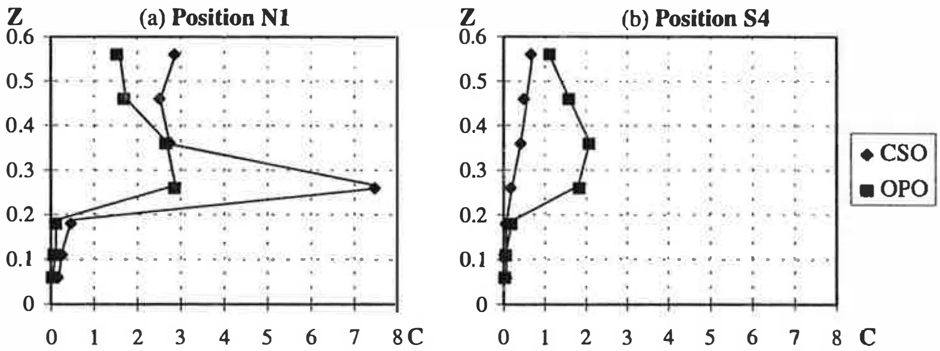
Case	T_{supply} (C)	$T_{exhaust}$ (C)	C_{supply} (ppm)	$C_{exhaust}$ (ppm)	u_{supply} (m/s)	$\dot{V}$ (l/s)
CSO	15.50	23.59	0.058	0.661	0.11	181.3
OPO	15.40	23.61	0.071	0.840	0.11	181.3

Figures 2a and 2b show comparative air temperature measurements between both office cases at locations N2 and S2. These locations are shown on the room layout in Figure 1. Both figures show a change in the temperature gradient above the Z=0.20 elevation.



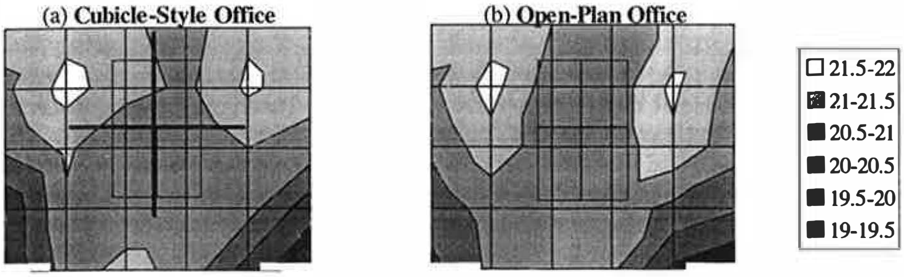
Figures 2a and 2b: Normalized Air Temperature Results
$\theta = (T - T_{supply}) / (T_{exhaust} - T_{supply})$, $Z = z / H_{room}$

Figures 3a and 3b show tracer gas concentrations at positions N1 and S4, respectively. These figures indicate that the stratification level occurs near the Z=0.22 elevation. Note that the partition height is Z=0.26. With the tracer gas injected atop the person simulator nearest measurement point N1, it becomes apparent that the partition walls act to confine the tracer gas within one office.



Figures 3a and 3b: Normalized Tracer Gas Concentration Results
$C = (c - c_{supply}) / (c_{exhaust} - c_{supply})$, $Z = z / H_{room}$

Delivery of supply air along the floor was not apparently hampered by the partitions in the CSO case. No noticeable differences can be discerned between Figures 4a and 4b, which show the floor temperature distribution for each case. In both cases, elevated floor temperatures toward the rear of the room were due in part to a relatively warm rear wall.



Figures 4a and 4b: Floor Temperature Distribution (°C)

ANALYSIS

Figures 2a and 2b clearly show that a constant temperature gradient would be a poor assumption for these cases. A design guideline which does not make this assumption has been developed recently (Chen, Glicksman 1998) and validated for spaces with lower ceilings. Weighting coefficients are applied to the three primary sources of heat transfer to the air layer between a sedentary occupant's head and foot levels. These sources determine the temperature rise (ΔT_{hf}) in this layer and are identified in Eqn. 1 below as Q_{oe} from occupants and equipment, Q_l from overhead lighting, and Q_{ex} from the building envelope.

$$n = \frac{1}{\Delta T_{hf} \rho C_p H A}\left(a_{oe}Q_{oe} + a_l Q_l + a_{ex}Q_{ex}\right) \qquad (1)$$

where: $a_{oe} = 0.295$ | $a_l = 0.132$ | $a_{ex} = 0.185$ | n = air changes per hour
H = space height | A = floor area | ρ = air density | C_p = const. press. specific heat

By rearranging Eqn. 1 and solving for ΔT_{hf}, given n = 3.17 ach measured for both test cases, we generate the results shown in Table 3 below. A 17% discrepancy between measured and calculated ΔT_{hf} values shows good agreement for the open-plan office case. A 19% discrepancy is observed for the cubicle-style offices once radiant influences upon the partitions from the room walls, person simulators, and PC simulators are accounted for.

A simplified formula to predict the dimensionless temperature rise (θ_f) of the supply air due to floor convective heat transfer was proposed by Mundt (1992) and is listed as Eqn. 2 below. Table 3 compares the measured (θ_f) values to those calculated using α_r = 4.73 W/m²K and α_{cf} = 4 W/m²K.

$$\theta_f = \frac{(T_{nearfloor} - T_{supply})}{(T_{exhaust} - T_{supply})} = \frac{1}{\frac{\dot{V}\rho C_p}{A}\left(\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}\right) + 1} \qquad (2)$$

TABLE 3
CALCULATED VS. MEASURED VALUES

Case	ΔT_{hf} (calc)	ΔT_{hf} (meas)	Θ_f (calc)	Θ_f (meas)	H_s (calc)	H_s (meas)
CSO	1.97 °C	2.42 °C	0.23	0.32	1.21 m	~1.44 m
OPO	1.76 °C	2.13 °C	0.23	0.33	1.20 m	~1.44 m

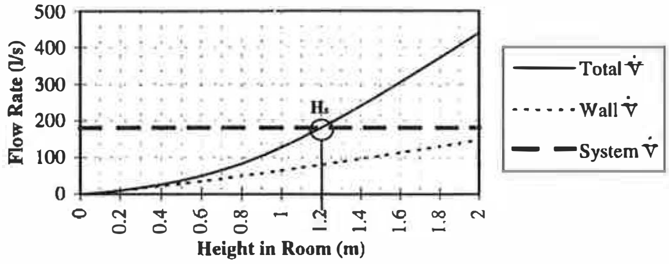


Figure 5: Volumetric Flow Rate of Room Air Supply vs. Convective Plume Flow Rate

The calculated θ_f value is noticeably less than the measured value for each case. This discrepancy results from the large influence of wall to floor radiative heat transfer for high spaces. In these cases, 70% of the radiative heat transfer from the building envelope to the floor originates from the walls. Eqn. 2 is based on the assumption that 100% of the radiation to the floor comes from the ceiling.

The walls also contribute substantially to the total plume flow rate in the occupied zone. Figure 5 above shows the total calculated plume flow rate against height within the room for the CSO case. Expressions for plume volume flow by Mundt (1992) were used to model the PCs and person simulators while an equation by Nielsen (1993) was used to model wall plumes. The point where these calculated flow rates intersect with the room's supply flow rate provides an estimation of the stratification height (H_s). This value is listed in Table 3 along with an actual H_s taken by inspection from tracer gas results.

CONCLUSIONS

A constant temperature gradient from floor to ceiling is a poor assumption to make for offices in a high ceiling space. Eqn. 1, developed for rooms with low ceilings, seems to work well for high spaces - as demonstrated by good agreement between predicted and measured values for ΔT_{hf} in both office cases. The large amount of surface area added to the occupied zone by the office partitions likely contributed to the greater ΔT_{hf} measurement obtained in the cubicle-style office test. One significant difference posed by high spaces is the stronger influence of wall temperature, both upon radiative heat transfer from wall to floor and upon free convective plume flows. Finally, substantial floor obstructions (office partitions in this case) appear to have little impact on the displacement system's ability to cool areas not within direct sight of a supply diffuser.

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