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Simulated and measured performance of displacement ventilation systems in large rooms

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ABSTRACT

Displacement ventilation (DV) systems were initially developed as an efficient buoyant pollutant removal strategy for Scandinavian industrial halls in the 1970s. In the following decades these systems started to be used in mechanical cooling of office buildings and auditoriums. Designing displacement ventilation systems is more challenging than conventional overhead mixing systems. Most DV system designs require simplified modeling tools. Existing simplified models of DV were validated using air temperature measurements performed in test cells that cannot reproduce the conditions that exist in large rooms with thermally active boundary conditions. There is a lack of measurements that investigate the performance of DV systems in occupied large rooms. With the goal of reducing this knowledge gap, this paper presents a set of detailed temperature and CO_2 measurements in two occupied large rooms with recently designed DV systems. The measurements were performed in two recently refurbished rooms located in Lisbon: a large Concert hall and an adjacent Orchestra rehearsal room. The measurements and subsequent analysis were used to assess the actual performance of large room, state of the art, DV systems. In addition, these measurements were used to determine the modeling error of the three-node DV model implemented in EnergyPlus when simulating large rooms. Comparison between simulations and measurements revealed a good agreement: the average simulation error obtained by averaging the error of all measurements in all temperature nodes is 5.9%, with the largest deviation occurring in the floor level node (7.1% ≈ 0.4 °C) average simulation error of 5.9% (the average of error of all measurements in all nodes).

1. Introduction

In the last decades the increasing amount of time that people spend in service buildings equipped with heating, ventilation and air conditioning systems (HVAC) led to a significant increase in HVAC related energy consumption. In non-domestic or service buildings HVAC accounts for up to 50% of the total energy consumed [1]. Due to improved thermal insulation and increased internal gains, HVAC systems are predominantly used to provide cooling and fresh air [2]. To minimize environmental impact, these systems must operate with the lowest energy consumption possible. In air based HVAC systems the room airflow distribution strategy (ADS) can have a large impact in overall system efficiency [3]. For a given room and internal load, different ADS's will use different inflow temperatures, amounts of outdoor air, and, consequently will have different cooling energy requirements [4].

The most commonly used ADS is overhead mixing ventilation (MV). In this strategy, fresh air is supplied through the upper part of the room at a temperature of 16 °C or less [5]. This approach mixes any high-level heat loads into the occupied zone promoting uniform air conditions throughout the whole space. For some heat load profiles and room geometries Displacement Ventilation (DV) can be an efficient alternative to overhead MV. In DV systems air is inserted at low level, spreading near the floor until it reaches a heat source where it expands and rises as a thermal plume. Heat and pollutants accumulate in the upper zone of the room, forming a nearly mixed air layer whose lower boundary is known as the interface height (h_i [6]). In these systems the room air outlet must be located above the interface height. In some DV rooms the proximity between air supply and occupants can result in discomfort due to cold air draft, particularly near the occupant feet and ankles. To control these problems the air supply velocity, V_{in}, must be low and the inflow air temperature, T_{in}, is higher

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Nomenclature

A	Area (m^2)
A_c	Ceiling surface area (m^2)
ADS	Room airflow distribution strategy
A_f	Floor surface area (m^2)
A_t	Sum of all surfaces area (m^2)
Avg. Bias	Averaged Bias
Avg. Dif	Average Difference
Avg. Error	Average Error
A_{wl}	Lateral area exposed to the lower zone surface area (m^2)
A_{wu}	Lateral area exposed to the upper zone surface area (m^2)
C_e	CO_2 concentration in the exhaust zone (ppm)
CFD	Computational fluid dynamics
CGF	Calouste Gulbenkian Foundation
CO_2	Carbon dioxide
C_{oz}	CO_2 concentration in the occupied zone (ppm)
C_p	Thermal capacity of air at constant p (J/(kg K))
C_s	CO_2 concentration in the supply zone (ppm)
DV	Displacement ventilation
F	Inlet flow rate (m^3/s)
F_{GC}	Fraction of total heat gains that are convective
F_{GR}	Fraction of total heat gains that are radiative
F_{MO}	Fraction of the convective heat gains that is mixed into the occupied zone
G	Buoyancy source (W)
h_c	Heat transfer coefficient of ceiling surface (W/(m^2 K))
h_f	Heat transfer coefficient of floor surface (W/(m^2 K))
hi	*Interface height (m)*
h_{ip}	*Predicted interface height (m)*
h_{im}	*Measured interface height (m)*
h_{rc}	Radiative heat transfer coefficient of ceiling surface (W/(m^2 K))
h_{rf}	Radiative heat transfer coefficient of floor surface (W/(m^2 K))
h_{rwl}	Radiative heat transfer coefficient of the lateral surface that is bellow the mixed layer (W/(m^2 K))
h_{rwu}	Radiative heat transfer coefficient of the lateral surface that is above the mixed layer (W/(m^2 K))
h_{TMX}	Room height where zero temperature gradient region begins
HVAC	Heating ventilation and air conditioning
h_{wl}	Heat transfer coefficient of the lateral surface that is bellow the mixed layer (W/(m^2 K))
h_{wu}	Heat transfer coefficient of the lateral surface that is above the mixed layer (W/(m^2 K))
IAQ	Indoor air quality
I_M	Inflow degree of mixing
l	Plume length(m)
MV	Mixing ventilation
T_{Af}	Temperature of room air in the horizontal layer adjacent to the room floor (^{0}C)
T_c	Temperature of ceiling surface (^{0}C)
T_f	Temperature of floor surface (^{0}C)
T_{in}	Inflow air temperature (^{0}C)
T_{MX}	Temperature of mixed layer node (^{0}C)
T_{OC}	Temperature of room air in the occupied zone (^{0}C)
T_{wl}	Temperature of lateral surface that is bellow the mixed layer (^{0}C)
T_{wu}	Temperature of lateral surface that is above the mixed layer (^{0}C)
Tz	Room air temperature at z height (m)
T_{z0}	Temperature of inflow air (^{0}C)
T_{ztotal}	Room exhaust air temperature (^{0}C)
UFAD	Underfloor air distribution
V_{in}	Air supply velocity (m/s)
w	Plume width (m)
X_{12}	Distance between two thermal plumes (m)
Z_0	First measurement point height (m)
z_{total}	Total room height (m)
$Z1_0$	Virtual origin of plume number 1
$Z2_0$	Virtual origin of plume number 2
Δt	Temperature difference
ε_p	*CO_2 removal efficiency*
ρ	Air density (Kg/m^3

than in MV systems (V_{in} <0.2 m/s [7], T_{in} >18-19 °C [8,9]).

DV systems were initially developed as an efficient buoyant pollutant removal strategy for use in Scandinavian industrial halls (in the 70́s [10]). In the 80's these systems started to be used in mechanical cooling of office buildings, with the goal of reducing room airflow velocities, ventilation induced noise, and HVAC energy consumption [11]. Despite these possible advantages, the application of DV in office buildings is limited by lack of space to install diffusers in the floor or office walls, combined with the obstacles created by office furniture. These space restrictions limit the airflow rate, leading to a maximum space sensible cooling capacity of 25-35 W/m^2 [12, 13, 14] (see equation (1) and Fig. 1). In contrast, in auditoriums and large lecture rooms there is more space available to install inflow air diffusers under the seats, and the fresh inflow air is supplied directly to the occupants (see Fig. 1), leading to a much higher sensible cooling capacity (≈180 W/m^2, up to five times more than in office buildings DV system, see equation (2)). Imposing sensible heat conservation on the two rooms shown in Fig. 1 we obtain:

$$\text{Offices}: \frac{F \times \rho \times Cp \times \Delta t}{A} = \frac{0.03 \times 1.2 \times 1000 \times 4}{(2 \times 2)} = 37W/m^2 \tag{1}$$

$$\text{Large rooms}: \frac{F \times \rho \times Cp \times \Delta t}{A} = \frac{0.03 \times 1.2 \times 1000 \times 4}{(0.9 \times 0.9)} = 181W/m^2 \tag{2}$$

Most large rooms and auditoriums are characterized by variable non-permanent use, with occupancy levels ranging from nearly empty to completely full. In most rooms, the non-permanent use makes HVAC related energy consumption a secondary priority. Instead, designers focus on achieving high thermal comfort and low HVAC noise levels. In some room configurations these goals can only be achieved with the low velocity and low noise supply that characterizes modern DV systems.

However, designing DV systems is more challenging than