Energy and Buildings 108 (2015) 44–54

Contents lists available at ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

Simplified modeling of displacement ventilation systems with chilled ceilings

Nuno M. Mateus*, Guilherme Carrilho da Graça

Instituto Dom Luiz, Faculdade de Ciências, Universidade de Lisboa, 1749-016 Lisboa, Portugal

ARTICLE INFO

Article history:
Received 29 May 2015
Received in revised form 28 August 2015
Accepted 30 August 2015
Available online 2 September 2015

Keywords:
Displacement ventilation
Chilled ceilings
Neutral height
Model validation
Design charts

ABSTRACT

Displacement ventilation (DV) flows are more complex than conventional overhead mixing systems since the stratified room environment cannot be modeled using the traditional fully mixed room air approach. A successful DV designer must be able to control the vertical room temperature profile and manage the position of the lower boundary of the upper air layer that contains heat and pollutants. The inclusion of a chilled ceiling (CC) in the DV system increases the complexity by adding the need to manage the CC cooling power so that it does not disrupt the DV stratification. This paper presents the extension of an existing DV nodal model so that the effects of the CC in room airflow and air temperatures can be predicted. The model uses three air nodes and focuses on the thermal plumes as the drivers of the airflow and room air heat exchange. The proposed model is validated using twelve different test chamber configurations from three independent experimental studies. When compared with existing models the proposed model achieves improved precision and model flexibility while using less air nodes. The last section of the paper presents a set of CC/DV design charts that can assist system designers in early design phases.

1. Introduction

Displacement ventilation (DV) systems have been used in mechanical cooling of office buildings since the 80s [1]. In DV systems inflow air is introduced near the room floor with a low velocity and an inflow temperature that is just 4–6 °C lower than the desired comfort temperature, thereby avoiding cold draft discomfort [2,3]. The negatively buoyant inflow air spreads over the room floor until it reaches the heat sources where it expands and rises as a thermal plume. In DV systems the convective heat loads located above the occupied zone are removed in an ideal way, with limited impact in the occupant's thermal comfort. Whenever the room internal gains occur predominantly in the form of plumes, a noticeable interface occurs between the occupied zone of the room and a mixed hot layer near the ceiling of the room. This temperature and contaminant stratification removes heat and pollutants from the occupied zone with high ventilation efficiency [4,5]. In contrast, the more commonly used overhead mixing ventilation systems tend to transport all heat loads into the occupied zone. As a result of a higher inflow to room air temperature difference (up to 15 °C [6]), mixing ventilation systems have more cooling power than conventional DV systems.

Within the HVAC design community, DV is considered to be an effective air distribution strategy for office buildings due to its potential to reduce room air velocities, ventilation induced noise and HVAC energy consumption [7]. In spite of these well-established qualities these systems do not have widespread use due to poor heating performance and limited space cooling capability (25–35 W/m^2 [8–10]). Continuous improvement in building envelope insulation has greatly decreased the need for space heating. Still, in many office buildings the sensible cooling loads often exceeds 50 W/m^2: a load that conventional DV systems cannot meet. To overcome this cooling limitation the HVAC design and research community has developed two DV system variants: under floor air distribution (UFAD [11,12]) and the combination of DV with chilled ceiling systems (CC/DV [13,14]). In UFAD systems air is inserted into the room using swirl diffusers supplied by an under floor plenum. These diffusers induce more mixing than standard low velocity DV diffusers, allowing for a higher differential between inflow and room air temperature difference (10 °C) and, consequently, higher cooling capacity [15]. In the CC/DV approach, shown in Fig. 1, DV inflow air removes the latent loads and a portion of the sensible load, while the CC system removes the remaining sensible load (mostly by radiative heat transfer). With this combined

* Corresponding author.
E-mail address: namateus@fc.ul.pt (N.M. Mateus).

http://dx.doi.org/10.1016/j.enbuild.2015.08.054
0378-7788/

Nomenclature

DV displacement ventilation
CC chilled ceiling
CC/DV displacement ventilation system with chilled ceilings
HVAC heating, ventilation, and air conditioning
UFAD under floor air distribution
CFD computational fluid dynamics
θ adimensional temperature
T temperature (°C)
T_{in} temperature of inflow air (°C)
T_{out} room exhaust air temperature (°C)
z^* adimensional height (m)
z height (m)
z_{total} total room height (m)
F inlet flow rate (m^3/s)
α plume entrainment constant
g acceleration of gravity (m/s^2)
β coefficient of thermal expansion (K^{-1})
W heat flux plume (W)
h neutral height (m)
ρ air density (kg/m^3)
C_p thermal capacity of air at constant p ($W\,m^3/(kg\,K)$)
n number of thermal plumes
Z_0 virtual origin of thermal plume
T_{OC} temperature of room air in the occupied zone (°C)
T_f temperature of floor surface (°C)
T_{Af} temperature of room air in the horizontal layer adjacent to the room floor (°C)
T_{wl} temperature of lateral surface that is below the mixed layer (°C)
T_{wu} temperature of lateral surface that is above the mixed layer (°C)
T_{MX} temperature of mixed layer node (°C)
T_c temperature of ceiling surface (°C)
T_{in} inflow air temperature (°C)
T_{CC} chilled ceiling surface temperature (°C)
T_{NCC} non-chilled part of ceiling surface temperature (°C)
A_f floor surface area (m^2)
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)
A_c ceiling surface area (m^2)
A_{CC} chilled ceiling surface area (m^2)
A_{NCC} non-chilled part of ceiling surface area (m^2)
A_t total area (m^2)
h_f heat transfer coefficient of floor surface ($W/(m\,K)$)
h_{wl} heat transfer coefficient of the lateral surface that is below the mixed layer ($W/(m\,K)$)
h_{wu} Heat transfer coefficient of the lateral surface that is above the mixed layer ($W/(m\,K)$)
h_c Heat transfer coefficient of ceiling surface ($W/(m\,K)$)
h_{rc} radiative heat transfer coefficient of ceiling surface ($W/(m\,K)$)
h_{Rf} radiative heat transfer coefficient of floor surface ($W/(m\,K)$)
h_{rwl} radiative heat transfer coefficient of the lateral surface that is below the mixed layer ($W/(m\,K)$)
h_{rwu} radiative heat transfer coefficient of the lateral surface that is above the mixed layer ($W/(m\,K)$)
G total internal heat gains (W)
F_{MO} fraction of the convective heat gains that is mixed into the occupied zone
F_{GC} fraction of total heat gains that are convective
F_{GR} fraction of total heat gains that are radiative
I_M inflow degree of mixing
Sim. simulation result
Meas. measurement result
Avg. error average error
Avg. dif. average difference
Avg. bias averaged bias
h_{TMX} room height where zero temperature gradient region begins
R cooling loads ratio
Q_{DV} portion of the total sensible gains that is removed by displacement ventilation

approach the cooling capacity can reach 100 W/m^2 [16,17] while maintaining the use of standard low velocity DV diffusers. In a successful CC/DV system the CC is able to increase the cooling power without compromising the stratified DV flow. This paper focuses on the combined CC/DV approach.

Design of stratified ventilation systems is more complex than conventional overhead mixing systems since the perfectly mixed room air approximation is not adequate to model the internal conditions. The main goal in DV modeling is to predict the vertical temperature gradient in the room and also manage the position of the lower boundary of the upper layer of room air (where indoor pollutants are concentrated). These seemingly simple tasks present a great challenge since many flow and room geometry features contribute to the stratification: room height, airflow rate and temperature, type, location and strength of the buoyancy sources. Accurate prediction of the vertical temperature gradient is key for fine tuning of system design and sizing as well as accurate predictions of energy consumption and thermal comfort. The inclusion of a CC system increases the complexity of the design by adding the need to use its cooling power without compromising the stratified environment of the DV system. An excessively low CC surface temperature can destroy the stratification and even create condensation in the CC surface [18]. The stratification is disrupted when the upper mixed layer temperature approaches the occupied zone temperature, effectively creating a mixed room air environment. Condensation occurs when the CC surface temperature drops below the dew-point temperature. This problem that is more likely to occur in rooms with a high portion of latent heat gains [14].

To size and predict energy consumption of CC/DV systems designers can use three approaches: simplified design methods [19,20], computational fluid dynamics simulations (CFD, [21]) and room air stratification models implemented in dynamic thermal simulation tools. Simplified sizing methods are useful in early design but cannot predict whole year energy consumption (a common requirement for building energy certification [22]). Reynolds averaged Navier–Stokes CFD simulations are increasingly used in design but remain a computationally heavy tool for CC/DV design analysis and annual simulations. Further, in these applications, CFD is hampered by errors in the estimation of mixed convection that have been known for a long time [23] but are still being resolved [24,25]. Further, designers are often faced with the need to simulate multiple rooms as well as several building occupation and outside weather scenarios. For these purposes CFD is still very time consuming, both in terms of simulation setup and runtime for yearly simulations. In this context, simplified models implemented