AN EXTENDED MODEL FOR UNDER FLOOR AIR DISTRIBUTION

ABSTRACT

Previous work on an Underfloor Air Distribution (UFAD) system with a single heat source and a single cooling diffuser at floor level developed by Lin (2003) has been extended to study the effects of the vertical location of the heat source and of multiple cooling diffusers. This is an attempt to produce more realistic models of UFAD systems. Both experimental and theoretical modeling are described in this paper. In the experiments, a plume and fountains represent a heat source and cooling diffusers, respectively. The experiments suggest how the entrainment rate of ambient fluid into the fountain at the interface changes with the local Richardson number. They show that for a fixed ventilation rate, as the source elevation is increased, the return temperature remains unchanged and the height of the cooler, lower zone increases. The effect of multiple cooling diffusers is to decrease the height and the temperature of the cooler zone for a fixed ventilation rate, and to cool both layers for a fixed under floor plenum pressure.

INTRODUCTION

Underfloor Air Distribution (UFAD) technology is currently experiencing a rapid growth in North America because it offers a broad range of important benefits over conventional ceiling-based air distribution (CBE 2002). Well-designed UFAD systems can reduce life-cycle building costs; improve thermal comfort; improve ventilation efficiency, indoor air quality; conserve energy; and reduce floor-to-floor height. However, engineers and designers lack information about UFAD systems because the technology is still in its infancy, and standardized methods and guidelines are under further development (Bauman et al. 2001)

The goal of this research is to help UFAD technology achieve its full potential by providing a sound theoretical understanding of the behavior of UFAD systems to enable the design of UFAD systems that are energy efficient, and effective in their performance.

Lin (2003) developed an experimental and theoretical model for a simplified UFAD system. In his model, a single heat source, represented by a plume and a single cooling diffuser, represented by a fountain, are located at floor level. Small-scale laboratory experiments were used to model the UFAD system with salt and fresh water used to produce the plume and the fountain. The intensity attenuation of a tracer dye under a constant lighting source was analyzed by visualization software, DigImage (Dalziel 1992-1998), to determine the local density averaged across the width of the tank. Lin noted that the performance of a UFAD system depends on the entrainment of warmer air from the upper zone by the flow from the cooling diffuser, and he suggested a constant entrainment rate E in a range between $0.5 < E < 0.8$. Most of his theoretical predictions agree with his experiment results within 10% (Lin 2003).

Lin's model provides an understanding of the basic UFAD technology, and an efficient experimental method to simulate UFAD systems. However, in reality a UFAD system is much more complicated than Lin's model. Heat sources will not be always on floor level, and there are generally multiple cooling diffusers and heat sources in a UFAD room.

This paper considers the effects of the vertical location of a heat source to simulate a heat source, say, a computer or a lamp on a table, in a UFAD room. We also consider the effects of multiple diffusers. The theoretical UFAD model is also compared with real room test results.

LABORATORY EXPERIMENTS

The salt-bath technique used in this research has been used in buoyancy-driven flows for half a century, since Batchelor (1954) first used it to study the heat convection and buoyancy effects in fluids. Salt solution has a negative buoyancy force in fresh water, which is in contrast to the heat convection problems in building ventilation. However, for the Boussinesq flows, this reversal of the direction of the buoyancy force is unimportant to the dynamics. If salt solution is introduced through a source nozzle at the top of a tank of fresh water, a plume forms in the tank. On the contrary, if fresh water is injected downward into a tank of salt solution, a fountain forms (Lin 2003).

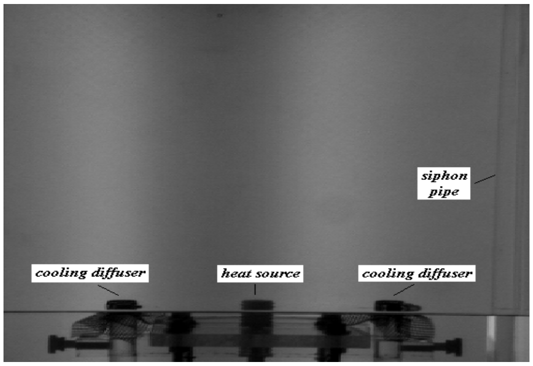

Figure 1 An image of the experiment set-up with one heat source and two cooling diffusers.

Experiments were conducted in a clear Plexiglas tank (with uniform cross-sectional area 30.6cm × 15.3cm and 30.6cm deep) filled with fresh water. A plume whose source is vertically adjustable and one or two cooling diffusers are set up on a mount immersed in the tank. The circular plume nozzle (diameter = 0.5cm) used in the experiments was designed by Dr. Paul Cooper in the Department of Engineering, University of Wollongong, NSW, Australia. This design has a sharp expansion which excites a turbulent flow in a large chamber and, therefore, a turbulent plume is produced at the point of discharge. The cooling vent sources in the experiments are plexiglas pipes (1.27cm inner diameter) with a piece of fine mesh (aperture size about 0.1cm × 0.1cm) wrapped over one end to produce turbulent fountains into the ambient environment. A siphon pipe with an inner diameter 1.57cm was used to ensure a constant volume of fluid in the tank.

Since it has a similar diffusion coefficient as the salt, the dye acts as a tracer for density. Light intensity signals are used to measure the density distribution and investigate the flow pattern evolution.

All experiments were recorded by capturing images with a 4910 series monochrome CCD camera at one-minute intervals through a DT-2862 60Hz frame grabber card into a computer hard drive. Tracing paper was used between the lighting source and the water tank to diffuse the light to make it as uniform as possible. The intensity attenuation of a tracer dye under a constant lighting source was analyzed by visualization software, DigImage, to determine the local density averaged across the width of the tank.

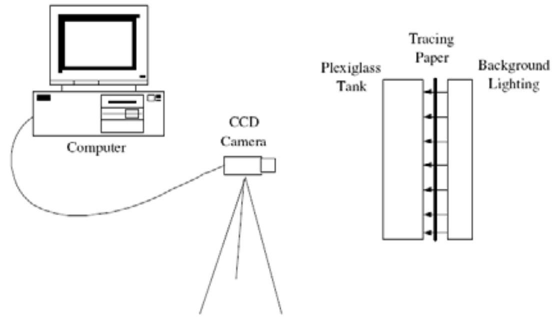

Figure 2 Lighting signals processing set-up.

MODELING AND DISCUSSION

Layered-flow assumption

In this study, we concentrate on the steady state.

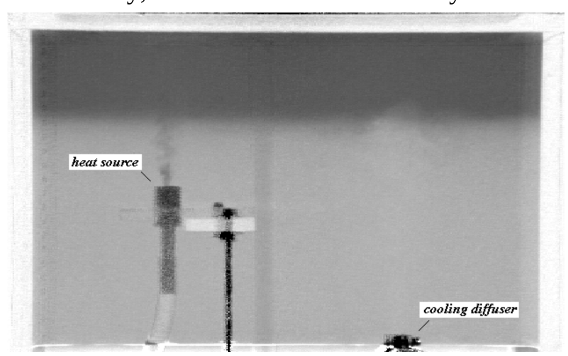

Figure 3 A steady-state image of the experiment with the heat source at ½ room height.

In this case, with no heat conduction or losses/gains through walls, the temperature equation is

$$\frac{DT}{Dt} = u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} = 0, \tag{1}$$

Assume that $\frac{\partial T}{\partial x}$ and $\frac{\partial T}{\partial y}$ are small, as observed in all experiments, then

$$w\frac{\partial T}{\partial z} = 0. \tag{2}$$

This implies that

$$\begin{cases} w = 0, & \frac{\partial T}{\partial z} \neq 0; & (a) \\ w \neq 0, & \frac{\partial T}{\partial z} = 0; & (b) \\ w = 0, & \frac{\partial T}{\partial z} = 0. & (c) \end{cases} \tag{3}$$

In our experiments, except at the interface, $w \neq 0$ as there is a flow in at the bottom and out at the top of the space. Therefore, $\frac{\partial T}{\partial z} = 0$ except at the interface. This suggests layered flows will occur in the steady state, as was observed in all experiments.

The effect of the vertical location of the heat source

Consider a single space in which a plume source with a buoyancy flux B, is at a height h_s above the floor and a cooling diffuser with a volume flux Q_f and a momentum flux M is set on the floor. In this model, the small but finite volume flux supplied from the heat source will be neglected, except in the determination of a virtual origin correction z_v linking the position of an actual plume to an ideal plume (Hunt et al. 2001).

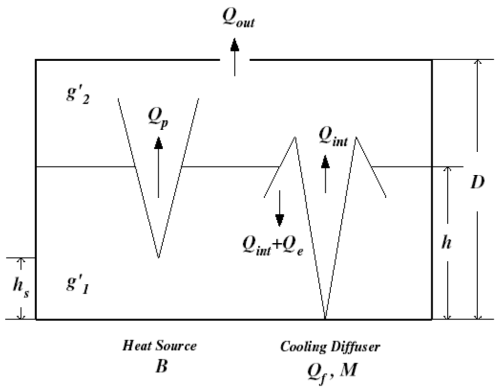

Figure 4 A sketch of a UFAD system with an elevated heat source and a cooling diffuser at floor.

The variables for the model are shown in Figure 4. At an interface height h, the plume and the fountain carry a volume flux Q_p and Q_{int} respectively. An amount Q_e is entrained back into the lower layer by turbulence created by the cooling diffuser, leaving a net flow rate Q_{out} through the system. The interface separates a lower layer of reduced gravity $g_1^{'} = g\frac{(\rho_1 - \rho_f)}{\rho_f}$, and an upper layer of reduced gravity $g_2^{'} = g\frac{(\rho_2 - \rho_f)}{\rho_f}$.

Volume flux conservation applies

$$Q_p = Q_f + Q_e. \tag{4}$$

Buoyancy flux conservation gives

$$g_2^{'} = \frac{B}{Q_{out}} = \frac{B}{Q_f}. \tag{5}$$

The reduced gravity of the lower layer $g_1^{'}$ is determined by the mixture of the fluid ejected from the cooling diffuser and that due to the penetrative entrainment from the upper layer

$$g_1^{'} = \frac{g_2^{'}Q_e + g_f^{'}Q_f}{Q_e + Q_f}. \tag{6}$$

For a self-similar plume, Q_p is determined by

$$Q_p = C\left[B_r(h + z_v - h_s)^5\right]^{1/3}, \tag{7}$$

where $C = \frac{6}{5}\alpha(\frac{9}{10}\alpha)^{\frac{1}{3}}\pi^{\frac{2}{3}}$ is the universal constant and $B_r = B - g_1^{'}Q_s$ is the buoyancy flux of the plume in the environment.

The volume flux of the fountain at interface height can be obtained by solving the differential equations

$$\frac{dQ}{dz} = 2\pi^{1/2}\alpha_f M^{1/2} \tag{8}$$

$$\frac{dM}{dz} = \frac{FQ}{M} \tag{9}$$

$$\frac{dF}{dz} = 0 \tag{10}$$

with the source conditions $Q_o = Q_f$, $M_o = M$ and $F_o = -g_1^{'}Q_f$. $\alpha_f = 0.075 \pm 0.015$ was provided by new experiments with the present sources, which is slightly smaller than $\alpha_f = 0.085 \pm 0.01$ found by Bloomfield and Kerr (1998) for a point source.

It is assumed that

$$Q_e = Q_{\text{int}}E \tag{11}$$

Experiments suggest the entrainment rate of ambient fluid into the fountain at the interface changes with the local Richardson number Ri

$$Ri = \frac{(g_2^{'} - g_1^{'})b_{\text{int}}}{w_{\text{int}}^2}. \tag{12}$$

At low Richardson numbers the entrainment rate tends to be a constant, as found by Lin (2003). At higher Richardson numbers, when the temperature contrast between the upper and lower zones is larger, the entrainment rate is a decreasing function of Ri, and we assume that E is proportional to Ri^{-1}.

The empirical formula for E

$$E = \begin{cases} 0.6 \pm 0.1, & Ri \le 8; \\ 4.8Ri^{-1} \pm 0.1, & Ri \ge 8. \end{cases} \tag{13}$$

is applied throughout this paper.

The equivalent temperature profiles from experimental measurements are compared with the theoretical predictions. For a fixed ventilation flow rate, raising the heat source above the floor increases the depth and decreases the temperature of the lower cooled zone, but leaves the temperature of the upper warmer zone unchanged (see Figure 5).

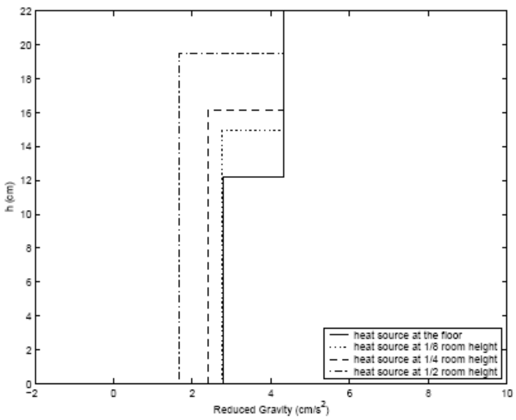

Figure 5 The comparisons of theoretical predictions show the effects of the vertical location of a heat source.

The comparison between the laboratory experiments and the model are shown in Figure 6.

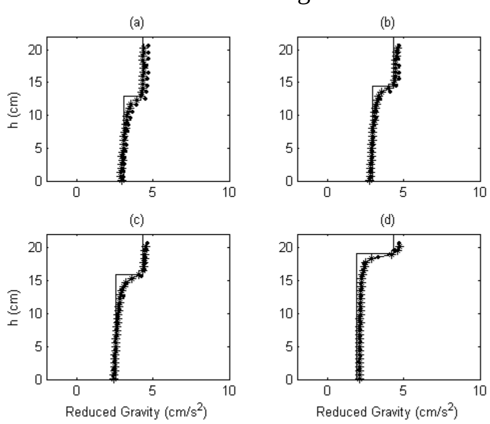

Figure 6 Experimental data agree well with the theoretical prediction. (a) heat source at floor; (b) heat source at 1/8 room height; (c) heat source at ¼ room height; (d) heat source at ½ room height.

Each plot includes two sets of experimental data obtained with different measurement methods. The profiles indicated by "*", are the data analyzed by DigImage software; and the profiles with "." are the measurement data of fluid samples extracted during the experiment at different vertical positions by an Anton Paar density meter. The solid lines in these plots indicate the theoretical predictions. Both theoretical and experimental models show that two-layer temperature stratification forms for all source heights. The agreement between the model and the experimental results is within the uncertainty of the measurements.

The effect of multiple cooling diffusers

We now describe experiments and theoretical modeling of a UFAD system containing multiple cooling diffusers, in an attempt to produce more realistic models of practical problems. The model indicated in Figure 7 indicates a UFAD system with one single heat source and two equal cooling diffusers all at floor level.

We assume that the heat source and the cooling diffusers are separated far enough for them to develop independently. We restrict attention to the case where the cooling diffusers have the same characteristics. It is also assumed, as before, that the volume flux from the heat source $Q_s << Q_f$ is neglected.

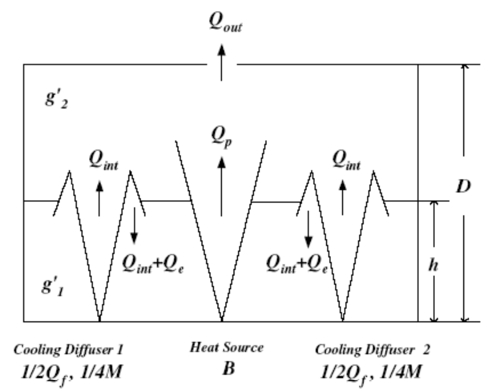

Figure 7 A sketch of a UFAD system with one heat source and two cooling diffusers.

Using similar analysis to that in §3.1, and considering n cooling diffusers,

$$Q_p = n(\frac{1}{n}Q_f + Q_e) \tag{14}$$

$$g_2^{'} = \frac{B}{Q_{out}} = \frac{B}{Q_f} \tag{15}$$

$$g_1^{'} = \frac{g_2^{'}Q_e}{Q_e + \frac{1}{n}Q_f} \tag{16}$$

Since $g_2^{'}$ does not change, and Q_e decreases due to the lower momentum of the fountain, the lower layer temperature $g_1^{'}$ decreases as n increases. When n is infinite, no penetrative entrainment will be observed, because the initial momentum of each fountain is too small. Therefore, $g_1^{'} \to 0$ according to (14), and this limit recovers displacement ventilation.

Given the same conditions (heat load, total ventilation flow rate), but with the supply air divided equally between two diffusers, the occupied zone has a lower temperature and depth than for a single cooling diffuser (see Figure 8).

Table 1. These are experimental inputs for 1 cooling diffuser and 2 cooling diffusers, given the same heat load and total ventilation flow rate.

No. of diffusers	B (cm^4sec^{-3})	Q_f (cm^3sec^{-1})
1	70.11	13.43
2	71.76	13.4

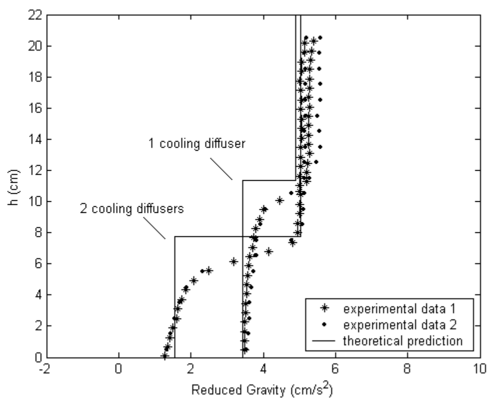

Figure 8 Given the same heat load and ventilation flow rate, both theoretical and experimental data show that the lower layer in 2-diffuser system is cooler and thinner than in single-diffuser system.

Two cooling vents with unequal ventilation flow rates are found to produce a vertical temperature profile consisting of three distinct layers.

The other case is to consider multiple diffusers in a UFAD room with a fixed heat load and a constant pressure under the floor plenum. Instead of $1/2Q_f$ and $1/4M$ at the source of each diffuser in Figure 7, Q_f and M are the source volume flux and source momentum flux, respectively, of each diffuser. Mathematically, (14), (15) and (16) become

$$Q_p = n(Q_f + Q_e) \tag{17}$$

$$g_2^{'} = \frac{B}{Q_{out}} = \frac{B}{nQ_f} \tag{18}$$

$$g_1^{'} = \frac{g_2^{'}Q_e}{Q_e + Q_f} \tag{19}$$

where, Q_e is a different value determined by the interface position.

Apparently, Q_p increases with n. It implies that the interface rises and $g_2^{'}$ decreases. A decreased Q_e due to a higher interface and a smaller $g_2^{'}$ make the lower occupied zone cooler. At the limit when $n \to \infty$, $g_2^{'} \to 0$ according to (18), and $g_1^{'} \to 0$ resulted from (19), thus, the two-layer stratification vanishes, and a uniform temperature as the supply air temperature will be obtained.

Given the same conditions (heat load, underfloor plenum pressure), opening an additional diffuser will cool both layers.

Table 2. These are experimental inputs for 2 cooling diffuser and 3 cooling diffusers, given the same heat load and underfloor pressure.

No. of diffusers	B (cm^4sec^{-3})	Q_f (cm^3sec^{-1})
2	71.76	13.4
3	70.95	19.5

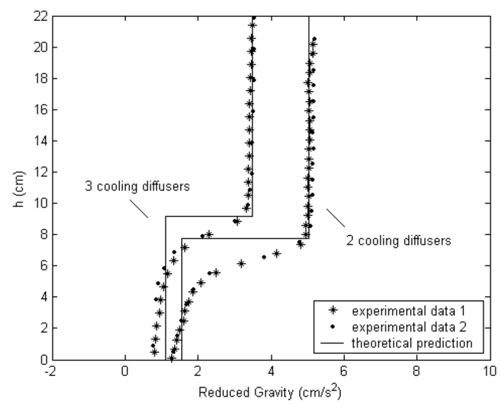

Figure 9 Given the same heat load and underfloor plenum pressure, both theoretical and experimental data show that changing from 2 cooling diffusers to 3 cooling diffusers makes the lower layer cooler and thicker.

Comparisons with real room test

The full-scale room air stratification test chamber operated by York International provides the real room test results, which are compared with our UFAD model.

It is useful in a real situation to divide the number of diffusers up into a number of separate heat sources so that the multiple-diffuser with single-heat source UFAD model can be applied.

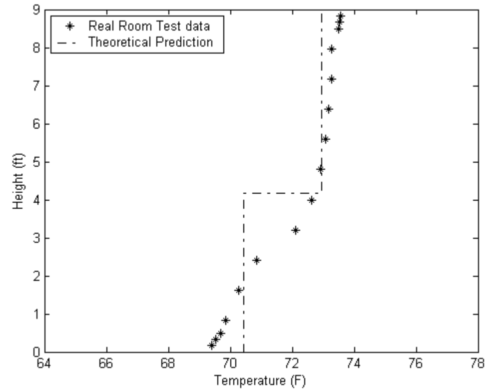

Figure 10 The theoretical UFAD model shows good predictions for real room test.

CONLUSIONS

An analytical model for a UFAD system was extended to provide better understanding of UFAD. Two-layer temperature stratification was assumed in the model and observed in experiments. Theoretical predictions compare well with measurements made in both small-scale laboratory model and full-scale room test.

For a fixed ventilation rate, the source elevation does not affect the return temperature but increases the height of the cooler, lower zone. With a fixed heat load and ventilation flow rate, the effect of multiple cooling diffusers is to decrease the height and the temperature of the cooler zone. With a large amount of cooling diffusers, the model recovers displacement ventilation. In the conditions with a fixed heat load and under floor plenum pressure, the effect of multiple cooling diffusers is to diminish the two-layer stratification by cooling both layer at the same time with different degrees.

ACKNOWLEDGMENTS

The authors wish to thank Dr. C. Caulfield and Dr. Y.P. Lin for their help. This project is sponsored by a grant from the California Energy Commission.

NOMENCLATURE

Subscript

p of a plume;
f of a fountain;
int at the interface;
o at the source;
1 of the lower layer;
2 of the upper layer;
e entrainment;
out out of the chamber;
v virtual;
s of the heat source;

T temperature, K
t time, s
u flow velocity along x direction, cm/s
v flow velocity along y direction, cm/s
w flow velocity along z direction, cm/s
B buoyancy flux at the source of a plume, cm^4/s^3
M momentum flux at the source of a fountain, cm^4/s^2
F_o buoyancy flux at the source of a fountain, cm^4/s^3
Q_f volume flux at the source of a fountain, cm^3/s
Q_p volume flux at the interface of a plume, cm^3/s
Q_{int} volume flux at the interface of a fountain, cm^3/s
Q_e entrainment by a fountain from the upper layer to the lower layer, cm^3/s
Q_{out} volume flux extracted from the ceiling, cm^3/s
h_s height of a plume source, cm
h interface height, cm
D room height, cm
C universal constant for a plume
B_r buoyancy flux of a plume at the source corrected by the lower layer environment, cm^4/s^3
z_v virtual origin of a non-pure plume, cm
α_f entrainment rate for a fountain
g gravitational acceleration, m/s^2
$g_1^{'}$ lower layer reduced gravity, m/s^2
$g_2^{'}$ upper layer reduced gravity, m/s^2
ρ_1 density of the lower layer, g/cm^3
ρ_2 density of the upper layer, g/cm^3
ρ_f reference density (fresh water), g/cm^3
Ri interface Richardson's number
w_{int} vertical velocity of the buoyancy plume plume at the density interface position, cm/s
b_{int} radius of the buoyant plume at the density interface position, cm
E penetrative entrainment rate
n number of diffusers

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