Analytical estimation of optimal minimum airflow for air circulation

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Abstract
Space conditions are directly controlled by terminal boxes in variable air volume (VAV) air handling unit (AHU) systems. The terminal box either modulates airflow or adjusts discharge air temperature. Conditioned space will have thermal discomfort by less air circulation if the airflow and discharge air temperature are not suitable. The objective of this study is to estimate an optimal value of the airflow and discharge air temperature, which maintains room thermal comfort. This paper is conducted to determine the optimal room airflow and discharge air temperature and verify the impact of room airflow and discharge air temperature on thermal stratification through CFD (Computational Fluid Dynamics) simulations.

Keywords: Air circulation, Thermal comfort, CFD simulation, VAV, AHU

Introduction

Terminal boxes control space conditions in variable air volume (VAV) air-handling unit (AHU) systems. Terminal boxes may cause occupant discomfort if the airflow and discharge air temperature are not suitable. During the winter period stratification and short-circuiting can be cause of thermal discomfort. The higher the supply air temperature above the space temperature, the lower air circulation.

In ASHRAE standard 62, the room temperature difference should not exceed 15℉(-9.4 ℃) to avoid excessive temperature stratification. The supply air temperature is no higher than about 90℉(32.2℃) which limit assumes 75℉(23.9℃) space temperature. But, some zone may require higher supply air temperature to meet peak heating load requirements with low minimum airflow setpoint. Liwerant (2008) suggested the maximum leaving-air temperature (LAT) is 90℉(32.2℃). A LAT above 90℉(32.2℃) will result in stratification and/or short-circuiting. To maintain a LAT of 90℉(32.2℃), minimum (heating) airflow can be adjusted upward. It will have high reheating energy consumption and AHUs will consume more fan power. Therefore, it should be studied to determine an optimal value of the airflow and discharge air temperature, which maintains room thermal comfort and save energy.

The objective of this study is to determine the optimal room airflow and discharge air temperature on thermal stratification. The analytical estimation of optimal value is conducted by numerical simulations and verified by results of temperature and velocity distributions.

Determined minimum value of the inlet air velocity

A balance of a momentum flux of the natural convection and the air flow into the room from the VAV terminal boxes was a basic solution.

Momentum flux of the natural convection

When heat is added to an air and the air density varies with temperature, an airflow can be induced due to the force of gravity acting on the density variations. The importance of buoyance forces in a mixed convection flow can be measured by the ratio of the Graf and Reynolds numbers:

$$Gr = \frac{g \cdot \beta \cdot (T_s - T_\infty) \cdot L^3}{\nu^2} \tag{1}$$

$$\mathrm{Re} = \frac{V \cdot L}{\nu} \tag{2}$$

$$\frac{Gr}{\mathrm{Re}^2} = \frac{g \cdot \beta \cdot (T_s - T_\infty) \cdot L}{V^2} \tag{3}$$

Generally, the combined effects of free and forced convection must be considered when $(Gr/\mathrm{Re}^2) \approx 1$. If the inequality $(Gr/\mathrm{Re}^2) << 1$ is satisfied, free convection effects may be neglected and forced convection is negligible if $(Gr/\mathrm{Re}^2) >> 1$. When this number approaches or exceeds unity, you should expect strong buoyancy contributions to the airflow. Conversely, if it is very small, buoyancy forces may be ignored.

Buoyancy influences within room spaces flow patterns may be significant and it is appropriate to include a gravity term in the vertical component :

$$g \cdot (\rho - \rho_\infty) \tag{4}$$

According to the Boussinesq approximation,

$$\rho = \rho_\infty \cdot (1 - \beta \cdot (T - T_\infty)) \tag{5}$$

The buoyancy term in the momentum equation,

$$\dot{F}_{a,room} = -\rho_{a,box} \cdot \beta \cdot (T_{a,room} - T_{a,box}) \cdot g \tag{6}$$

Momentum flux of the air flow into the room

For the air stream blown into the room, the momentum flux can be calculated by the following equation,

$$\dot{F}_{a,box} = \dot{m}_{a,box} \cdot V_{a,box} = \rho_{a,box} \cdot A_d \cdot V_{a,box}{}^2 \tag{7}$$

Determination of the minimum value of the inlet air velocity

In order to maximize air circulation, the following relation can be found:

$$\dot{F}_{a,box} \geq \dot{F}_{a,room} \tag{8}$$

$$\rho_{a,box} \cdot A_d \cdot V_{a,box}{}^2 \geq -\rho_{a,box} \cdot \beta \cdot \left(T_{a,room} - T_{a,box}\right) \cdot g \tag{9}$$

Consequently, a minimum value of the inlet air velocity can be expressed as

$$V_{a,box} \geq \sqrt{\frac{\beta \cdot \left(T_{a,box} - T_{a,room}\right) \cdot g}{A_d}} \tag{10}$$

Figure 1 shows a relation between a discharge air temperature and room air temperature and resulting minimum inlet air velocity. Figure 2 shows a relation between a discharge air temperature and inlet diameter of terminal box and resulting minimum airflow to prevent a short air circulation.

NUMERICAL SIMULATION

Simulation condition

A simulation model was developed with FDS. The Building and Fire Research Laboratory at NIST has developed a computational fluid dynamics (CFD) fire model using large eddy simulation (LES) techniques. This model, called the NIST Fire Dynamics Simulator (FDS), has been demonstrated to predict the thermal conditions resulting from a compartment fire. A CFD model requires that the room or building of interest be divided into small three-dimensional rectangular control volumes or computational cells. The CFD model computes the density, velocity, temperature, and pressure and species concentration of the gas in each cell. Based on the laws of conservation of mass, momentum, species and energy, the model tracks the generation and movement of fire gases. The basic sets of governing equations are written as follows:

Conservation of Mass :

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \cdot u = 0 \tag{11}$$

Conservation of momentum (Newton's second law) :

$$\frac{\partial}{\partial t}(\rho \cdot u) + \nabla \cdot \rho \cdot u \cdot u + \nabla \cdot p = \rho \cdot f + \nabla \cdot \tau_{ij} \tag{12}$$

Conservation of energy (First law of thermodynamics) :

$$\frac{\partial}{\partial t}(\rho \cdot h) + \nabla \cdot \rho \cdot h \cdot u = \frac{Dp}{Dt} + \dot{q}''' - \nabla \cdot q + \Phi \tag{13}$$

Geometry and a coordinate system of the model are shown in Fig. 3. The model for the simulation is selected as the office as the field experiments to verify simulation results. The one wall of model faces to the south. Boundary conditions that were assumed for numerical simulations are presented in Table 1.

Simulation method

CFD simulations of temperature and velocity distributions are done. Details of 8 simulation items with varying inlet air values and outside air temperature are presented in Table 2. The discharge air temperature was calculated by room load when the room air temperature was maintained at 75°F(23.9°C). Simulation 1~4 was conducted to verify the determined minimum inlet air velocity according to discharge air temperature. Minimum airflow of Simulation 5 is set twice as lower and that of Simulation 6 is set twice as higher as determined minimum value. Calculations were repeated until the room air temperature was maintained its setpoint at height points of 5 ft(1.52 m).

Validation of the models

The FDS originally describes the transport of smoke and hot gases during a fire in an enclosure. Most of the validation work has evaluated the model's ability to predict the transport of heat and exhaust products from a fire through an enclosure. Initial comparisons with experimental data collected in a controlled environmental test chamber [Yuan et al. 1998] have produced encouraging results, while identifying the importance of issues such as representation of boundary conditions at the inlet diffuser and modeling of the heat transfer boundary condition at room walls. Room airflow applications were considered by Emmerich and McGrattan [1997, 1998]. In general, the FDS code is able to produce results that agree reasonably with the experimental data. However, a systematic modeling approach that considers the problem solving technique employed by the code is essential to obtaining the best results.

To verify the simulation results, measurements were performed by vertical difference of the room air temperature. Three data loggers were installed at 6.5 ft(2.0m), 3.6 ft(1.1m) and 0.3 ft(0.09m) above the floor. The height where the room air temperature is controlled is 3.6 ft(1.1m), the height at which room temperature controllers are generally installed. Simulations were conducted by same measurement condition. Two discharge air temperatures 82.5 °F(28.1℃) and 86.0 °F(30.0℃) were applied in the analysis. Figure 4 shows the comparison data between measurement and simulation data at different height. The simulation results can predict the performance within 5% of experimental results. In a simplified simulation, which is modeled by Fire Dynamics Simulator (FDS), the simulated results agree reasonably with the predicted results.

Simulation results and discussion

Figure 5 shows the CFD simulation results of the temperature distribution at XY plan along the y=5 ft(1.52m) in the room for all cases. Figure 6 shows the velocity distribution for case 1, 2, 5 and 6 to see the impact of room airflow and discharge air temperature on thermal stratification. If the minimum airflow is lower than determined value (case 5), it can be poor air circulation in the room and cold air temperature near the occupant can be comfort issue.

The results show that the type of temperature profile ensures a suitable thermal comfort for occupants with minimum airflow of case 1~4 as shown Figure 7. The vertical distribution is kept lower than the value proposed by the study on comfort as the vertical temperature difference is below 5.4 °F(-14.8℃) between the head and ankles (3.6 ft(1.1m) and 0.3 ft(0.09m) above the floor) (ASHRAE 1992; Olesen, B.W. 2002).

CONCLUSION

In this study, optimal minimum airflow for air circulation of single duct VAV Terminal boxes are determined. The analytical estimation of optimal value is conducted by numerical simulations and verified by results of temperature and velocity distributions. The results are as follows:

(1) Determined value of minimum airflow is confirmed by results of CFD simulations.

(2) Determined minimum airflow can prevent occupants from short air circulation.

(3) The vertical distribution is kept lower than the value proposed by the study on comfort.

NOMENCLATURE

Gr – Grashof number

Re – Reynolds number

β - thermal expansion coefficient

g - acceleration due to gravity

$\dot{m}_{a,box}$ - mass air flow rate of terminal box, lbm/hr(g/s)

ρ - the local air density, lbm/ft3(kg/m^3)

ρ_∞ - the constant effective mean air density, lbm/ft3(kg/m^3)

ν - viscosity

V - velocity

$\dot{F}_{a,box}$ - momentum flux of terminal box

$\dot{F}_{a,room}$ - buoyancy term in the momentum equation

$T_{a,room}$ - room air temperature

$T_{a,box}$ - discharge air temperature

f - external force vector (excluding gravity)

h - enthalpy; heat transfer coefficient

p : pressure

$\dot{q}'''$: heat release rate per unit volume

t : time

$u = (u, v, w)$: velocity vector

Φ : dissipation function

ρ : density

τ_{ij} : viscous stress tensor

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References

1. ASHRAE, “Thermal environmental conditions for human occupancy,” *ASHRAE Standard*, 1992, pp.55.

2. ASHRAE, “Ventilation for Acceptable Indoor Air Quality (IAQ),” *ASHRAE standard*, 2004, pp.62.

3. Emmerich S.J., and K.B. McGrattan, “Application of a large eddy simulation model to study room airflow,” 104(1), *ASHRAE Transactions*. 1998, pp.1~9.

4. Int-Hout, D., “Air Distribution For Comfort and IAQ,” *HPAC Engineering*, 1998.

5. Liwerant, E., “VAV-Box Selection, Code Conformance : Demystifying the application of cooling-with-reheat VAV terminals,” *HPAC Engineering*, 2008.

6. McGrattan, K., “Fire Dynamics Simulator (Version 4) User’s Guide,” *NIST Special Publication 1019*, 2008,

7. Olesen, B.W., “Introduction to thermal comfort standards and to the proposed new version of EN ISO 7730,” Vol. 34, *Energy and Buildings*, 2002, pp.537~548.

8. Taylor, S. T., J. Stein, “Sizing VAV Boxes,” Vol. 46, No. 3, *ASHRAE Journal*, 2004, pp.30~35.

9. Webster, T., 2002, “Thermal stratification performance of underfloor air disstibution (UFAD) systems,” *Indoor Air*, 2002.

10. Yuan, X., Q. Chen, L.R. Glicksman, Y. Hu, and X. Yang. 1999. “Measurements and computations of room airflow with displacement ventilation,” Technical Report RP -949, *American Society of Heating, Refrigerating, Air-Conditioning Engineers.*

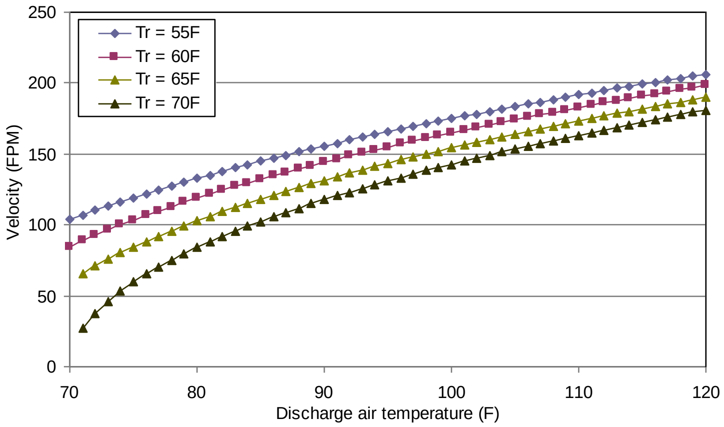


Fig. 1. Minimum inlet air velocity according to discharge air temperature

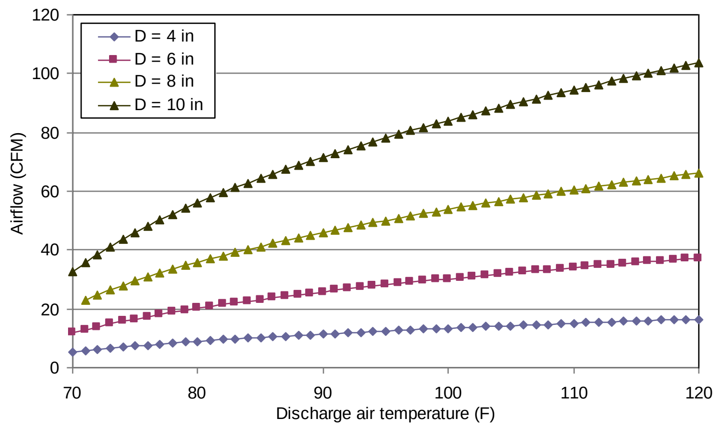


Fig. 2. Minimum airflow according to discharge air temperature

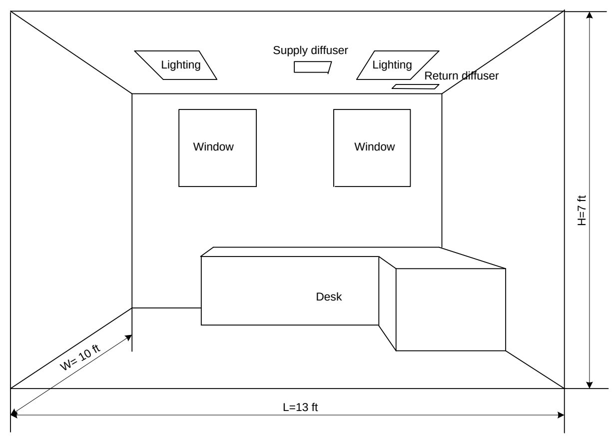


Fig. 3. model for the simulation

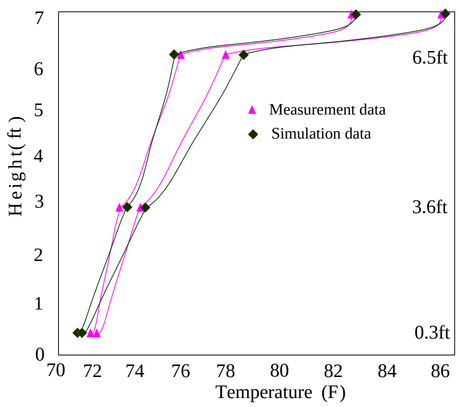
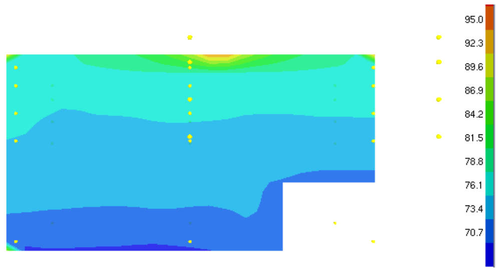
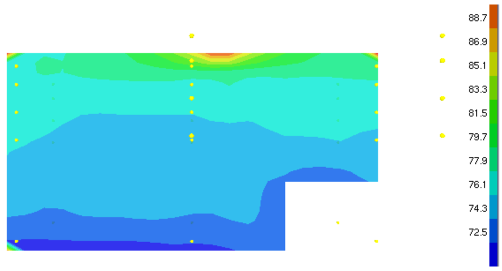


Fig 4. Comparison data for verification of simulation results



Case 1



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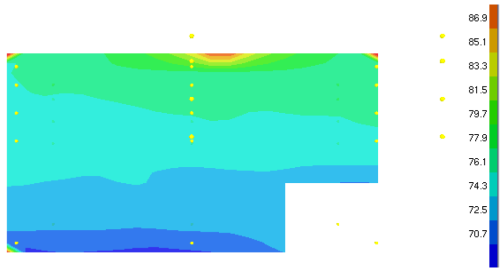

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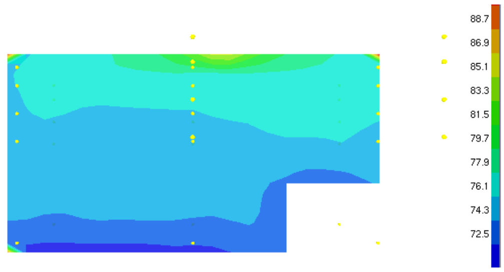
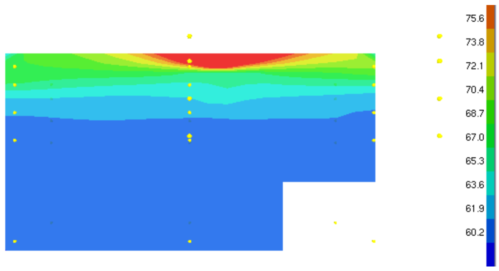
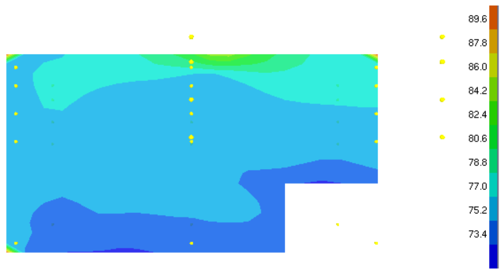


Case 4

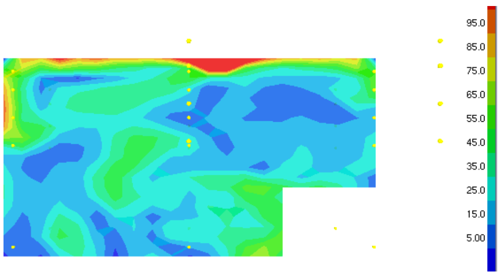


Case 5

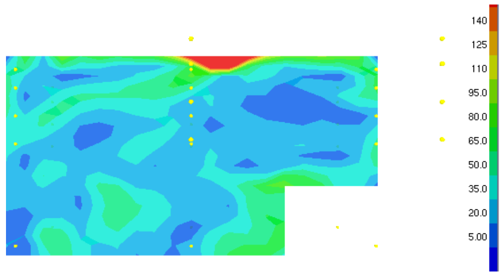


Case 6

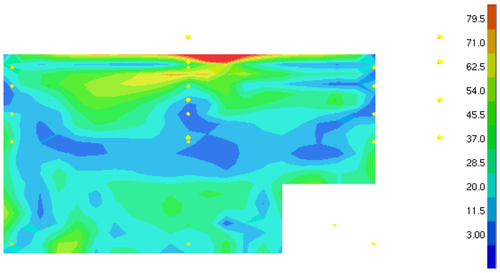
Fig 5. Room air temperature contours (XY plan along the y=5 ft(1.54m), unit : °F)



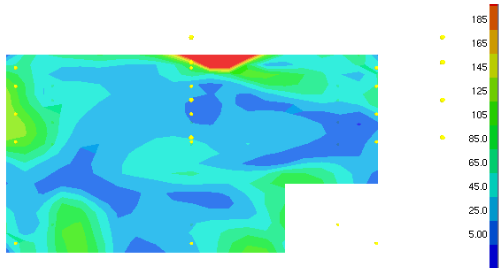
Case 1



Case 2



Case 5



Case 6

Fig 6. Velocity contours (XY plan along the y=5 ft(1.54m), unit : m/s)

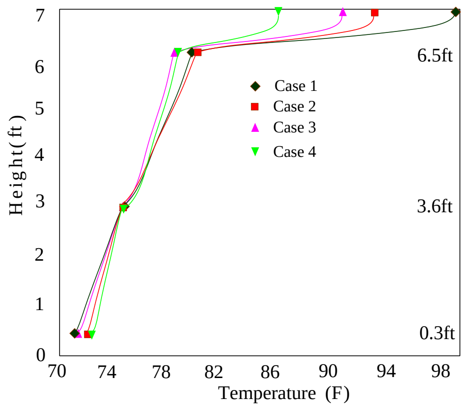


Fig 7. Vertical room air temperature

Table 1. Input data for the simulation

Item	Input data	
Simulation model	Location	Omaha, Nebraska
	Model room	Office rooms
	Floor area	12.08m^2 (3.96*3.04 m)
	Volume	25.77m^3 (3.96*3.04*2.13m)
	Occupancy	1 person
Initial condition	Room air Temperature	12.8℃ / 18.3℃
Boundary condition	Outside air temperature	-12.2℃ / -1.1℃
	Discharge temperature	30.7℃ ~51.1℃
	Supply air velocity	0.38m/s , 0.76m/s , 1.01m/s , 1.52m/s
	Supply diffuser	Louvered face diffuser with 127mm inlet diameter

Table 2. Simulation items.

	Ta, out (℃)	Room load (W)	Va,box (FPM m/s, CFM L/s)	Ta,dis (℃)
Case 1	-12.2℃	147W	0.76m/s, 9.63L/s	37.2℃
Case 2	-12.2℃	147W	1.01m/s, 12.84L/s	34.1℃
Case 3	-1.1℃	101W	0.76m/s, 9.63L/s	33.3℃
Case 4	-1.1℃	101W	1.01m/s, 12.84L/s	30.9℃
Case 5	-12.2℃	147W	0.38m/s, 4.81L/s	51.1℃
Case 6	-12.2℃	147W	1.52m/s, 19.25L/s	30.7℃