Proceedings of The Canadian Society for Mechanical Engineering International Congress 2014
CSME International Congress 2014
June 1-4, 2014, Toronto, Ontario, Canada

HYBRID VENTILATION DESIGN FOR A DINING HALL USING COMPUTATIONAL FLUID DYNAMICS (CFD)

Amir A. Aliabadi, Ehsan Faghani, Hugo A. R. Tjong, and Sheldon I. Green
Department of Mechanical Engineering
University of British Columbia
Vancouver, BC, Canada
aliabadi@aaa-scientists.com

Tariq Amlani and Paul Marmion
Stantec Consulting
Vancouver, BC, Canada

Abstract—Computational Fluid Dynamics (CFD) is used to assist design and investigate performance of a hybrid ventilation scheme (stratified or mixing) for a dining hall. Effects such as occupancy load, cooling and heating modes, ceiling fans operation, and low level bypass exhaust are investigated to predict temperature and odor concentrations within the domain. Under cooling mode, it is recommended to use the displacement ventilation strategy with no ceiling fans operation and no low level bypass exhaust. Under heating mode, it is recommended to use ceiling fans, pushing warm air downwards, and the low level bypass exhaust. The heating mode configuration is based on air mixing, and it results in better thermal comfort and lower odor concentrations at the occupied zone.

Computational Fluid Dynamics (CFD); displacement ventilation; mixing ventilation; temperature stratification; pollutant dispersion; airflow distribution

I. INTRODUCTION

Displacement ventilation (DV) has found a popular place in ventilation design in the recent years. This ventilation strategy relies on the buoyancy force exerted on warm parcels of air in rooms with calm airflow conditions. These warm parcels of air rise (plumes) in cool background air, move pollutants upward, and therefore stratify temperature and pollutant concentration in the room vertically. The pollutants are subsequently removed by a high elevation exhaust, without having a chance to mix with air and contaminate the entire space. If designed and implemented correctly, DV reduces building energy demand and provides better thermal comfort and air quality. The performance of DV is well understood and characterized for small rooms, with simple geometries, and favorable boundary conditions. Complexities arise, however, when one tries to design and implement DV for larger and more complex industrial spaces. An effective DV design for a large and complex premise requires a technical analysis understanding cooling versus heating modes, partial loads, auxiliary heating and cooling devices, solar gains, thermal plumes due to occupants and heat generating appliances, and internal momentum sources such as ceiling fans, and movement of people. Many studies have addressed design challenges in DV pertaining to key effects above and are systematically reviewed here:

A. Heating and Cooling Modes

Lee et al. [1-2] showed that for heating mode, the traditional DV systems with corner diffusers malfunction since the warm jet from the diffuser rises immediately due to buoyancy. As a result, the air velocity in the occupied zone is lower than in cooling mode. The rising warm jet mixes with contaminated air in the higher zone and upon cooling replaces the air in the lower portion of the room. In some instances DV systems are not even recommend for use in heating mode. Similarly, Xu et al. [3] performed a simulation study to show that the traditional DV does not perform well in winter since warm supply air rises immediately upon injection and destroys the vertical stratification (from here on, for both temperature and pollutant concentration). It is recommended to use supplemental heating as opposed to injecting warm air from a diffuser. For cooling mode Lee et al. [2] used simulations to show that supplying air at lower flow rates and lower temperatures results in stronger vertical stratification so that the ventilation effectiveness in the occupied zone of a room is improved. There is, however, a limit to how much air temperature can be reduced since undesirable draft and inadequate thermal comfort may result. Rooms with excessive cooling loads require supplementary cooling systems such as radiant slabs.

B. Auxiliary Heating and Cooling

Auxiliary heating is used with DV to prevent the immediate rise of air supply due to excessive buoyancy. Lee et al. [2] found that baseboard heating for a low-throw underfloor air distribution system did not improve ventilation effectiveness appreciably. Also secondary heating did not help improve the ventilation effectiveness of a traditional DV system in heating mode. Xu et al. [3] suggested that the use of radiant panels are preferred over baseboard heaters since they provide thermal comfort by skin radiation as opposed to changing airflow patterns undesirably. As a result, the ventilation effectiveness does not seem to be altered significantly using radiant panels.

C. Heat Gains and Losses

Lee et al. [1] observed that high heat sources in a room can help vertical stratification in DV but can cause large temperature gradients that result in thermal discomfort. For example, under cooling mode in DV, a temperature difference as high as 3^0C was observed over the height of a person. Lee et al. [2] also showed that increasing the cooling load of a room with DV improves the ventilation effectiveness slightly. This is expected since stronger thermal plumes and cooler air supply improve the vertical stratification.

D. Occupancy

Occupation density is an important factor affecting airflow distribution in a space [4]. Occupants generate thermal plumes that interact with ventilation airflow and other plumes in ventilated space [5-6]. In some situations these plumes can assist contaminant removal, but in other circumstances they can prevent contaminant removal from ventilated spaces (e.g. creating recirculation zones that keep pollutants in the domain for prolonged periods).

E. Space Geometry

The location, size, and volumetric airflow of supply and extraction openings affect flow patterns and pollutant dispersion within a space. The arrangement of inlet and outlet openings can cause different flow recirculation regimes which may change the mean age of indoor contaminants. Chung and Hsu [7] showed that diffuser and exhaust locations greatly affect the removal of contaminants in DV. Interestingly, although the total air exchange rate is insensitive to the location of the diffuser and exhaust, the local ventilation effectiveness varies greatly from one case to another. Yin et al. [8] observed that the performance of the DV system for small rooms is very sensitive to the location of all exhausts. If any exhaust is located at low level, the pollutant concentration at breathing zone will be worse than mixing type ventilation. For small rooms, all exhausts must be located at high elevations, directly on top of the pollutant source, but for large spaces the sensitivity of pollutant concentration to the location of exhausts should be investigated on a case by case basis. A simulation study by Lee et al. [2] showed that using a higher number of diffusers (with lower airflow rate per each diffuser) reduces mixing and results in a higher ventilation effectiveness, particularly in lower elevations of a space. A number of studies [2-3,6] have shown that high ceilings help with vertical stratification in the room. In addition, a higher location for the exhaust results in more effective removal of pollutants.

II. Research Objectives

We intend to analyze performance of a hybrid ventilation system in the ventilation design process of the Crofton House school dining hall located at 3200 West 41st avenue, Vancouver, British Columbia, Canada. The ventilation performance is studied under various scenarios such as heating versus cooling modes and full versus partial ventilation loads. Two additional ventilation strategies are investigated to alter air distribution within the space. These are operating ceiling fans (reversible) and a low elevation bypass exhaust. The bypass exhaust refers to a kitchen door that allows air exchange in the hall in addition to the main exhaust that is provisioned near the ceiling. The performance indices are temperature and tracer gas concentration profiles within the breathing zones.

III. Methodology

A. Geometry, Main Features, and Ventilation Strategy

Figure 1 shows the computational domain for the dining hall developed by Design Modeler in ANSYS 14.0. The occupants sit at the lower floor and around the mezzanine. The total volume of the air in the hall is 5378.49m^3. The heat sources for the domain are as follows: occupants (maximum 1160 people), lighting (8 hung on top of mezzanine level), uninsulated walls and windows under cooling mode, and radiant panels under heating mode (2 at lower level ceiling and 1 at mezzanine level ceiling). The heat sinks for the domain are as follows: uninsulated walls and windows under heating mode, and radiant panels under cooling mode. The hall roof and lower floor are considered thermally insulated. Ten tables at lower floor and a kitchen door are specified as sources for odor from food. Ventilation effectiveness for occupant-generated CO_2 was not studied in this project. However, provided that the odor and occupant-generated CO_2 sources are colocated, their ventilation effectiveness will be similar. Six large square diffusers and a long linear diffuser supply air at the lower floor, while 21 smaller square diffusers supply air at the mezzanine level. The diffusers are distributed, as desired for the ideal DV, around the mezzanine, while due to architectural limitations, only few, but large, panel diffusers are provisioned for the lower floor. The total face area for the diffusers is 50.35m^2. A high level fan-assisted exhaust with an area of 4.55m^2 is implemented as standard outlet. Under heating mode, a kitchen door with an area of 26.14m^2 is provisioned as bypass exhaust as an alternative strategy against using the fan-assisted exhaust only. Driven by building code, the thermal load and ventilation calculations result in an approximate air supply of 13.29m^3/s to the hall under full occupancy. This is equivalent to an air change rate of 8.9 per hour. The air change rate is reduced to 10% with 10% occupancy. Four ceiling fans (reversible) are provisioned to assist movement of air, particularly under heating mode condition.

B. Mesh

The Cut-Cell structured hexahedral element mesh scheme is used. Mesh refinement at boundaries were performed based on expected velocities. The required mesh length scale close to the walls was determined by the 'law of the wall' chart. The standard wall functions in k-ε turbulence models require a length scale for the boundary control volumes in the range 50-500 for y^+. Using y^+=250, the required length scale at diffusers, exhaust, and ceiling fan boundaries are 0.28, 0.03, and 0.02m respectively. The entire volume is filled using proximity and curvature methods. The interior elements are limited in the size

Financial support by Natural Sciences and Engineering Research Council of Canada (NSERC) and Stantec Consulting.

range 0.02-0.81m. The mesh contains 874,026 elements and most parametric simulations are run on this mesh.

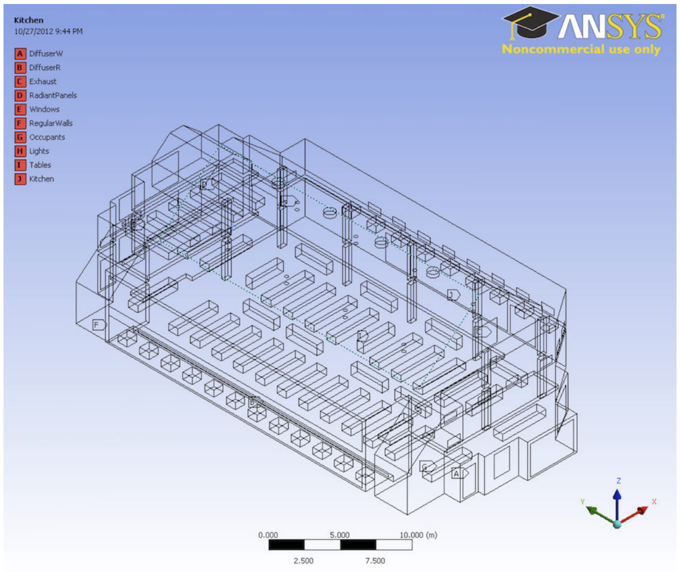

Figure 1. Dining hall geometry – produced by ANSYS 14.0 Design Modeler

A grid independence convergence study has also been performed on the Cut-Cell mesh with two additional sizes of 621,027 and 442,064 elements. The Grid Convergence Index (GCI) was used for this purpose. GCI is a measure of the percentage the computed solution is away from the asymptotic numerical, but not necessarily the true physical, solution [9].

C. *Transport Phenomena and Solution Methodology*

ANSYS FLUENT 14.0 solves conservation equations for mass, momentum, energy, and species. The mixture fluid consists of carbon dioxide (tracer for odor) and air (background gas). Density of the mixture is calculated by the ideal gas law for incompressible flow. Specific heat capacity is determined by the mixing law. Reynolds Averaging Navier Stokes (RANS) turbulence modeling, based on eddy viscosity concept, is used and turbulent diffusion is determined by turbulent viscosity. For brevity of this article, details of modeling are not reproduced but can be found in more detail in [10]. The SIMPLE scheme is used for pressure-velocity coupling, and the PRESTO scheme is used for pressure discretization, which is suitable for buoyant flows [11]. Second order upwind schemes are used for momentum, turbulent kinetic energy, turbulent dissipation rate, species, and energy. To improve convergence, under-relaxation was used for most solution variables.

D. *Performance Indices*

The most natural indices to use for ventilation performance in our study are temperature and odor concentration. Non-dimensional temperature is given by,

$$\Theta=(T-T_s)/(T_e-T_s) \tag{1}$$

where T, T_s, and T_e are local, supply, and exhaust temperatures, respectively. Ventilation effectiveness follows the same concept as air change per hour, but it is used to reveal air refresh rate for a specific point in the ventilation domain [12]. The ventilation effectiveness can be defined as the ratio of two contaminant concentration differentials,

$$E=(C_x-C_s)/(C_m-C_s) \tag{2}$$

where C_x, C_s, and C_m are contaminant concentrations under complete mixing, at supply, and for a subspace of interest, respectively. C_m can be averaged over a subspace zone, in this case horizontal planes, covering the entire floor plan at different elevations. A higher E indicates a more effective mechanism for the removal of contaminants in a given location. The hypothetical completely mixed ventilation systems has the same ventilation efficiency everywhere. If E is greater than 1 for a given location, then it surpasses the performance of a highly mixed system [13].

E. *Case Studies*

Table 1 shows a summary of boundary conditions for simulation cases. For cases with maximum occupancy, diffusers supply flow at 0.25 and 0.26m/s for linear and square diffusers, respectively. These flows are reduced to 10% when occupancy is 10%. Supply temperatures are set to 17.2^{0}C under cooling and 18.3^{0}C under heating modes. The exhaust stack gage pressure is set to -2.394Pa under both cooling and heating modes. In real building controls, this pressure could change with different outside temperatures and wind effects, but we do not consider such variations for simplicity of analysis.

TABLE I. CASE STUDIES FOR HYBRID VENTILATION STRATEGY

Case	*Mode*	*Inlet Flow [%]*	*Outlet*	*Ceiling Fans*	*Occup. [%]*
1	Cooling	100	Ex	Off	100
2	Cooling	100	Ex	Up	100
3	Cooling	10	Ex	Off	10
4	Cooling	10	Ex	Up	10
5	Heating	100	Ex	Off	100
6	Heating	100	Ex	Down	100
7	Heating	100	Ex/BP	Off	100
8	Heating	100	Ex/BP	Down	100
9	Heating	10	Ex	Off	10
10	Heating	10	Ex	Down	10
11	Heating	10	Ex/BP	Off	10
12	Heating	10	Ex/BP	Down	10

EX: Fan-assisted Exhaust, BP: Bypass Exhaust (Kitchen Door)

The space occupied by ceiling fans are treated with cell zone conditions. With ceiling fans operation strategy, a momentum source equal to 78.55N/m^3 is imposed. This quantity is calculated given commercial fan specifications. Under cooling mode this momentum source is imposed upward (+z), while under heating mode this source is imposed downward (-z). The rationale for this is to assist buoyancy (pushing warm air up near the ceiling) under cooling mode and negate it (pushing warm air down near the ceiling) under heating mode. Species (CO_2) mass fraction at the kitchen and tables' boundaries are set to 0.001. The results of the analysis is likely to be insensitive of this mass fraction, given that it is small relative to the background air. Under normal operation the kitchen boundary is treated as an insulated wall, but under half of heating mode scenarios (bypass exhaust) it is defined as a pressure outlet so some air (about 20%) could exit at the bottom floor. The lights dissipate 1561.2W/m^2 of thermal energy. With 100% occupancy, 1170.0W/m^2 of thermal energy is dissipated, while under 10% occupancy this reduces to 10%. Under cooling mode the radiant panels' temperature is set to 15.6^{0}C, but under heating mode it is set to 37.8^{0}C. Under cooling mode regular walls and windows gain 1.6 and 15.8W/m^2 of heat respectively. The heat losses for regular walls and windows under heating mode are 9.0 and 90.0W/m^2 respectively.

IV. Results and Discussion

A. Convergence and Mesh Size

The computed GCI for velocity, temperature, and species at exhaust are 11e-2, 1.2e-5, and 7.1e-3 respectively. The best grid independence convergence is achieved for temperature. We use the finest mesh for the parametric study simulations.

B. Convergence and Turbulence Models

Figures 2 and 3 show the convergence history for case 1 temperature and species, respectively, when different turbulence models are used. Standard k-ε model and Reynolds Stress Model (RSM) fail to converge for temperature on exhaust, given the computational budget (1000 iterations). Standard k-ε, RSM, and k-ω fail to converge for species on exhaust with the same computational budget. Only Shear Stress Transport (SST) and Renormalization Group (RNG) k-ε models succeed to converge within the available computational budget. Since RNG k-ε provides converged results and is less computationally intensive than SST, we use it for parametric study simulations. It is worthwhile to keep in mind that SST is only superior to RNG k-ε when flow exhibits high Rayleigh number buoyancy-driven conditions (natural convection) as opposed to conditions involving both natural and forced convection mechanisms [14], which is the case here. Concerning turbulence models and variations in boundary conditions, we observe that most models have difficulty converging with lower pressure, lower diffuser speed, no ceiling fans operation, and open kitchen door (bypass exhaust). In such situations, the flow Reynolds and Grashof numbers reduce. These flow conditions violate the assumptions in eddy viscosity turbulence modeling, which require that the time scale for turbulent fluctuations be short compared to those of the mean flow and that the flow be dominated by simple shear mechanisms, and freely decaying and isotropic turbulence. As a result, the solution takes longer to converge under aforementioned boundary conditions [12].

C. Temperature

Figures 4 and 5 show vertical profiles of non-dimensional temperature for cooling and heating scenarios, respectively. Each data point is obtained by computing the area-weighted average of temperature across the entire domain at multiple heights. Under cooling mode and full occupancy, the operation of the ceiling fans does not significantly alter the temperature profiles. However, with 10% occupancy, these fans induce significant mixing such that the temperatures increase, shifting profiles undesirably to the right. Under 10% occupancy the mixing is severe since air supply is 10% of the maximum flow rate, and buoyancy is weakened, resulting in an easy disruption of thermal stratification.

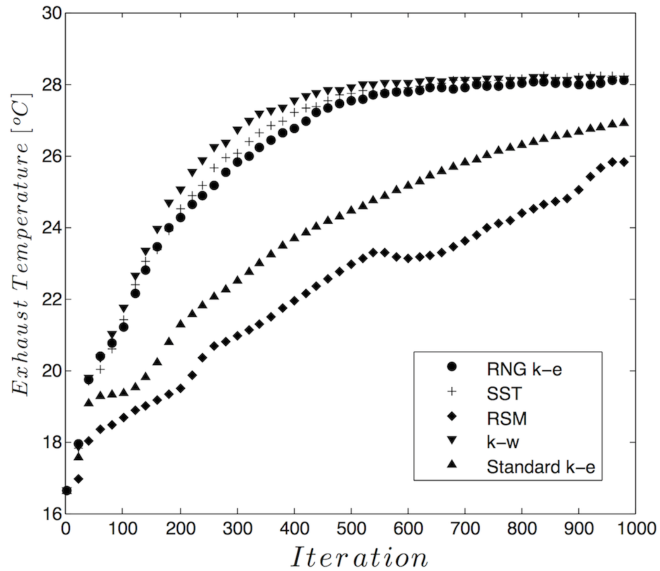

Figure 2. Exhaust temperature convergence with various turbulence models

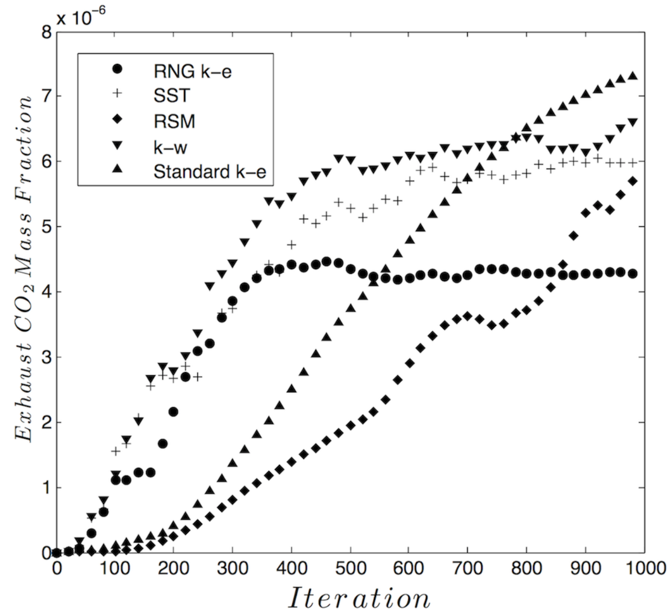

Figure 3. Exhaust species convergence with various turbulence models

Under heating mode and full occupancy, the operation of the ceiling fans alters the temperature profiles appreciably, this time for the desired effect of *pushing* warm air down. This mechanism is useful for either situations when the kitchen door is closed or open (bypass exhaust).

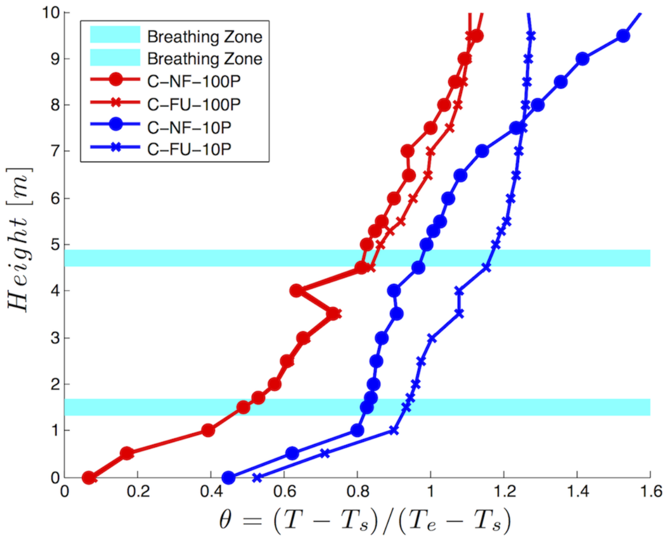

Figure 4. Profiles of non-dimensional temperature for cooling mode (C: Cooling, NF: No Fan, FU: Fan flow Upward, 100P: 100% occupancy, 10P: 10% occupancy)

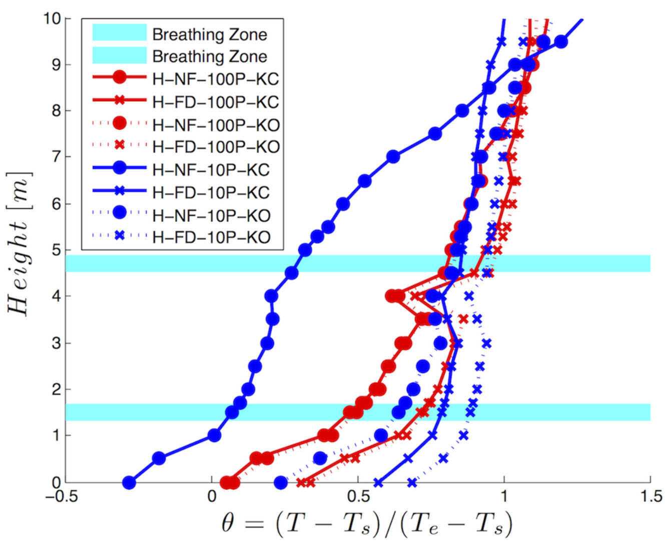

Figure 5. Profiles non-dimensional temperature for heating modes (H: Heating, NF: No Fan, FD: Fan flow Downward, 100P: 100% occupancy, 10P: 10% occupancy, KC: Kitchen Closed, KO: Kitchen Open (bypass exhaust))

An interesting observation is the significant change of temperature profiles, due to kitchen door being closed or open, for heating mode under 10% occupancy. Even with no ceiling fans operation, the profiles shifts to the right, desirably, indicating that some heating source contributes to warming the space uniformly. A detailed look at volume flow rates at inlets and outlets reveals this mechanism. When the kitchen door is closed the exhaust flow is calculated as -1.506m^3/s. When the kitchen door is open (bypass exhaust) the fan-assisted exhaust and kitchen door flows are calculated as -9.603m^3/s and +8.250m^3/s. This hints that actually warm air (23.88^{0}C) from the kitchen enters the domain, as pulled by the fan-assisted exhaust. This is no surprise since the fan-assisted exhaust stack pressure was not adjusted to the low inlet flow rate. If it is not desired to have kitchen air flow into the domain, it is necessary to adjust the fan-assisted exhaust stack pressure. Under heating mode with full occupancy and no ceiling fans operation, the volume flow rates at inlets and outlets were checked for kitchen door being closed or open. When the kitchen door is closed, the exhaust flow rate is calculated as -13.685m^3/s. When the kitchen door is open (bypass exhaust), the fan-assisted exhaust and kitchen flow rates are calculated as -10.952m^3/s and -2.776m^3/s. This indicates that, roughly, 20% of air is exhausted through the kitchen door.

D. *Species*

Figures 6 and 7 show vertical profiles of ventilation effectiveness for cooling and heating scenarios respectively. Each data point is obtained by computing the area-weighted average of species (CO_2) across the entire domain at a given height. To calculate ventilation effectiveness, the fully mixed concentration was approximated by the volume average of species in the entire domain for each case. Overall, the ventilation effectiveness (E) varies between 0.7 and 1.2 near the breathing zones under both cooling or heating modes, indicating moderate departures from concentrations under fully mixed conditions. Under cooling mode, operation of the ceiling fans does not significantly alter the ventilation effectiveness. However, 100% occupancy results in much less species concentration within domain, as compared to 10% occupancy (not shown here). It is speculated that the stronger thermal plume associated with full occupancy and higher ventilation airflow velocities assist the transport of species towards the high elevation exhaust. Under heating mode, the combination of kitchen door being open (bypass exhaust) and 10% occupancy causes a surge in species concentration in the entire domain (not shown). The reason is that +8.250m^3/s of kitchen air is supplied to the domain. Again, the fan-assisted exhaust stack pressure must be lowered, in accordance with inlet flow rate, to prevent kitchen air from entering the domain in large volumes. Under heating mode and no bypass exhaust, the ceiling fans cause significant amount of mixing so that the species are dispersed at higher concentrations throughout the entire domain (i.e. lower ventilation effectiveness). The effect for when the kitchen door is open is different. Operating the ceiling fans actually reduces species concentration and improves ventilation effectiveness. Again assessment of volumetric airflow at inlets and outlets assist understanding this behavior. When the ceiling fans are pushing air downward, the volumetric airflow at kitchen door is changed from -2.776m^3/s to -2.880m^3/s. This means that more air exits through the kitchen door and consequently the species concentration near the floor is reduced.

V. Conclusions

Computational Fluid Dynamics (CFD) is used to assist design and investigate performance of a hybrid ventilation system for a dining hall. While there are uncertainties in the model, the technique enables qualitative comparisons under a variety of operating conditions, suggesting a design approach to reduce ventilation energy consumption, improve thermal comfort, and control odor concentrations in the hall.

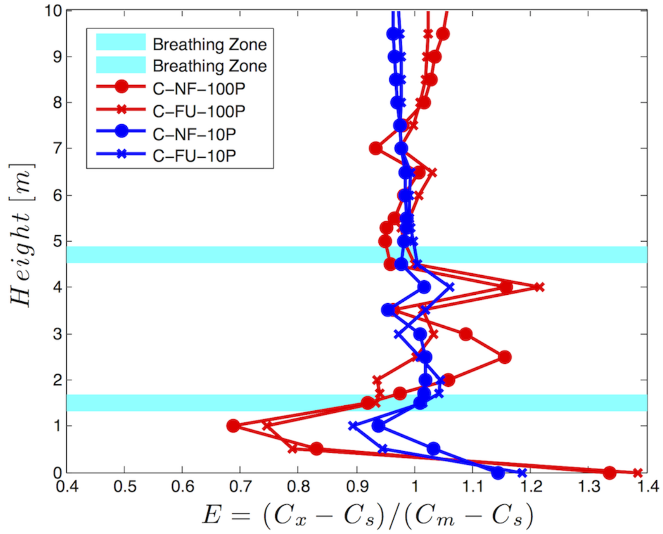

Figure 6. Profiles of ventilation effectiveness for cooling mode (Abbreviations same as in figure 4)

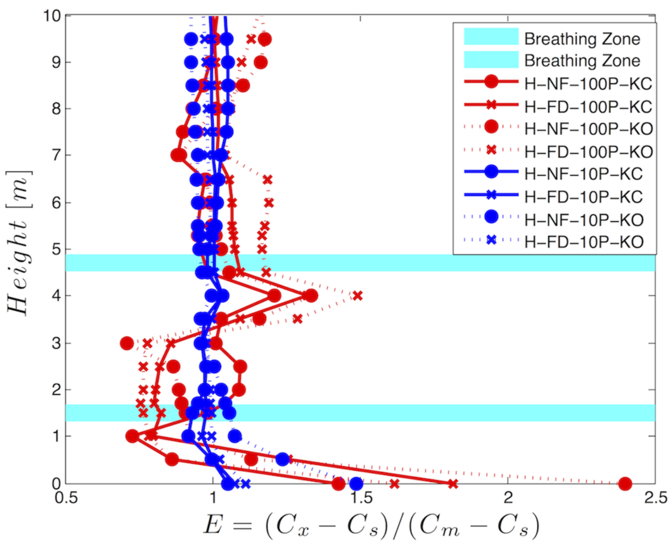

Figure 7. Profiles of ventilation effectiveness for heating modes (Abbreviations same as in figure 5)

The numerical solution is sensitive to the choice of turbulence models selected. RANS models provide converged, but qualitative, solutions, so caution is necessary when using them. RANS simulations are useful and economical in the fast pace consulting world, when it is desired to investigate different classes of design options that have major differences as opposed to fine variations within one class. For this study, it is recommended to use the RNG k-ε model over standard k-ε, k-ω, SST, or RSM models. Under cooling mode, it is recommended to use pure displacement ventilation with high level fan-assisted exhaust. Under heating mode, it is recommended to operate ceiling fans, pushing warm air downward, and to use bypass low level exhaust, in addition to the high level fan-assisted exhaust. For both cases auxiliary radiant heating or cooling is recommended.

Acknowledgments and Disclaimer

The Authors thank Jimmy Ng and Alireza Khaleghi at Stantec Consulting for technical advice. The authors also thank Simon Richards at Cornerstone Architecture, the principal architect of the Crofton House school project, and end client Patricia Dawson, Crofton House school head, for authorizing this publication. This is a scientific study only, and authors assume no responsibility if ideas here are used in any way to potentially cause any damage to any parties.

References

[1] K. Lee, T. Zhang, Z. Jiang, and Q. Chen, "Comparison of airflow and contaminant distributions in rooms with traditional displacement ventilation and under-floor air distribution systems (RP-1373)," ASHRAE Transactions, Vol. 115, No. 2, pp.306-321, 2009.

[2] K. Lee, Z. Jiang, and Q. Chen, "Air distribution effectiveness with stratified air distribution systems," ASHRAE Transactions, Vol. 115, No. 2, pp.322-333, 2009.

[3] W. Xu, A. Manning, A. Guity, B. Gulick, P. Marmion, Y. Yin, and Q. Chen, "Numerical studies of ventilation systems in a patient room," Technical Report, Mentor Graphics Corporation, 2009.

[4] J. W. Tang, Y. Li, I. Eames, P. K. S. Chan, G. L. Ridgway, "Factors involved in the aerosol transmission of infection and control of ventilation in healthcare premises," Journal of Hospital Infection, Vol. 64, pp. 100-114, 2006.

[5] B. A. Craven and G. S. Settles, "A computational and experimental investigation of the human thermal plume," Journal of Fluids Engineering, Vol. 128, pp. 1251-1258, 2006.

[6] H. Qian, Y. Li, P. V. Nielsen, and C. E. Hyldgaard, "Dispersion of exhalation pollutants in a two-bed hospital ward with a downward ventilation system," Building and Environment, Vol. 43, No. 3, pp. 344-354, 2008.

[7] K. C. Chung and S. P. Hsu, "Effect of ventilation pattern on room air and contaminant distribution," Building and Environment, Vol. 36, No. 9, pp. 989-998, 2001.

[8] Y. Yin, W. Xu, J. K. Gupta, A. Guity, P. Marmion, A. Manning, B. Gulick, X. Zhang, and Q. Chen, "Experimental study on displacement and mixing ventilation systems for a patient ward," HVAC & R Research, Vol. 15, No. 6, pp. 1175-1191, 2009.

[9] P. J. Roache, "Perspective: a method for uniform reporting of grid refinement studies," Journal of Fluids Engineering, Vol. 116, No. 3, pp. 405-413, 1994.

[10] A. A. Aliabadi, "Dispersion of expiratory airborne droplets in a model single patient hospital recovery room with stratified ventilation," Thesis, URL https://circle.ubc.ca/handle/2429/43801, 2013.

[11] R. Peyret, Handbook of computational fluid mechanics, USA: Academic Press Limited, 1996.

[12] A. A. Aliabadi, S. N. Rogak, K. H. Bartlett, and S. I. Green, "Preventing airborne disease transmission: Review of methods for ventilation design in health care facilities," Advances in Preventive Medicine, Vol. 2011, Article ID: 124064, (doi:10.4061/2011/124064), 2011.

[13] J. Atkinson, Y. Chartier, C. L. Pessoa-Silva, P. Jensen, Y. Li, and W. H. Seto, "Natural ventilation for infection control in healthcare settings," Technical Report, World Health Organization (WHO) Publications/Guidelines, 2009.

[14] Z. Zhang, W. Zhang, Z. Zhai, and Q. Chen, "Evaluation of various turbulence models in predicting airflow and turbulence in enclosed environments by CFD: Part 2- Comparison with experimental data from literature," HVAC & R Research, Vol. 13, No. 6, pp. 871-886, 2007.