INTRODUCTION: TASKS OF VENTILATION IN OCCUPIED SPACES

Two major tasks of ventilation are to enhance thermal comfort and Indoor Air Quality (IAQ). At the same time, energy consumption for operation must be minimized or optimized. Thermal comfort relies on many factors including air temperature, Relative Humidity (RH), Mean Radiant Temperature (MRT), wind velocity, clothing, and activities. In order to quantify thermal comfort, many indicators have been developed. Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfaction (PPD) are the indexes that are widely accepted (ASHRAE 2005a; Awbi 2003; Fanger 1972). However, the calculation process of these indicators is quite complicated; thus, common indicators such as air temperature and Relative Humidity (RH) are commonly used. Besides Thermal Comfort, IAQ is one of the important parameters for indicating human well-being. Many indoor air pollutants comprising organic and inorganic gases, infectious microorganisms, biological agents, and nonbiological particles and fibers are responsible for IAQ degradation (ASHRAE 2005b). In the worst case scenarios, inappropriate IAQ can lead to Sick Volume 2, Number 2 131

HVAC VENTILATION STRATEGIES: THE CONTRIBUTION FOR THERMAL COMFORT, ENERGY EFFICIENCY, AND INDOOR AIR QUALITY

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ABSTRACT
In Heating Ventilating and Air Conditioning (HVAC) systems, ventilation strategies impact building energy consumption, occupants’ thermal comfort and Indoor Air Quality (IAQ). Ventilation strategies such as Mixing Jet Ventilation (MJV), Displacement Ventilation (DV), and Impinging Jet Ventilation (IJV) are operated on the different principals. MJV relies on dilution, while DV and IJV rely on both dilution and stratification. Due to climatic variation, ventilation strategies must be operated under different cooling and heating load scenarios. Typically, each ventilation strategy controls the indoor environment through a single adequate flow rate with suitable supply parameters such as temperature, pollutant concentration, vapor, velocity, etc. Hence, the indoor thermal and IAQ condition are independently impacted. A room with excellent thermal condition is possible to have poor IAQ. Given this limitation, vast air flow variables, and occupants’ activities, the performances evaluation of these strategies are complicated. In this study, three ventilation strategies, MJV, DV, and IJV are thoroughly investigated. The Computational Fluid Dynamics (CFD) simulation was mainly utilized to handle the complexity of this study. The parametric studies of 48 CFD simulations are presented. Referring to ASHRAE RP-1133, the experimental data from a specially built HVAC-IEQ laboratory was used to validate the CFD data. The research results indicate both advantages and disadvantages in all three strategies. In addition, there is no single strategy that can perform excellently in all indexes. Using the well-known index called ventilation effectiveness (VEF), DV performs outstandingly. However, under a newly proposed index called ventilation performances, DV fails because the stratification discomfort exceeds 36% of room area. MJV suffers from low VEF and excessive draft. However, the IAQ of MJV is not as poor as expected. IJV can be an alternative especially for space where sleeping and sitting activities dominate. IJV can conserve HVAC energy, while maintaining good IAQ. Compared to DV, although VEF is lower, stratification discomfort is minimized to 24%–12% (depending on supply velocity). Overall, this study demonstrates that ventilation strategies are the key to enhance IAQ. Therefore, the utilization of an appropriate ventilation strategy might increase, Leadership in Energy and Environmental Design (LEED) score, particularly for Indoor Environmental Quality, Innovation and Design Process, and Energy and Atmospheric categories.

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Building Syndrome (SBS) and even Legionnaire’s Disease (Baechler 1991). Many techniques such as source removal, air filtration, air tightness, ventilation air exchange, and ventilation strategies are recommended to improve IAQ (ASHRAE 2004; ASHRAE 2005b; Baechler 1991). Among these techniques, ventilation is one of the most effective techniques that must be appropriately designed in any occupied space.

Ventilation can be induced by natural and mechanical means. Natural ventilation is preferred when the outdoor climate is appropriate. The utilization of natural ventilation becomes an advantage because it is energy-free. Also, natural ventilation is positively perceived in architectural practices. On the negative side, the control of natural airflow is not precise. The room temperature, humidity, and wind velocity fluctuate with outdoor climatic variation. When outdoor climate is poor, the occupants can be impacted by pollutants, odors, particles, and noise. To avoid these consequences, mechanical ventilation might be necessary. Mechanical ventilation can control room temperature and IAQ more precisely. Mechanical ventilation can heat, cool, filter, and distribute the conditioned air throughout the building. Thus, mechanical ventilation is commonly referred to as the Heating, Ventilating, and Air Conditioning (HVAC) system. The filtration system reduces air-borne particles before the air is distributed to the occupied space. When air is distributed through ductwork, the windows can be sealed. Therefore, the room acoustic can be isolated from outdoor noise. In a crowded city, mechanical ventilation is sometimes necessary because it can isolate the conditioned space from a polluted and noisy environment (Stein et al. 1992). The drawback of mechanical ventilation is energy waste. The annual energy review of 2006 reveals that about 55% of total US annual energy consumption is used in industrial and commercial sectors. Out of this 55%, 18% is used in HVAC (DOE 2006). Given the energy situation, environmental problem such as global warming, and the concerns of human well-being, the energy consumed by HVAC systems must be minimized. The energy consumption of HVAC systems can be reduced by both increasing system efficiency and improving architectural design. By reducing fresh air intake, the energy consumed by HVAC systems also drastically drop; however, IAQ degradation is a drawback. Currently, the solutions to improve IAQ without wasting HVAC energy are the topics which many practitioners are investigating.

In HVAC systems, the room’s air distribution relies on supply and return characteristics. The design of the supply and return locations, and supply conditions such as velocity, temperature, humidity, and pollutant concentration are called ventilation strategies. The ventilation strategies strongly affect a room’s thermal comfort and IAQ (ASHRAE 2004). Given the appropriate ventilation rate, many researches suggest that ventilation strategies can improve IAQ without thermal comfort reduction (ASHRAE 2005a; Skistad 1994). *ASHRAE Handbook of Fundamental 2005* has suggested four ventilation strategies including Mixing System, Displacement Ventilation, Unidirectional Airflow Ventilation (ceiling-to-floor and wall-to-wall), Underfloor air distribution and Task/Ambient Conditioning (ASHRAE 2005a). The categorization of ventilation strategies is also suggested by Awbi in the book called *Ventilation of Building* (Awbi 2003). Five ventilation strategies including Mixing Jet Ventilation (MJV), Local Exhaust Ventilation (LEV), Piston Ventilation, Displacement Ventilation (DV), and Impinging Jet Ventilation (IJV) are mentioned (Awbi 2003). Diagrams of these systems are shown in Figure 1.

Four ventilation strategies including MJV, Piston, LEV, and DV are already well-known in ventilation practice. A detailed discussion of each method can be found in Awbi et al (Awbi 2003). Since the IJV was first introduced in the 90s, it is less common to many practitioners. This system is operated by facing the supply terminals toward the floor at appropriate height. The high momentum jet hits the floor, which drastically decreases jet velocity. The velocity near the floor is higher than DV; thus, the momentum is high enough to prevent supplied warm air from floating to the ceiling. Therefore, the heating system can be easily applied. In cooling mode, the stratified layer is preserved because high turbulent flow only occurs near the supply terminal area. Theoretically, the same IAQ as that of DV can be achieved (Karimipanah et al. 2000). Overall, IJV smartly utilizes the advantages of MJV and DV. Recently, an IJV system called Air Queen® has started to be commercialized by Fresh AB Company.
Since ventilation strategies strongly impact thermal comfort and IAQ, ventilation strategies can contribute to LEED scoring criteria in three categories: Indoor Environmental Quality (IEQ), Innovation and Design Process, and Energy and Atmosphere (USGBC 2003). In Indoor Environmental Quality (IEQ), a point from high air change effectiveness/ventilation effectiveness (VEF) can be expected. The ventilation strategies directly affect the air change effectiveness. LEED requires air change effectiveness to be higher than 0.9 at more than 95% of occupancy. This threshold can be met by using the ventilation strategies such as DV and IJV (Karimipanah and Awbi 2002; Skistad 1994). Further discussion can be found in “the ventilation effectiveness and performances” section. In Innovation and Design Process, new ventilation strategies, such as DV can be considered as an innovative item. The duct space underneath the floor must be specially designed if underfloor displacement ventilation is used. These items must be well-integrated with the architectural elements. Thus, the design and integration of ventilation strategies might be eligible to obtain a point from this category. In Energy and Atmosphere, energy reduction can increase the score when compared to a base case that complies with ASHRAE/IESNA 90.1 1999–2004 (as a prerequisite), every 10% of energy reduction gains two points. The energy conservation ranges from 20–60% in a new building while an existing building is 10–50%. When appropriate ventilation strategies are applied, energy conservation can be expected. To reach the LEED scoring range, ventilation strategies alone might not be adequate. Incorporating ventilation strategies with other energy efficient designs such as a proper fenestration system, ventilation strategies should increase the energy conservation margin and help in obtaining scores from Energy and Atmosphere category.

Given the importance of IAQ, three ventilation strategies MJV, DV, and IJV are presented through case studies. Once these systems’ performances are compared, the advantages and disadvantages of each system can be identified. The results from this study could be utilized by HVAC practitioners to design the conditioned spaces where thermal comfort and IAQ are most critical. Before studying the impact of each ventilation strategy in a case study, three related areas including design and or research tools, for ven-
tilation control, and space indexes for ventilation strategies should be described.

- **Design and or research tools for ventilation study** include both Computational Fluid Dynamics (CFD) and full scale experiment. This section discusses the strength and weaknesses of each tool and how it was used in this study.
- **Ventilation control** describes the key parameters, such as ventilation requirements, that are crucial for building operations, and thus for computer simulation.
- **Indexes for ventilation strategies** of both existing and new are compared. New indexes that can effectively quantify the energy, thermal comfort, and IAQ of different ventilation strategies were developed.

Using these three topics, the performances of different ventilation strategies are presented. The results are presented in “The Case Study: Ventilation Strategies Comparison.”

**VENTILATION STUDY TOOLS**

Computational Fluid Dynamics (CFD) simulation and full scale experiment are the main research tools in this study. CFD is very useful for ventilation practitioners. For example, CFD allows users to easily visualize flow parameters such as temperature, velocity, pollutant and vapor concentration distribution. Also, since CFD is a computer simulation technique, the time and research budget can be minimized (as compared to full scale experiment). However, the most questionable issue of CFD is its accuracy. ANNEX20 concludes that the CFD accuracy depends on many factors (IEA 1993; Luo et al. 2004). These factors includes user understanding of CFD codes, grid density, turbulent models, etc (Chen and Srebric 2001). All in all, the best way to improve CFD accuracy is by correlating measured data with a full scale experiment.

The HVAC-IEQ laboratory, at TCAUP College of Architecture, The University of Michigan, was specially built for ventilation research (Figure 2). The lab space is 12mx12m with 5m height. A single Air Handling Unit (AHU) with Variable Fan Speed drive (VFS) supplies a maximum flow rate of 9,600 cubic feet per minute (cfm). The supply temperature can be controlled by chilled/hot water coil and fresh air damper. Two supply terminals equipped with Variable Air Volume (VAV) boxes allow users to adjust the flow rate manually or automatically (through thermostat). Two IJV tubes are attached to both VAV boxes. Each tube is moveable and height adjustable. Key ventilation control parameters including flow rate, supply temperature, fresh air intake rate, etc., can be controlled through the Building Automation System (BAS) software called INSIGHT by Siemens (SIEMENS 2004).

In order to quantify flow parameters such as air temperature and velocity in the HVAC laboratory, the Indoor Flow System (IFS-200) by Dantecdynamics is used (Dantecdynamics 2002). The system consists of four omni-transducers. These transducers can measure the air velocity and temperature of unidirectional flow. The system can measure air velocity ranging from 0–6 m/s and temperature ranging from 0–50°C. Based on manufacturer’s specifications, maximum velocity error is 0.02 m/s and maximum temperature error is 0.2°C.

**VENTILATION CONTROL**

The technique called ventilation requirement is used in both computer simulation and actual building control. The major task of any ventilation strategy is to provide at least the minimum ventilation requirement for each criterion. For the IAQ criterion, a maximum CO2 concentration of 1000 ppmv (or no greater than 700 ppmv of outdoor CO2 level) is commonly used as a threshold (see more details in ASHRAE 62-2004 (ASHRAE 2004)). There is some evidence that shows signs of headache when CO2 ex-
ceeds 1000 ppmv (Awbi 2003). LEED recommends that the relative CO₂ level should not exceed 530 ppmv compared to the outdoor condition at any time (USGBC 2003). For the thermal comfort criterion, the preferred air temperature should range from 20.5–23.3°C (69–74°F) for winter and 23–28.8°C (73.5–84°F) for summer, while the preferred RH in conditioned space should be 50%; however, RH ranging from 20%–75% can also satisfy the thermal comfort criteria. To prevent draft, the mean air velocity should not exceed 0.1–0.18 m/s (0.35–0.6fps), depending on the ambient air temperature (ASHRAE 2005a). The room humidity level can be controlled by the humidity level of the supply air. Since humans can tolerate a wide range of relative humidity, the ventilation control can be more flexible. On the other hand, thermal (temperature) control requires higher precision. In most cases, since the cooling load is frequently higher than that of IAQ and humidity, thermal ventilation requirement usually dominates. Consequences occur when the thermal ventilation requirement is lower than the IAQ. The low ventilation rate might cause the pollutant concentration to exceed the threshold. Equations 1 to 3 represent the air temperature, humidity, and IAQ control of complex mixed air. To control all three parameters, the strongest ventilation is always picked. In Equation 1, both supply temperature and flow rate are the key controls to manage room temperature when the thermal ventilation requirement is dominant. This research focuses on the HVAC control scheme, Variable Air Volume (VAV), which relies on flow rate (Q) variation (see more detail in ASHRAE (ASHRAE 2005a)).

In Table 1, Equations 1 to 3, are also used as input parameters for the CFD simulation set-up. However, the implementation of these equations is not simple. For instance, the most complex mechanism of actual room ventilation control is a thermostat. When a thermostat senses the room temperature, it sends the signal to control either room supply temperature or flow rate. In CFD simulation, only transient state simulation can replicate this dynamic thermostat behavior. However, transient state simulations are time and memory consuming. The alternative is to use steady state simulation. In steady state, the thermostat routine is not easily replicated. The detailed discussion on both states can be found in ANSYS CFX user’s manual (ANSYS 2005a) and typical CFD handbooks. When steady state is used, the flow rate can be specified but not the room temperature. The room temperature and other parameters are determined by a numerical balance embedded in CFD algorithms. To precisely control of the room temperature, at least three steady state cases are needed. The room average temperature can be computed by CFD grids in any given space. An example of grid selection for the case study space is shown in Figure 3. The selected grids placed from 0.1 to 1.6 meters plane represent the room temperature in breathing zone. After selecting CFD grids, Equation 1 is used to define the parameter for CFD#1. Theoretically, if the air temperature is completely mixed, the average temperature of the whole space should be

### TABLE 1. Equations of ventilation requirement.

<table>
<thead>
<tr>
<th>Equation 1</th>
<th>Equation 2</th>
<th>Equation 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ Q = \frac{q}{k(T_a - T_i)} ]</td>
<td>[ Q = \frac{W_p v_s}{(w_a - w_r)} ]</td>
<td>[ Q = \frac{M_p}{(C_a - C_s)} ]</td>
</tr>
<tr>
<td>Q = supplied flow rate (CFM, (m^3/sec))</td>
<td>Q = supplied flow rate (CFH, (m^3/\text{s}))</td>
<td>Q = supplied flow rate (CFH, (m^3/\text{s}))</td>
</tr>
<tr>
<td>(q) = total cooling load (Btu/h, W)</td>
<td>(W_p) = total vapor emission rate ((\text{lbf}/\text{h}, \text{kg}/\text{s}))</td>
<td>(M_p) = total CO₂ emission rate ((\text{lbf}/\text{h}, \text{kg}/\text{s}))</td>
</tr>
<tr>
<td>(T_a) = control room temperature (F, C)</td>
<td>(w_a) = room air humidity level ((\text{kgvapor/kg dry air}))</td>
<td>(C_a) = room air CO₂ level ((\text{lbf}/\text{cu.ft}, \text{kg/m}^3))</td>
</tr>
<tr>
<td>(T_i) = supplied temperature (F, C)</td>
<td>(w_r) = supplied air humidity level ((\text{kgvapor/kg dry air}))</td>
<td>(C_s) = supplied air CO₂ level ((\text{lbf}/\text{cu.ft}, \text{kg/m}^3))</td>
</tr>
<tr>
<td>(k) = 1.08 in IP system, \n1,227 in SI system</td>
<td>(v_s) = supplied air specific volume ((\text{cu.ft}/\text{lbs}, m^3/kg))</td>
<td></td>
</tr>
</tbody>
</table>
equal to the control temperature, 23.5°C ($T_s$). However, due to the stratification, the temperature in the breathing zone is usually less than the control temperature. See CFD#1 in Figure 3. Thus, since the warmer temperature is required, flow rate can be reduced. CFD#2 is another CFD simulation with lower supply flow rate. Usually, the warmer average temperature in the breathing zone can be expected from CFD#2. Using average temperature from CFD#1 and #2, the rate of temperature increase and the flow rate reduction can be plotted. CFD#3 is the simulation in which the supply flow rate balances the room temperature at the control set point. In this study, control temperature of 23.5°C is used. Then, the other environmental parameters such as velocity, CO$_2$, and RH can be obtained from CFD#3.

**INDEXES FOR VENTILATION STRATEGIES COMPARISON**

To compare different ventilation strategies, several indexes are recommended (ASHRAE 2004). The index called ventilation effectiveness (VEF, $\varepsilon$) is commonly used by practitioners. A good ventilated room should have high ventilation effectiveness. The ventilation effectiveness, calculated using the two equations in Table 2, can be considered as two indicators. The first indicator is related to energy consumption. Thermal ventilation effectiveness/thermal VEF ($\varepsilon_t$) can be calculated by using air temperature at the supply ($t_o$), exhaust ($t_i$), and room average ($T$). See Equation 4. When the exhaust temperature is high and the supply temperature and room temperature are similar, the heat is effectively removed. Thus, the thermal VEF increases, and thus, energy consumption decreases. The second indicator relates directly to IAQ. Concentration Ventilation Effectiveness / Concentration VEF ($\varepsilon_c$) can be calculated by measuring pollution concentration at the exhaust ($C_o$), concentration at the supply terminal ($C_i$), and average concentration at a given space ($\bar{C}$). See Equation 5. By reducing $\bar{C}$ and increasing $C_o$, $\varepsilon_c$ can be increased. The measurement procedure of VEF using gas-tracing method can be found in ASHRAE 129 (BSR/ASHRAE 2002).

Depending on terminal configurations, ASHRAE 62-2004 recommends that concentration VEF of MJV in cooling mode ranges 0.6–1. Concentration VEF of DV is 1.2–0.7 in cooling mode and heating mode respectively (ASHRAE 2004). High concentration VEF not only can improve room’s IAQ but also reduce energy consumption of HVAC system. In HVAC system, the fresh air intake is essential for improving IAQ. However, the fresh air must be heated or cooled before supplying through the spaces. If the fresh air intake increases, the extra energy is sometimes needed. The fresh air intake can be minimized by increasing concentration VEF (Fisk 2006).

However, both indicators neglect the interaction of humans and indoor environments. Humans are very sensitive to environmental parameters. The discomfort caused by draft and stratification, which hu-

![Figure 3](http://meridian.allenpress.com/endnotesendnotes/10.3992/jgb.2.2.1.131/fig3.png)

**FIGURE 3.** The temperature control method of steady state simulation.

**TABLE 2.** Equations of ventilation effectiveness.

<table>
<thead>
<tr>
<th>Equation 1</th>
<th>Equation 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\varepsilon_t = \frac{t_o - t_i}{T - t_i}$</td>
<td>$\varepsilon_c = \frac{C_o - C_i}{C - C_i}$</td>
</tr>
</tbody>
</table>
mans are very sensitive to, is not included in the VEF. Also, energy conservation and pollutant removal rate are not fully included. Thus, the index called ventilation performances is introduced here to account for these factors. Based on thermal comfort range recommended by ISO 7730 (ISO 2005), stratification discomfort occurs when the temperature is too stratified. Percentage of Dissatisfied (PD) increases when stratification increases. The temperature difference from the height of 1.1 to 0.1m or from body core to ankle is referenced. If the temperature differs more than 3°C, PD of 10% can be expected (ASHRAE 2005a; Awbi 2003). Draft occurs when the high velocity impacts specific human parts such as the ear and neck. The ambient temperature also affects draft. In cool temperatures, humans can be more sensitive to air movement. At an air temperature of 23.5°C, air velocity of 0.15m/s at occupants’ ear and neck can cause PD of 10% (ASHRAE 2005a; Awbi 2003). To simplify both draft and stratification discomfort, the normalization by room area can be applied. Normalized stratification discomfort can be calculated by the ratio between room area where temperature difference exceeds 3°C (from 1.1m to 0.1m), to the total room area. See Equation 8. Normalized draft area can be calculated by the ratio between room area that velocity exceeds 0.15m/s (at the head height level) to total room area. See Equation 9. Besides these two indicators, the energy conservation and pollutant removal rate should be included. Compared to VEF, the thermal VEF also indicates energy conservation. However, using VEF, the amount of ventilation energy consumed is hard to quantify. If the HVAC system is VAV, another indicator called normalized flow rate should be applied. Normalized flow rate is the ratio between required flow rate to the flow rate of peak cooling scenarios. See Equation 6. The required flow rate is the flow rate that is adequate to maintain the control temperature (in this study, 23.5°C). When the required flow rate is reduced, energy can be conserved. Another indicator that should be included is the quantity of the pollutant concentration at occupant level. Since indoor pollutants vary, the measurement of every pollutant is nearly impossible. CO2 is commonly used for IAQ studies (ASHRAE 2004). In Equation 10, Normalized CO2 is the ratio between the average CO2 at nose level to the standard CO2 at 1000 ppmv (Awbi 2003). One last indicator, Relative Humidity (RH), is especially important under certain circumstances. If the ventilation rate is higher than the requirement for humidity load removal, the appropriate RH is automatically maintained. However, in high humidity load scenarios such as crowded occupied spaces for example in an indoor stadium, RH is another index that should be applied. See Equation 7.

In order to use ventilation performances, the CFD simulation is necessary. CFD provides the dense grid resolution which is required for normalized indicators. The full scale experiment is usually limited by the available sensors. Thus, the indicators such as stratification discomfort and draft are almost impossible to quantify.

**TABLE 3.** Ventilation performances calculation.

<table>
<thead>
<tr>
<th>Equation 6</th>
<th>Equation 7</th>
<th>Equation 8</th>
<th>Equation 9</th>
<th>Equation 10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy conservation</td>
<td>RH</td>
<td>Stratification discomfort</td>
<td>Draft</td>
<td>CO2</td>
</tr>
<tr>
<td>$\frac{Q_{cd}}{Q}$</td>
<td>$\frac{w_a}{w_p}$</td>
<td>$\frac{A_{&gt;3C}}{A}$</td>
<td>$\frac{A_{&gt;0.15m/s}}{A}$</td>
<td>$\frac{C_{cd}}{C_{1000ppmv}}$</td>
</tr>
<tr>
<td>$Q_{cd}$ = CFD flow rate</td>
<td>$w_a$ = humidity ratio</td>
<td>$A_{&gt;3C}$ = area of selected plane where temperature of 1–0.1 m exceeds 3°C</td>
<td>$A_{&gt;0.15m/s}$ = area of selected plane where velocity exceeds 0.15 m/s</td>
<td>$C_{cd}$ = CO2 concentration of selected plane</td>
</tr>
<tr>
<td>$Q$ = peak load flow rate</td>
<td>$w_p$ = saturated humidity ratio at a given air temperature</td>
<td>$A$ = room area</td>
<td>$A$ = room area</td>
<td>$C_{1000ppmv}$ = CO2 at 1000 ppmv</td>
</tr>
</tbody>
</table>
THE CASE STUDY: VENTILATION STRATEGIES COMPARISON

To demonstrate the utilization of the previously discussed research tools, ventilation control method and indexes, a case study of a virtual residential unit was selected. The unit, with a gross area of 55 m² (6.5mx8.5m) and room height of 2.7m, would be located in Las Vegas, Nevada. It consists of a living room, bedroom and bathroom with main glazing oriented south. See Figure 4.

The project aims at obtaining LEED certification, so energy conservation, thermal comfort, and IAQ become important. The ventilation system must be properly selected. MJV, DV and IJV are the candidates. Since the mock-up will be built when the design is finalized, CFD is being used as a primary research tool for this preliminary design process. However, the verified experimental data from HVAC-IEQ laboratory was utilized to assure the CFD accuracy.

Systematic verification and validation was performed. The template of this CFD report is suggested by ARSHRAE RP-1133 (Chen and Srebric 2001). The report consists of two main parts, verification and validation and reporting results.

Verification
Suggested by ASHRAE RP-1133, the verification is reported in five categories. Most of the verification process is a comparison between data from the CFD model and measured data from HVAC laboratory.

1. Basic flow features of this study are the commercial CFD software called ANSYS CFX 10.0. The software is widely used in many applications including HVAC applications. Among available software, this software can handle the complex geometry, and has outstanding visualization features (Awbi 2003). The validation of this software is well-known and can be founded in the company website, www.ansys.com. However, since the IJV system is a new application, it was tested in the HVAC-IEQ laboratory as shown in Figure 2. To verify the CFD code, the CFD model of this laboratory is simplified and tested. See Figure 5.

The model consists of one round IJV tube with a free area of 1 sq.ft. The return is directly located at the bottom of Air Handling Unit (AHU). The heat sources including baseboard heater and fluorescents were modeled for cooling scenarios. The partitions and room furniture were modeled as wall-type boundaries. The velocity and temperature were measured from different nozzle distances (x/d). In the following session, both CFD and experimental measured from HVAC-IEQ laboratory data are compared.

2. Turbulent Models were tested. Some previous literature suggested that the renormalization group k-ε (RNG k-ε) model is suitable for ventilation studies especially for studies of IJV systems (Awbi 2003). However, contrary to these previous studies, the k-Ω model shows the closest agreement
with measured data. In Figure 6, the normalized velocity profile \( \frac{U}{U_o} \) which is the ratio between velocity at any given point \( U \) divided by nozzle velocity \( U_o \) was plotted. At any nozzle distance \( x/d \), the velocity profiles of k-\( \Omega \) model are closer to the measured data than RNG k-\( \varepsilon \) especially in the far field range.

3. **Auxiliary heat transfer and flow model** consists of the adiabatic wall surface temperature and the fixed surface temperature of the heat sources. Using thermal imaginary camera (infrared camera), the surface temperature of the heat sources in the HVAC laboratory were recorded (FLIR 2005). See Figure 7. The fluorescent lights indicate the surface temperature of 60°C, while the baseboard heater shows the surface temperature of 30°C. In CFD simulation, these data was implemented. The other surfaces were set as adiabatic heat transfer. The initial temperature of 25°C was applied. Located 0.75 m above the ground, IJV terminal were modeled identical to the actual setup. The supply velocity at the center of the nozzle is 3.2 m with temperature of 55°F or 13°C. The turbulent intensity is set to 10%.

4. **Numerical Methods** were performed by using three CFD models with 100,000, 250,000, and 500,000 grids (cells). The velocity and temperature profile of 250,000 and 500,000 grids are almost in agreement. Thus, the 250,000 grid resolution is fine enough to represent the actual flow condition (reach grid-independency (Nielsen 2004)). Since the supply velocity dominates the indoor flow, the upwind scheme is adequate to predict the flow condition accurately. In all monitored parameters, the worst final average residual level reached 1E-4 within 40 iterations.

5. **Assessing the CFD prediction** indicates the same agreement between measured data and CFD results. In Figure 8 and Figure 9, the CFD velocity profile of the IJV system is almost identical to the gas-tracing results from studies by Karimipanah and Awbi (Karimipanah and Awbi 2002). In cooling mode testing, the velocity profile of CFD simulation and infrared images taken from HVAC laboratory are roughly similar. See Figure 10 and
Two IJV terminals of low supply velocity (larger supplies), IJV#1, and high supply velocity (smaller supplies), IJV#2, were tested separately. The summary of surface (boundary) area and CFD grids are presented in Table 4.

2. Turbulence model and auxiliary heat transfer and flow models.

In the verification process, since the k-Ω model provides better results, this model was selected. A detailed discussion of this model can be found in ANSYS CFX manual and ventilation handbooks (ANSYS 2005a; Zhang 2005). Typical Radiation Transport Equation (RTE) was applied. The Monte Carlo method was used to simulate the south glass radiation. The histories of 300,000 were used to obtain the finer results (ANSYS 2005a). The various cooling loads were also set up. The vapor and CO2 concentrations were determined by transport equations embedded in CFD code. The kinematic diffusivity of CO2 which is 5.32E-6 m²/s (Zhang 2005) and vapor

Validation and Reporting Results

After verification, the case study shown in Figure 4 was simulated. Since there is no experimental mock-up test of this virtual case study, the CFD parameters in verification process obtained from HVAC-IEQ laboratory are maintained as closely as possible. ASHRAE RP-1133 demands CFD results to be reported systematically.

1. CFD design. In Figure 13, the CFD model was created to replicate the case study as much as possible. Since the room lay-out is yet to be finalized, the furniture details were excluded. Two occupants and seven luminaries were modeled. The directional radiation was set at the south glass. The supply terminals of MJV, DV, and IJV were placed. Two IJV terminals of low supply velocity (larger supplies), IJV#1, and high supply velocity (smaller supplies), IJV#2, were tested separately. The summary of surface (boundary) area and CFD grids are presented in Table 4.
uniform velocity profile is assigned to all supplies. Humans are the main vapor and CO₂ sources from exhalation at a velocity of 0.5 m/s. The ventilation exhausts/returns pressures were assigned to zero.

4. Numerical method. Similar to the verification process, the CFD models of 100,000, 250,000, and 300,000 grids, were used. The CFD model meshed by ANSYS ICEM 10 with a 250,000 grids is shown in Figure 14 (ANSYS 2005b). The unstructured mesh is applied globally. However, since DV and IJV involve flow near the floor surface, the prism layer is extruded from the floor surface. Ten layers of prism mesh with total thickness of 0.45 m increase the meshing resolution for both DV and IJV. The grid size of each surface is shown in Table 4. The temperature, velocity, and CO₂ of 250,000 and 300,000 grids are similar. Hence, the CFD with a 250,000 grids is fine enough for CFD analysis. The final residuals of each variable range from 5E-4 to 1E-6, which are good enough for demonstrating flow patterns (ANSYS 2005a). However, the time step must be reduced to be less than ten seconds.

5. Accessing CFD prediction. Although experimental data is not available in the validation process of this virtual case study, the validation of CFD simulation was well established in the verification process with the HVAC-IEQ laboratory. This CFD simulation setting was utilized for ventilation strategy comparisons to assure the highest ac-

### TABLE 4. The CFD case study surface area and grids.

<table>
<thead>
<tr>
<th>Components</th>
<th>Surface Size (sq.m.)</th>
<th>Grids size (m)</th>
<th>Grids number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ceiling</td>
<td>57.00</td>
<td>0.45</td>
<td>2,872</td>
</tr>
<tr>
<td>All wall</td>
<td>143.74</td>
<td>0.45</td>
<td>8,719</td>
</tr>
<tr>
<td>South glass</td>
<td>22.86</td>
<td>0.45</td>
<td>1,122</td>
</tr>
<tr>
<td>Floor</td>
<td>57.00</td>
<td>0.22</td>
<td>4,706</td>
</tr>
<tr>
<td>Light (x7)</td>
<td>0.19</td>
<td>0.07</td>
<td>288</td>
</tr>
<tr>
<td>Human (x2)</td>
<td>4.19</td>
<td>0.10</td>
<td>3,040</td>
</tr>
<tr>
<td>Nose (x2)</td>
<td>7.26E-4</td>
<td>0.01</td>
<td>25</td>
</tr>
<tr>
<td>Exhausts (x3)</td>
<td>0.42</td>
<td>0.15</td>
<td>143</td>
</tr>
<tr>
<td>MJV (x2)</td>
<td>0.210</td>
<td>0.07</td>
<td>303</td>
</tr>
<tr>
<td>DV (x2)</td>
<td>1.547</td>
<td>0.30</td>
<td>249</td>
</tr>
<tr>
<td>IJV1 (x2)</td>
<td>0.186</td>
<td>0.07</td>
<td>288</td>
</tr>
<tr>
<td>IJV2 (x2)</td>
<td>0.093</td>
<td>0.07</td>
<td>127</td>
</tr>
</tbody>
</table>

which is 2.57E-4 m²/s (Montgomery 1947) were also inputted.

3. Boundary conditions. The surface temperature of most of the walls was set to adiabatic. The details of the cooling, vapor, and CO₂ load components are presented in Table 5. Using Equation 1 and CFD ventilation control techniques, the flow rate for CFD#1 and CFD#2 can be set up. See Table 6. The different supply velocities for each ventilation strategy according to supply size and different cooling load were presented in Table 6. In all cases, the CO₂ and vapor concentrations are constant at 3.4E-4 kg/m³ (350 ppmv) and 0.0101 kg/m³, respectively. The outdoor intake CO₂ level can be found in ASHRAE 62 (ASHRAE 2004).
In the following section, the ventilation effectiveness and performances of four ventilation strategies are presented.

**Ventilation Effectiveness**

From Figure 15 to Figure 18, Ventilation Effectiveness Map (VEM) of MJV, DV, IJV#1, and IJV#2 are plotted. Only the selected scenarios of low (9 W/m²) and high (116 W/m²) cooling loads are shown. Figure 15 and Figure 16 represent thermal VEF using Equation 4, while Figure 17 and Figure 18 represent concentration VEF using Equation 5. In Figure 15 and Figure 16, overall, when cooling increases, thermal VEF is more differentiated, while when cooling load decreases, thermal VEF is less differentiated. In Figure 15 (high cooling load), the breathing zone of DV and IJV#1 have the highest thermal VEF. IJV#2 has a lower thermal VEF, while MJV has the lowest thermal VEF. MJV homogenously mixes the air in the space; thus, the thermal VEF is equally distributed. On the contrary, IJV and DV utilize stratification. Thus, the thermal VEF in the breathing zone is higher. Since the supply velocity of IJV#2 is faster than IJV#1, supply temperature mixing with ambient air is faster. Accordingly, the stratification occurs less. In Figure 16 (low cooling load), only DV performs distinctively, while MJV still performs worst. Both IJV still maintain high thermal VEF near the floor surface.

Calculated by Equation 5, the concentration VEF of four strategies is plotted in Figure 17 and Figure 18 to represent high and low cooling load scenarios respectively. The concentration VEF near the human nose is low (less than 1) because of the high CO₂ level produced by exhaling. MJV indicates the most uniform thermal VEF. Thermal VEF slightly improves by stratification in IJV#2. However, due to lower supply velocity range, IJV#1 performs similarly to DV. When MJV’s supply velocity increases (cooling load increases), MJV eliminates
the stratification. The low VEF from exhaling tilts and mixes with the supply air jet. As a result, the overall VEF is lowest. The concentration VEF slightly improves in IJV#2. The low VEF profile still forms vertically toward the ceiling. DV and IJV#1 indicate the highest concentration VEF especially near floor level. Low VEF forms toward the ceiling. When cooling load reduces, all supply velocity is minimized. DV and IJV#1 still perform excellently, in that the stratification is maintained. The perform-
ance of IJV#2 also improved. Although the low concentration VEF of MJV forms toward the ceiling, the overall concentration VEF is still lowest. All in all, concentration VEF depends on not only the ventilation strategies but also the supply velocity at any given load scenario.

**Ventilation Performances**

The ventilation performances can be calculated by Equation 6 to Equation 10, as described above. The CFD plots of room temperature and normalized CO$_2$ level are shown in Figure 19 and Figure 22 the patterns are the same as the thermal VEF. Tempera-

![FIGURE 17. Concentration ventilation effectiveness at high cooling load (116 W/m$^2$).](image1)

![FIGURE 18. Concentration ventilation effectiveness at high cooling load (9 W/m$^2$)](image2)
ture stratification increases when cooling load increases. In high cooling load, MJV has the most homogeneous air temperature. DV and low velocity IJV#1 have the highest stratified temperature. The high nozzle velocity of IJV#2 increases the mixing process and reduces the temperature stratification.

The additional information that is not available in the thermal VEF is the stratification discomfort. If the temperature difference along the room height is too high, the stratification discomfort increases. See calculation of normalized stratification discomfort in Equation 8. Conversely, if the temperature difference
is not too great, the cool air layer can benefit the activities that utilize the lower part of the room such as sitting and sleeping. Therefore, the strategies that utilize stratification such as DV and IJV are potentially effective. In low cooling load, the stratification reduces in all cases. MJV still has the most homogeneous air temperature, while the others reduce temperature stratification to be almost none. Since the supply vapor concentration of all strategies is similar, the RH of all strategies is similarly acceptable. The RH might become problematic if Constant Air Volume (CAV) is applied. A discussion on this topic can be found in (Varodompun et al. 2006).

In Figure 21 to Figure 22, the patterns of normalized CO₂ level and concentration VEF are similar, as was expected. Equation 10 is used to plot the normalized CO₂ level. In high cooling load (Figure 21), when supply velocity increases, the stratification pattern is disturbed. DV and IJV#1 can preserve the smoke (high CO₂ level) vertically. In IJV#2, the CO₂ smoke still forms vertically, but is slightly tilted. Except the MJV, the rest can maintain the clean air near floor level. Hence, activities such as sleeping might be benefited. Unlike the concentration VEF, normalized CO₂ of high and low cooling load conditions is distinctively different. Due to low supply flow rate, the CO₂ level elevates when cooling load is reduced. However, both DV and IJV can still maintain the cleaner air near floor level, while MJV cannot.

**DISCUSSION: THE COMPARISON OF VENTILATION EFFECTIVENESS AND VENTILATION PERFORMANCES**

In Figure 23, the thermal and concentration VEF are plotted against cooling load variation. Both VEFs are calculated by averaging the grids data at 1.1m height. In this simulation, all strategies have a VEF higher than one which is recommended by LEED for IAQ scoring. Using these indicators and MJV as a base case, DV performs best in almost every scenario. When cooling load exceeds 40 W/m², concentration VEF increases. The thermal VEF reaches 1.3 at cooling load of 120 W/m². IJV#1, low supply velocity, maintains high concentration VEF in all cooling load scenarios. If cooling load is 40–125 W/m², the concentration VEF of IJV#1 exceeds DV; while, thermal VEF of IJV#1 drops if cooling load exceeds 120 W/m². In peak cooling load, thermal VEF of IJV#1 becomes identical to MJV, while concentration VEF is still higher than MJV. With high supply velocity, IJV#2 performs worse than IJV#1 but still better than MJV. However, if cooling load goes above...
125–130 W/m², the VEF of IJV#2 drops below that of MJV. Since VEF of IJV is located between DV and IJV, the approximate concentration VEF of 1.1–1.2 should be added to ASHRAE 62-2004.

Although both VEFs suggest that DV performs excellently, some contradictory outcomes are found when ventilation performances are applied. In Figure 24, four indicators including energy conservation, stratification discomfort, draft, and IAQ are plotted using the calculations of ventilation performances from Equation 6 to Equation 10. Since the space can be utilized by different activities such as standing, sitting, and sleeping, the breathing zone height varies. Thus, the range of each indicator can be obtained. Similar to VEF, ventilation performances vary when cooling scenarios change. In energy conservation, DV performs best in all load scenarios. IJV performs slightly better than MJV. However, in peak cooling

**FIGURE 22.** CO₂ concentration profile at low cooling load (9 W/m²).

**FIGURE 23.** Thermal VEF (left) and concentration VEF (right).
load, the same agreement with VEF analysis is found. The energy performance of IJV drops if velocity exceeds appropriate range. When stratification discomfort is concerned, the advantage of IJV is elevated. IJV can reduce stratification discomfort significantly. With high supply velocity, IJV#2 can control stratification discomfort below 12% in all cases. Since the supply velocity of IJV#1 is lower, the maximum stratification discomfort increase to 24%. DV suffers from the highest stratification discomfort in almost all load scenarios. Stratification discomfort of DV reaches 36% in high cooling load cases. The supply velocity is too low to mix and dilute the cool supply air temperature, while high supply velocity of IJV can. In consideration of IAQ, most strategies perform similarly. However, DV causes a high CO2 level in a peak cooling scenario due to over-energy conservation (too low flow rate). IJV has an almost identical or slightly better CO2 level than MJV, while energy is less conserved. In the draft indicator, MJV performs worst, while DV is almost draft-free. Both IJV perform acceptably. IJV#2 has a worse draft problem but still less than that of MJV.

The selected ventilation performances are plotted against cooling load scenarios in Figure 25 (left side). The supply velocity of the IJV system is marked. The yellow highlighted area indicates the energy conservation zone of both IJV. Within this zone, the flow rate can be reduced, while the control temperature (23.5°C) is achieved. Both IJV similarly and effectively performs if cooling load is lower than 80 W/m² of 5.5 ACH. When cooling load increases, the supply flow rate limit is founded in both IJV. However, IJV#1 with lower supply velocity has higher supply flow rate threshold than IJV#2. The supply velocity thresholds of IJV#1 and IJV#2 are 2.60 m/s (11 ACH) and 3.84 m/s (8 ACH) respectively. In Figure 25 (right side), the normalized stratification

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**FIGURE 24.** Ventilation performances of three ventilation strategies (MJV, DV, IJV).
discomfort of IJV#1 and IJV#2 is compared. In IJV#1, the peak normalized stratification discomfort is 0.24 or 24% of room. IJV#2 significantly reduced stratification discomfort to be lower than 12%. Peak stratification discomfort occurs at 5–5.5 ACH. Though this flow rate range is suitable for energy conservation, the stratification discomfort is a drawback. By increasing the supply velocity, stratification discomfort can be reduced by half. However, if lower stratification discomfort is preferred, the warmer supply velocity is also an alternative. By increasing supply temperature only 1 to 2°C, the stratification discomfort problem is almost remedied (Varodompun et al. 2006).

CONCLUSION
Using experimental data (from HVAC-IEQ laboratory), CFD results were systematically verified, validated, and reported by following ASHRAE RP-1133. The CFD results were used to reveal the performances of different ventilation strategies. Existing and new indicators including ventilation effectiveness and ventilation performances of Mixing Jet Ventilation (MJV), Displacement Ventilation (DV), and Impinging Jet Ventilation (IJV) are compared. Though DV performs excellently if using ventilation effectiveness as an index, DV fails one of the ventilation performances, the normalized stratification discomfort. To solve this problem, IJV is an alternative. The stratification discomfort significantly reduces when an appropriate supply velocity of IJV system is implemented. The ventilation effectiveness of IJV might be lower than DV but still higher than MJV if suitable supply velocity is applied. The approximate concentration VEF of 1.1–1.2 for IJV system (VAV with cooling mode) is founded in this study. As a result, the IJV system might be an alternative to gain LEED scores especially in categories of Indoor Environmental Quality, Innovation and Design Process, and Energy and Atmosphere.

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REFERENCES


