Experimental Evaluation of the Thermal and Ventilation Performance of Stratified Air Distribution Systems Coupled with Passive Beams

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Abstract

This report presents our research on the thermal and ventilation performance of displacement ventilation coupled with passive chilled beams, sponsored by ASHRAE TC 5.3 – Room Air Distribution through research project 1666.

In the research, we first reviewed the state-of-the-art of displacement ventilation systems and passive chilled beam systems. Although displacement ventilation (DV) can provide good indoor air quality, its ability to remove heat is limited. On the other hand, passive chilled beams (PCBs) can have a high heat-removal capability. Through the literature review, we identified the gaps in current studies on displacement ventilation and passive chilled beam systems and concluded that a coupled DV-PCB system had the potential of merging the advantages of both systems, but further investigation was needed to evaluate the coupled system.

To evaluate the performance of a DV-PCB system, this study first conducted experimental measurements in a full-scale environmental chamber with the coupled system to obtain the distributions of air velocity, temperature and contaminant concentration. A computational fluid dynamics (CFD) model was developed to simulate air distribution in an enclosed environment with the DV-PCB system, which was then validated by the measured data. The validated CFD model was employed to analyze thermal comfort and indoor air quality in the enclosed environment with the DV-PCB coupled system using four indices: vertical temperature gradient, draft rate, normalized contaminant concentration and age of air. Furthermore, this research used the CFD model to establish a database of 70 cases that include four typical types of indoor spaces with DV-PCB systems, and developed a mathematical model for predicting the thermal and ventilation performances of DV-PCB based on various design parameters. The four spaces were office, classrooms, workshops, and restaurants. With the model, this investigation proposed a step-by-step procedure for designing a DV-PCB. Moreover, a user-friendly design interface was developed such that engineers can use it to make design decisions in a convenient way.

The results showed that PCBs were quite effective in reducing the temperature gradient created by DV. However, the downward air jet generated by the PCBs could to some extent disrupt the contaminant stratification developed by DV, and the ventilation effectiveness of the coupled system depends on various design parameters. The established design guide provides a procedure for determining the maximum load that could be removed by PCB and designing a DV-PCB system that meets thermal and ventilation requirements without condensation risk. Finally, the coupled system has more significant energy saving potential when it is used in dry and hot climate regions.
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## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACH</td>
<td>Air change per hour</td>
<td>h⁻¹</td>
</tr>
<tr>
<td>A</td>
<td>Area of room</td>
<td>m²</td>
</tr>
<tr>
<td>$A_{PCB}$</td>
<td>Total cross-sectional area of PCB used</td>
<td>m²</td>
</tr>
<tr>
<td>C</td>
<td>Contaminant concentration</td>
<td>-</td>
</tr>
<tr>
<td>$C_e$</td>
<td>Contaminant concentration at exhaust</td>
<td>-</td>
</tr>
<tr>
<td>$C_{res}$</td>
<td>Convection due to respiration</td>
<td>W</td>
</tr>
<tr>
<td>$C_{sk}$</td>
<td>Skin convection</td>
<td>W</td>
</tr>
<tr>
<td>$C_s$</td>
<td>Contaminant concentration at supply</td>
<td>-</td>
</tr>
<tr>
<td>$C^*$</td>
<td>Normalized contaminant concentration</td>
<td>-</td>
</tr>
<tr>
<td>E</td>
<td>Energy consumption from HVAC components</td>
<td>kWh</td>
</tr>
<tr>
<td>$E_{res}$</td>
<td>Evaporation due to respiration</td>
<td>W</td>
</tr>
<tr>
<td>$E_{sk}$</td>
<td>Skin evaporation</td>
<td>W</td>
</tr>
<tr>
<td>$e_i$</td>
<td>Average energy consumption per hour in section $i$</td>
<td>W</td>
</tr>
<tr>
<td>H</td>
<td>Room height</td>
<td>m</td>
</tr>
<tr>
<td>$h^*$</td>
<td>Normalized height</td>
<td>-</td>
</tr>
<tr>
<td>$I_{cl}$</td>
<td>Clothing level</td>
<td>°C m²/W</td>
</tr>
<tr>
<td>L</td>
<td>Length of PCB</td>
<td>m</td>
</tr>
<tr>
<td>M</td>
<td>Metabolism</td>
<td>W</td>
</tr>
<tr>
<td>$m_{air}$</td>
<td>Air mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>$m_{inf}$</td>
<td>Air mass flow rate through infiltration</td>
<td>kg/s</td>
</tr>
<tr>
<td>$P_i$</td>
<td>Probability of the weather data being in section $i$</td>
<td>-</td>
</tr>
<tr>
<td>PD</td>
<td>Percentage dissatisfied people due to draft</td>
<td>-</td>
</tr>
<tr>
<td>PMV</td>
<td>Predicted mean vote</td>
<td>-</td>
</tr>
<tr>
<td>PPD</td>
<td>Percentage predicted dissatisfied people</td>
<td>-</td>
</tr>
<tr>
<td>$Q_{PCB}$</td>
<td>Load removed by PCB</td>
<td>W/m²</td>
</tr>
<tr>
<td>q</td>
<td>Chilled water flow rate</td>
<td>L/s</td>
</tr>
<tr>
<td>$R_{sk}$</td>
<td>Skin radiation</td>
<td>W</td>
</tr>
</tbody>
</table>
\( s \) Water vapor generation rate per occupant [g/s]

\( S_\phi \) Source

\( \tau \) Air temperature [°C] or [°F]

\( T_{dp} \) Dew point temperature [°C] or [°F]

\( \Gamma_{\phi,\text{eff}} \) Turbulent diffusion coefficient

\( T_s \) Air temperature at supply [°C] or [°F]

\( T_{sp} \) Setpoint temperature in room [°C] or [°F]

\( t \) Time [s]

\( T_u \) Turbulence intensity [-]

\( u \) Air velocity [m/s]

\( u^* \) Normalized velocity [-]

\( VE \) Ventilation effectiveness [-]

\( \chi_i \) Coordinates in i direction [m]

\( \Delta T_{ha} \) Air temperature difference between head and ankle [°C] or [°F]

\( \theta \) Normalized temperature [-]

\( \rho \) Density [kg/m³]

\( \tau \) Mean age of air [s]

\( \eta \) Percentage of load removed by PCB [-]

\( \nu \) Kinematic viscosity [m²/s]

\( \nu_t \) Turbulent eddy viscosity [m²/s]

\( \Phi \) Relative humidity [-]

\( \omega \) Absolute humidity [g/kg]
1. Introduction

1.1 Statement of the Problem

According to the Environmental Protection Agency (EPA), an average American spends 93% of his or her lifetime indoors. It is reported, however, about 30 to 70 million people in U.S. suffer from indoor air quality (IAQ) related “sick building syndromes” that include drowsiness, dizziness, headache, eye irritation, itching skin, allergies and recurrent fatigue. Besides, thermal comfort problems, such as draft, were shown to adversely affect individuals’ performance in thinking, typing and skilled manual work (Wyon 2000). These above-mentioned issues are mainly caused by unsatisfactory distributions of indoor air velocity, air temperature and airborne contaminant distributions. Therefore, maintaining good air quality and acceptable thermal comfort levels in indoor environments is not only critical for occupants’ health, but also essential for their productivity at work. To achieve these goals, an optimal heating, ventilation, and air conditioning (HVAC) system is needed to meet ventilation and thermal needs of an indoor space.

Meanwhile, the annual building energy consumption in US in 2017 was 38 quadrillion British thermal units, which increased by 46% from 1980. Among this total building energy consumption, 44% was for HVAC systems. The Department of Energy (DOE) also reported that residential and commercial buildings sector accounts for 39% of greenhouse gas emissions, which is more than any other sectors in the US. Therefore, development of energy-efficient HVAC systems is vital for reducing overall consumptions of fossil fuels such as coal and natural gas as well as maintaining sustainable development of human society. In the long term, increasing energy efficiencies in buildings is also critical for forestallment of climate change. Hence, in the design of a HVAC system, strategies should be implemented to not only meet thermal and ventilation requirements, but also to minimize its impact on the environment by reducing energy utilization.

1.2 Displacement Ventilation

Among the various HVAC systems, displacement ventilation (DV) has been thoroughly studied in the past forty years since its first application in Scandinavian countries. This system was originally used in industrial facilities, such as welding factories, to provide good indoor quality and to save energy and was later extended to other applications. It was reported that in Nordic countries, half of industrial buildings and 25% of offices were equipped with displacement ventilations by 1989.
In the US, there is also an increasing interest for applying displacement ventilation in buildings, especially in computer rooms, classrooms and large offices (Burt 2007).

In a typical DV system, fresh air is supplied from lower part of the room and exhausted near ceiling, as shown in Fig. 1.1. The discharged cool air first spreads horizontally along the floor due to negative buoyancy and then rises after it is heated by thermal plumes generated by occupants, equipment, and other heat sources. As a result, thermal stratification is formed in an indoor space with a DV system. Since supply air is discharged to occupied zone directly, the supply air temperature must not be too low. Lau and Chen (2007) indicated the supply air temperature in a DV system is normally higher than 16 °C, while supply air temperature in a MV system could be as low as 5 °C (Titus HVAC 2013). This feature might limit the cooling capability of a DV. Some studies (Alain et al. 2012; Rees and Haves 2013) indicated that a DV system can only provide acceptable thermal comfort when the corresponding cooling load is less than 40 W/m². In prior practices (Niu and Kooi 1994), chilled ceiling panels were also used to enhance the cooling capability of a DV system.

When the supplied air moves up from occupied zone, it also brings airborne contaminants upwards, which creates a contaminant stratification. Hence, a DV system can provide better air quality in occupied zone than a corresponding MV system. This improved ventilation efficiency also implies a reduction of fresh air supply, which saves the operating cost. Due to these merits of DV, ASHRAE has promoted this system and developed guidelines for its design (Chen and Glicksman 2003; ASHRAE 2013b).
1.3 Chilled Beams

Chilled beams were first developed in Europe and later emerged in the US as an alternative to conventional systems such as variable air volume (VAV) systems (Pope and Leffingwell 2010). Different from chilled ceiling systems, chilled beams cool the indoor spaces mainly through convection instead of radiation. There are mainly two types of chilled beams systems (Fig. 1.2): active and passive. In passive chilled beams (PCB), sheet metal encloses cooling coils on four sides, while air can enter the PCB through the top surface and exit through the bottom. In active chilled beams (ACB), ducted primary air is mixed with part of room air and reconditioned by cooling coils before it is supplied to an indoor space. Compared with PCB, ACBs are much more complex and expensive.

![Fig. 1.2 Photos of active and passive chilled beams](a) A typical PCB (Price Industries 2017)  (b) A typical ACB (TROX Technik 2017)

Research showed that PCBs can be used to remove large cooling loads. Fredriksson and Sandberg (2009) showed that a cooling efficiency (percentage of heat removed by the system) of 80% could be achieved by PCBs. Other advantages of PCB include low noise level, less maintenance and reduction in energy use. However, since PCB system only provides sensible cooling, it needs to be combined with an air system to meet ventilation requirement and to remove latent load of an indoor space. Hence, there is a potential to couple DV with PCB to merges the advantages of the two systems.

1.4 Research Objectives and Procedure

DV system can improve air quality in enclosed environment but cannot remove high cooling load. On the other hand, PCB system can remove high cooling load and save energy but needs to be used with a ventilation system. Therefore, the major objective of this research project was to
systematically evaluate the overall performance of the coupled DV-PCB system, and to develop a
guide for its design and operation. The developed guideline should be able to provide engineers
with a step-by-step procedure for designing a DV-PCB system that can meet thermal and
ventilation needs without risk of condensation on PCB coils. To achieve those goals, this
investigation conducted the following studies:

- Review of the state-of-the-art on DV and PCB in terms of thermal comfort, IAQ and energy
consumption. Identification of key parameters that could have an impact on the
performance of a coupled DV-PCB system.
- Setting up a DV-PCB system in an environmental chamber and performing measurements
to collect high-quality air velocity, air temperature and contaminant distribution data.
- Development of a CFD model for predicting the air velocity, air temperature and
contaminant distribution and validation of the model using the measured data.
- Using the validated CFD model to simulate additional cases that covered a wider range of
design parameters (cooling load, room geometry, indoor space layout and percentage of
cooling load removed by PCB etc.) of a DV-PCB system. These design parameters and
predicted thermal comfort and indoor air quality levels formed a database for further
analysis.
- Analysis of the established database and development of mathematical models needed to
develop a design guideline
- Development of design guideline to assist engineers for designing an optimal DV-PCB
system
- Energy analysis of the DV-PCB system and comparison between this coupled system with
a conventional all-air system.

In this report, Chapter 2 reviews prior researches on DV and PCB systems and presents the research
gaps based on the literature review. Chapter 3 reports our efforts on developing the experimental
plan, setting up the measurement system and data acquisition. Chapter 4 details CFD modeling
specifics and the comparison between predicted results with measured data. Chapter 5 introduces
the specifics of the CFD database established with the validated CFD model. This chapter also
illuminates how simple mathematical models were developed for predicting the thermal comfort and
IAQ in a DV-PCB system. Chapter 6 presents a step-by-step procedure for designing a DV-PCB
system that can meet thermal and ventilation needs. A design tool that automates the design process
is also illustrated. Chapter 7 shows the energy performance of the DV-PCB system in various US
climate zones and compares it with a conventional DV system. Lastly, Chapter 8 summarizes the research results and concludes the research findings achieved from this project.
2. Literature Review

This section presented our review on existing studies of thermal comfort, IAQ and energy consumption of DV and PCB systems. This chapter also introduced how these two systems were designed in indoor spaces according to prior studies and identified the gaps for further research.

2.1 Displacement Ventilation System

A DV system supplies fresh air from lower part of the room and exhausts return air near the ceiling. This feature brings relatively higher potential of draft at floor level and creates thermal and contaminant stratification in the room. Hence, many previous studies focused on the airflow, air temperature and contaminant distribution in indoor spaces with DV systems. Other studies also compared the energy consumption of DV systems with mixed mode ventilation systems.

2.1.1 Airflow and temperature distributions

Earlier studies (Baturin 1972; Nickel 1990; Nielsen 1993; Skistad 1994) used experimental and theoretical analysis methods to investigate the airflow characteristics and temperature distributions of DV system. Airflow in DV system is driven by both inertial force and buoyancy force, which stems mainly from discharged air from DV diffusers and thermal plumes from heated surfaces in the room respectively. Nielsen (1993) expressed $u_{\text{max},x}$, the maximum velocity in the secondary region of DV diffuser-induced jet flow, with the following expression:

$$u_{\text{max},x} = K_{dr} \left( \frac{h}{x} \right) u_f$$  \hspace{1cm} (2.1)

where $u_f$ is the face velocity of the diffuser, $h$ is the diffuser height, $x$ is the distance from the diffuser and $K_{dr}$ is a coefficient that is dependent on diffuser geometry but is also a function of Archimedes’ number (Ar). The research also presented $K_{dr}$ that was measured from various diffusers (Fig. 2.1).
Baturin (1972) studied the airflow induced by heated object and used the following expression to calculate entrained airflow rate in a thermal plume:

$$V = 0.005 Q_c^{1/3} (y + y_0)^{5/3}$$

(2.2)

where $Q_c$ represents convective heat emitted from heated object and $y + y_0$ is the distance from the virtual origin of the flow.
As for temperature distributions, various experimental measurements were performed to obtain vertical temperature profiles in a room with displacement ventilation. Yuan et al. (1998) summarized multiple measurements as shown in Fig. 2.3, where \( \theta = (T - T_s) / (T_e - T_s) \). Although temperature stratification was observed in all these measurements, the results also showed that the vertical temperature profiles were not linear. \( \theta_{1.1m} \), or the ratio of temperature difference between standing height and supply air to the difference between exhaust air and supply air, ranged from 0.4 to 0.85 in these experiments. The variation of \( \theta_{1.1m} \) could result from differences of a lot of parameters such as ventilation rate, cooling load in occupied zone and space height etc.

With the development of numerical techniques and rapidly increasing computational power, it became common for more recent studies to employ combined experimental and CFD methods to explore the airflow and temperature distributions in spaces with DV systems (Yuan et al. 1999a; Loomans 1998; Gilani et al. 2016). For example, Yuan et al. (1999a) measured the airflow patterns using smoke visualization technique and hot-sphere anemometers in a DV system and also calculated the airflow distribution with CFD. The comparison in Fig. 2.4 indicated that the CFD method predicted the airflow distribution with good accuracy. A validated CFD model not only has the flexibility to be used in studying indoor spaces with different layouts and sizes, but is also valuable in helping researchers and engineers visualize and better understand the corresponding airflow distributions.

Fig. 2.3 Vertical temperature profiles in a room with DV system (Yuan et al. 1998)
2.1.2 Thermal comfort

Since temperature stratification exists in indoor spaces with DV systems, the resulting vertical temperature gradient could cause thermal discomfort. Increasing ventilation rate could mitigate the large temperature gradient problem but could also lead to increased draft rate at the floor level. Hence, most studies in literature that studied thermal comfort in DV systems focused on the corresponding temperature gradient and draft rate, although some investigations also analyzed the predicted mean vote (PMV) distributions in indoor spaces with DV systems.

**Vertical temperature gradient**

Large vertical temperature gradients in DV systems were reported in a lot of previous investigations. For example, Loveday et al. (1998) reported a temperature difference of 4 K between head (sitting height) to feet from measurements in a room with DV system whose ACH was 3.9 h⁻¹. Rees and Haves (2013) measured the vertical temperature profile in a room of 800 W heat gain and found that the temperature difference from head to feet for a seated person was 3.4 K. Both of these temperature differences exceeded the limit recommended by ASHRAE (2013). Mundt (1995) also showed the concern of large temperature gradient in DV systems using experiments and concluded that the gradient was very much dependent on ventilation rate.

This large temperature gradient in occupied zone hinders the capability of DV systems to remove large cooling loads. In fact, some studies (Alain et al. 2012; Rees and Haves 2013) recommended a DV system to be limited to indoor spaces where the corresponding cooling loads are less than 40 W/m² to ensure thermal comfort. On the other hand, some other studies also coupled chilled ceilings with DV systems to enhance its ability to remove large cooling load. For example, Loveday et al. (1998) measured the vertical temperature difference in the coupled system and concluded that
thermal comfort could be achieved when the heat source was as high as 62 W/m². Schiavon et al. (2012) found that chilled ceiling significantly reduced the vertical temperature gradient in a room developed by DV system, and the temperature difference in occupied zone decreased with the percentage of the cooling load removed by chilled ceiling. Coupled chilled ceiling and displacement ventilation has been shown to be quite effective in indoor spaces with high cooling loads, although the cost of chilled ceiling system could be very high (Weidner et al. 2009).

Draft rate

Since DV supplies cold air to indoor space from lower part of the room, multiple studies reported the high draft risk at floor level, especially near the diffusers. For instance, Yu et al. (2006) conducted a subjective study in a field environment chamber with a DV system and tested 30 males and 30 females. Their results showed that the draft risk was mainly on lower body segments and was affected by the overall thermal sensation. Melikov et al. (2005) studied the responses from over 200 occupants regarding the thermal environment in 10 Danish office buildings, and the results showed that 24% complained they were daily bothered by draft, mainly on lower leg. For this reason, Yuan et al. (1999b) recommended the face velocity at a DV diffuser to be smaller than 0.2 m/s in the design of a DV system.

On the other hand, although the draft risk at floor level in a DV system could be high, the numerical comparative study by Lin et al. (2005) showed that the overall draft rate in a room with DV system was lower than a corresponding MV system. Their research found that the average percentage dissatisfied due to draft (PD) in occupied zone was less than 10% for office, classroom and retail shop spaces with DV systems and 20% for the same spaces with MV systems. This was because the air movement in the occupied zone was mainly driven by buoyancy force in a DV system but more significantly driven by inertial force in a MV system.

PMV and PPD distributions

Although most studies used temperature gradient or draft rate to appraise the thermal comfort in DV systems, predicted mean vote (PMV) and predicted percent dissatisfied (PPD) were also adopted as one index to evaluate thermal comfort level. For example, Yuan et al. (1999b) used a validated CFD model to calculate the distribution of PPD at different cross-sections in a classroom with 24 students as shown in Fig. 2.5. Their results showed that PPD was lower than 10% in most region of the classroom, except in region close to DV diffusers where the PPD could be around 20%. It should be noted that one can always adjust the clothing level to adjust the PMV and PPD.
results. However, these results still illustrated clearly the spatial variation of predicted thermal sensation level in a DV system.

Fig. 2.5 PPD distribution in a classroom with 24 students (unit: %) (Yuan et al. 1999b)

(a) At ankle level

(b) At cross section A-A

2.1.3 Contaminant distribution

DV system provides an indoor space with stratified contaminant distribution and hence improves the air quality in occupied zone. Fig. 2.6 shows typical vertical contaminant concentration profiles in a DV system at various ventilation rates (Heiselberg and Sandberg 1990). Chen et al. (1988) showed that when contaminant source was combined with heat source, the ventilation efficiency of a DV system increased with cooling load or ventilation rate. Other studies (Skistad 1989; Niemela and Koskela 1996) found that the ventilation efficiency in a DV system increases with ceiling height.
A lot of more recent studies (Lau and Chen 2007; Yin et al. 2009; Xu et al. 2009) used combined experimental and CFD methods to investigate contaminant concentrations in DV systems. In the experiments, sulfur hexafluoride (SF₆) and carbon dioxide (CO₂) were two commonly used tracer gases for simulating airborne contaminants. CFD modeling calculated airborne contaminant concentration using the Eulerian method and the corresponding transport equation was solved to obtain mean contaminant concentration field. For example, Yin et al. (2009) simulated the airborne contaminant distribution in a DV system and compared the predicted results at three representative locations with measured data (Fig. 2.7). Contaminant stratification were observed in both measured and simulated results, and the CFD model predicted the contaminant stratification with reasonable accuracy.
2.1.4 Energy and first cost analysis

Since the exhaust air temperature in a DV system is higher than that in occupied zone, DV system has the potential to save energy. Several previous studies analyzed the energy consumption in DV systems and compared the results with that in MV systems, although different analysis methods were used. Chen and van der Kooi (1990) employed the energy analysis program ENERK to analyze the annual energy consumption in an office based on weather data of the Dutch short reference year and found there was a significant energy saving in a DV system as compared with a well-mixed ventilation system. Hu et al. (1999) further compared the energy consumption in DV and MV systems at five different US climate regions. Their results indicated that DV system may use more fan energy but less chiller energy than a DV system, and the overall cost of a DV system was cheaper if it was used in the core region of a building. Lau and Chen (2006) analyzed the energy consumption of floor-supply DV system and concluded that DV system saves energy because of improved chiller COP and longer free cooling period by outside air. However, their study also indicated that a DV system, if used in humid climate, might lead to high humidity level in the room since DV might not have enough capability to dehumidify outdoor air at a higher cooling coil temperature.

Some other studies also investigated the first cost of a DV system and compared it with that of a MV system. For example, Skitad (1994) reported that the difference between costs of a DV system and a corresponding MV system was not significant, although the cost of DV diffusers was higher than MV diffusers. Seppanen et al. (1989) also showed that costs of DV systems were similar as MV systems, except when chilled ceilings needed to be added to DV systems, which could greatly
increase the cost. In addition, Hu et al. (1998) found that DV might need a larger air handling unit, a smaller chiller and a similar size of boiler as compared to MV and concluded that the first cost of a DV was lower than MV if they were used in core region of a building. Overall, the cost of a DV-Only system was found to be not much higher than a MV system.

2.2 Passive Chilled Beam System

PCB system cools indoor air through the enclosed cooling coil, which can significantly affect the original airflow pattern developed by the air system. PCB removes sensible cooling load through chilled water and cannot remove latent load in indoor spaces, so the mechanism to remove indoor loads and the corresponding energy consumption could be very different from those of conventional air systems. Besides, since PCBs are installed near ceiling above occupied zone, condensation is a concern and strategies need to be implemented to avoid it from occurring. Hence, this project reviewed the existing studies on PCB system from the following three aspects: airflow, energy and cost, and risk of condensation.

2.2.1 Airflow induced by PCB

Fredriksson et al. (2001) introduced smoke above PCB and used laser light to visualize the airflow pattern beneath PCB. The airflow generated by PCB exhibited a two-dimensional jet flow that entrained surrounding air along the downward plume centerline. Meanwhile, their research used an anemometer system to quantitatively measure the airflow magnitude beneath PCB and found the maximum local velocity was close to 0.3 m/s. Kosonen et al. (2010) further used smoke visualization method to compare the airflow patterns when the heat dummy was beneath PCB or away from PCB in an office mock-up. Their results indicated that if the heat dummy was placed under PCB, its upward thermal plume significantly affected the PCB-induced downward jet flow. Their measurements showed that dummy could reduce the downward air velocity 10 cm above the dummy top surface by 0.1-0.15 m/s.

The increased air velocity and reduced air temperature inevitably enhance the draft risk under PCB. Based on the measured air velocity and temperature, Kosonen et al. (2010) reported that the draft rate at 1.2 m under PCB could be over 25%, which exceeded the recommendations of draft ratings from ASHRAE 55 and EN 13779. If a heat source was placed under PCB, it would decrease the strength of downward jet and hence reduce corresponding draft rate.
2.2.2 Energy and cost

Roth et al. (2007) summarized a few aspects where PCB can save energy. First, PCB system significantly reduces the amount of supply air needed for the indoor space, which results in reduced fan energy consumption. Second, since outdoor air is only needed to meet ventilation requirement in a PCB system, it also reduces the energy needed to condition outdoor air. Third, PCB system can use higher chilled water temperature (13 – 17°C) than conventional all-air system, the chiller dedicated for PCB could operate at a higher COP than an air system. Further, Kim et al. (2014) used experimental and simulation methods to compare the energy consumptions between a PCB system and a conventional all-air system by accounting for detailed internal gain schedules. The experimental measurements were performed in two full-scale open plan offices over a summer cooling season. Their results showed that a PCB system could save 10-12% of total energy, with 5-6% from the chiller and the rest from the fan. Using separate chillers for the PCB section and section of the PCB could increase the saving to 23%. The research results aligned with the findings in Roth et al. (2007).

Prior studies also investigated the first cost of a PCB system and compared it with that of an all-air system, which is crucial for evaluating the market value of PCB. Weidner (2009) evaluated and systematically compared three HVAC options for a Kentucky project of constructing a 376,000 ft² (34,900 m²) as shown in Table 2.1. Results showed that adopting the option of floor-supply DV with PCB system costs $100,000 more than the option of a single floor-supply DV system. The additional cost mainly originated from the extra expense associated with PCB equipment and labor. However, the option of floor-supply DV with PCB system still cost much less and has a much shorter payback period than a floor-supply DV with cooled ceiling system.
Table 2.1 Comparison of floor-supply DV system with or without water cooling systems
(Weidner 2009)

<table>
<thead>
<tr>
<th></th>
<th>UFAD Alone</th>
<th>UFAD With Radiant Cooled Ceilings</th>
<th>UFAD With Passive Chilled Beams</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply Air Quantity (cfm)</td>
<td>560,000</td>
<td>240,000</td>
<td>240,000</td>
</tr>
<tr>
<td>Supply Fan Power (hp)</td>
<td>600</td>
<td>280</td>
<td>280</td>
</tr>
<tr>
<td>Return Fan Power (hp)</td>
<td>280</td>
<td>120</td>
<td>120</td>
</tr>
<tr>
<td>Total Swirl Diffusers Required</td>
<td>5,600</td>
<td>2,400</td>
<td>2,400</td>
</tr>
<tr>
<td>Weighted Airflow (cfm/ft²)</td>
<td>1.6</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>Qualitative Flexibility</td>
<td>Good</td>
<td>Fair</td>
<td>Good</td>
</tr>
<tr>
<td>First Cost ($)</td>
<td>Reference</td>
<td>+4,250,000</td>
<td>+100,000</td>
</tr>
<tr>
<td>Operating Cost Payback</td>
<td>N/A</td>
<td>&gt;50 years</td>
<td>&lt;2 years</td>
</tr>
</tbody>
</table>

2.2.3 Condensation in PCB

Previous studies employed different methods to avoid condensation from forming on PCB coil surface. Fredriksson and Sandberg (2009) and Vastyan et al. (2011) suggested lifting chilled water temperature to control condensation. In order to meet cooling requirement, the chilled water flow rate needs to be increased accordingly. DOAS could also be managed to control the humidity level, and hence the dew point in the indoor space (Roth et al. 2007). In addition, advanced technologies were used in some PCB systems to control condensation. For example, Andrew and Farshad (1997) installed three condensation detectors on chilled coil surface to monitor the temperature and predict condensation formation. If condensation is detected, a mechanism is activated to stop the condensation from further accumulation. There is always a trade-off between implementing these condensation control strategies and increased cost due to additional equipment installation or energy consumption. However, if condensation is not well controlled, it may continually form and drip onto occupants or objects in occupied zone. More seriously, it may even lead to mold growth which can escalate to air quality problem (Mumma 2003; Brennan and Burge 2005). Hence, it is worthwhile and critical that proper strategies be adopted to avoid condensation risk.
2.3 Guides to Design

A proper design guideline is critical to make sure the selected design parameters can create a thermally comfortable and healthy indoor environment. Several guides from literature could be found for designing a DV system. For example, for a DV system, Skistad (1994) proposed the following design guideline:

- **Step 1** - Determine the needed airflow rate according to total cooling load and difference between air temperatures at supply and exhaust.

- **Step 2** - Determine the required air flow rate based on ventilation standard.

- **Step 3** - Select the greater airflow rate of the values determined in Step 1 and Step 2 to be the final supply airflow rate.

- **Step 4** - Find the supply air temperature based on the assumptions of constant temperature gradient and $\theta_f = 0.5$.

- **Step 5** – Select appropriate diffusers based on manufacturer’s manual to avoid draft risk.

Furthermore, Lee (2011) established a 6-step design procedure for designing a floor-supply DV system. This design guide was developed based on cooling load, floor area, room height, setpoint temperature etc. as follows:

- **Step 1** – Judge applicability of a floor-supply DV system.

- **Step 2** – Calculation of cooling load to be removed.

- **Step 3** – Estimation of required airflow rate based on thermal comfort.

- **Step 4** – Identification of minimum flow rate for acceptable indoor air quality.

- **Step 5** – Determination of supply airflow rate and floor-supply diffuser type and number.

- **Step 6** – Calculation of temperature difference between head and ankle of an occupant.

There are very limited published articles that proposed established guideline for designing a PCB system but PCB manufacturers normally provide PCB properties in their users’ manuals, which can be used in design. For example, TROX (2017) provides datasheet that demonstrates the cooling load removal rates at different chilled water flow rate and chilled water temperature for each of
their PCB products. Such information is important for a designer to select appropriate PCB specifications when they know the maximum cooling load to be removed by PCB system.

However, all existing design guidelines were developed for a separate DV system or PCB system. For a coupled DV-PCB system, the performances of both systems interplay with each other. Hence, a different design guideline should be established and followed if a DV-PCB system is to be designed for an indoor space.

2.4 Research Gaps

Previous investigations have studied DV system and PCB system from various aspects. However, there still exist research gaps that deserve further investigations. Below are a few research gaps identified based on the literature review.

(1) DV system cannot remove high cooling load but coupling it with a PCB system could potentially resolve this limitation of DV. Hence, it is worthwhile to conduct systematic investigation on the thermal performance of a coupled DV-PCB system. Such an investigation could answer questions such as how well a coupled DV-PCB system can remove high cooling load, how PCB change the temperature gradient in a DV system and what is the draft rate is in this coupled system.

(2) Although PCB might be able to increase the cooling capability of a DV system, it could also recirculate the airborne contaminants downwards to the occupied zone, which negatively affects the contaminant stratification developed by a DV system. Hence, research needs to be performed to study how significantly the contaminant stratification of a DV system could be damaged by the added PCB system. This investigation could enhance the understanding the ventilation performance of the coupled DV-PCB system.

(3) The thermal and ventilation performance of a coupled DV-PCB might depend on a lot of design parameters. Therefore, a proper design guide needs to be developed for designers to find out the appropriate design specifics, which can provide an indoor space with satisfying thermal comfort and good air quality. Meanwhile, the design should also incorporate strategies to avoid condensation from occurring on PCB coil surface.

(4) Energy consumption analysis of this coupled system should be performed and compared with that of the conventional all-air system. Such an analysis should be conducted at different climate zones in the US. Adding a PCB system to a DV system would inevitably increase its first cost.
However, the coupled system can save energy so it may be economically advantageous in the long term. These results are beneficial to evaluate the market value of a coupled DV-PCB system.

These research gaps were investigated in this research project.

2.5 Conclusions

This study reviewed literatures on DV and PCB systems over the last thirty years. Previous studies investigated DV system on airflow distribution, contaminant distributions, thermal comfort, and energy performance, while PCB-related investigations included PCB-induced airflow, energy analysis and condensation control. Few studies were performed to investigate thermal and ventilation performance of a coupled DV-PCB system. This research project used experimental and computational methods to systematically evaluate the performance of DV-PCB system and established a design guideline. The results would help designers to find appropriate design parameters that provide an indoor space with acceptable thermal comfort and air quality without condensation risk.
3. Experimental Setup and Measurements

Studies of indoor air environments can take two approaches: experimental measurements and numerical simulations. Experimental measurements provide straightforward indoor air environment information such as air velocity and air temperature, but constructing the experimental apparatus and running experiments can be quite time-consuming and costly. Moreover, the dimensions of a test chamber are usually fixed, and thus the size of the investigated indoor space is limited (Gilani et al. 2016; Lee et al. 2009a; Lee et al. 2016). With the rapid development of computer processing power, the computational fluid dynamics (CFD) method has also been widely used in the investigation of indoor air environments. For instance, in some researches (Gilani et al. 2016; Buratti et al. 2017; Shi et al. 2016) it was used for predicting indoor air velocity and temperature. In other studies, it was also used to simulate air quality related parameters such as gaseous contaminant (Barbosa and Brum 2018; Van Hooff and Blocken 2016; Sadrizadeh and Ploskic 2016), volatile organic compound (Rai et al. 2014) or particle concentrations (Chen and Zhao 2017). This method numerically solves the governing equations, and the results can provide a more comprehensive picture of the airflow at much lower cost than direct measurements. However, CFD uses models to approximate flow and heat transfer physics, which could lead to errors (Coleman and Stern 1997; Lau and Chen 2007). Hence, it is vital that experimental data be used to validate CFD results and that the appropriate approximations are used in the CFD method.

Hence, this investigation first constructed a coupled DV-PCB system in a full-scale environmental chamber to measure air velocity, air temperature and airborne contaminant concentration. The measurement data was then used for validating a CFD model that is used for simulating air distribution for rooms with DV-PCB systems. This section detailed the experimental setup and measurement method.

3.1 Test Chamber

In this research, we conducted all the measurements in an environmental chamber as shown in Fig. 3.1 in Herrick Labs at Purdue University.

In order to obtain high-quality experimental data for validating the CFD model, this study constructed a coupled DV-PCB system in this chamber as shown in Fig. 3.2. The chamber had dimensions of 6.08 m in length, 5.15 m in width, and 3.05 m in height. In this chamber, fresh air was supplied through two diffusers located at floor level at the corners of one of the side walls, and exhausted through an outlet on the opposite wall near the ceiling. The chamber contained tables;
heated boxes with dimensions of 0.41 m × 0.41 m × 1.13 m and 84 W power, each simulating a seated person; and heated boxes with dimensions of 0.41 m × 0.25 m × 0.51 m and 109 W power, each simulating a personal computer (PC). Heat was generated by light bulbs installed inside these boxes, and mini-fans were used to stir the inside air, so that the heated air could circulate. On the ceiling of the chamber, there were four lights. The number and locations of the above items could be changed, and thus the environmental chamber could be used to simulate different cooling loads and various room layouts. For example, Fig. 3.3(a) illustrates an office layout where simulated occupants are seated back to back at tables, Fig. 3.3(b) shows a classroom layout where simulated occupants are facing the same direction, and Fig. 3.3(c) is a conference room layout in which simulated occupants sit around tables. The circled numbers in the figure are the locations for measuring the air parameters.
3.2 Air Handling System and PCBs

Fig. 3.4(a) shows the air handling system of the environmental chamber, including a cooling coil, two heaters and two variable-speed fans. With the use of these components and control software, the supply-air flow rate and temperature could be adjusted as needed. Fig. 3.4(b) is the plumbing system for the PCBs that were installed inside the chamber. Chilled water was supplied from a water reservoir to beams. For each PCB, a flowmeter (OMEGA FTB-101, with ±0.1% accuracy) was used to monitor the water flow rate, which could be varied by turning the valve. Meanwhile, supply and return water temperatures for each PCB were measured. The water temperature could also be adjusted in the building automation system. In this study, the supply water temperature was controlled in the range of 12 - 18°C, and the maximum waterflow rate for each beam was 13 L/min. By combining water supply temperature, return temperature and flow rate, one can calculate and adjust the heat removal rate for each PCB.

Fig. 3.4(c) and Fig. 3.4(d) show a photograph and a 3D model of a typical PCB, respectively. Sheet metal encloses cooling coils on four sides, while air can enter the PCB through the top surface and exit through the bottom. Detailed dimensions of PCBs vary from manufacturer to manufacturer.
This study used three types of PCB (labeled A, B and C) made by three different companies, and Table 3.1 lists their dimensions.

Fig. 3.4 Schematic of coupled DV and PCB system

(a) Air handling system  
(b) Chilled water supply to PCB

(c) Photograph of a typical PCB  
(d) 3D model of detailed geometry of a PCB

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Table 3.1 Dimensions of the three types of PCB investigated in this study

<table>
<thead>
<tr>
<th>Type</th>
<th>Length</th>
<th>Width</th>
<th>Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type A</td>
<td>2.52 m</td>
<td>0.25 m</td>
<td>0.58 m</td>
</tr>
<tr>
<td>Type B</td>
<td>1.98 m</td>
<td>0.20 m</td>
<td>0.30 m</td>
</tr>
<tr>
<td>Type C</td>
<td>2.94 m</td>
<td>0.12 m</td>
<td>0.43 m</td>
</tr>
</tbody>
</table>

3.3 Experimental Instruments and Methods

Air velocity, air temperature and contaminant concentration are important pieces of information in regard to indoor air quality and thermal comfort, and were thus measured in the current study. Air velocity and temperature were measured using hot-sphere anemometers, the accuracy of which was ± 0.01 m/s for velocity and ± 0.3°C for temperature. The hot-sphere anemometers have measurement frequencies of 1 Hz. This investigation released a tracer gas (sulfur hexafluoride, or SF₆) above a simulated occupant to simulate contaminant emission by the occupant, and measured the tracer-gas concentration in the chamber using a multi-point sampler (LumaSense Technologies 2017a) and a photoacoustic multi-gas analyzer (LumaSense Technologies 2017b), with minimum reading of 0.01 ppm. Measurements were conducted along 5 poles that were evenly distributed throughout the chamber, with 7 measurement heights on each pole. The distance between adjacent measurement locations on the same pole was 0.46 m. This study also used an infrared thermometer (with reading repeatability of ±0.02% of absolute temperature) to measure surface temperatures. Before each set of measurements, this investigation operated the HVAC system for more than two hours to reach a steady state.

During the experiment, three computers (1, 2 and 3) were placed outside of chamber for experimental operation and data collection. Computer 1 was used to control air handling and PCB systems via a building automation system. Computer 2 was for reading air velocity and temperature results through a LabView program. Computer 3 was set up for obtaining SF₆ concentration readings. All the operations and data monitoring were performed in a non-intrusive way in order not to affect airflow and contaminant concentration inside chamber.
3.4 Specifics of Measurement Cases

In the measurements, this research investigated four design parameters that are important for the airflow and contaminant distribution in the coupled system: PCB type, room layout, cooling load, percentage of cooling load removed by PCB. With the orthogonal method (Taguchi and Rajesh 2000), this study identified the following 9 test cases which contain three levels for each parameter.

Table 3.2 Specifics of measurement cases

<table>
<thead>
<tr>
<th>Case #</th>
<th>PCB type and number</th>
<th>Room layout</th>
<th>Cooling load (W/m²)</th>
<th>Percentage of cooling load removed by PCB (η)</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1</td>
<td>2A</td>
<td>Classroom</td>
<td>30</td>
<td>40%</td>
</tr>
<tr>
<td>T2</td>
<td>2A</td>
<td>Office</td>
<td>61</td>
<td>60%</td>
</tr>
<tr>
<td>T3</td>
<td>2A</td>
<td>Conference Room</td>
<td>90</td>
<td>80%</td>
</tr>
<tr>
<td>T4</td>
<td>2B</td>
<td>Office</td>
<td>30</td>
<td>80%</td>
</tr>
<tr>
<td>T5</td>
<td>2B</td>
<td>Conference Room</td>
<td>61</td>
<td>40%</td>
</tr>
<tr>
<td>T6</td>
<td>2B</td>
<td>Classroom</td>
<td>90</td>
<td>60%</td>
</tr>
<tr>
<td>T7</td>
<td>3C</td>
<td>Conference Room</td>
<td>30</td>
<td>60%</td>
</tr>
<tr>
<td>T8</td>
<td>3C</td>
<td>Classroom</td>
<td>61</td>
<td>80%</td>
</tr>
<tr>
<td>T9</td>
<td>3C</td>
<td>Office</td>
<td>90</td>
<td>40%</td>
</tr>
</tbody>
</table>

During each test, in addition to obtaining measurement data that is mentioned in Section 3.3, this study also collected heated surface temperature, supply air flow rate, supply air temperature and tracer gas flow rate which are important boundary conditions for CFD modeling. The measurement data and corresponding measured boundary conditions can be found in Appendix 1.

3.5 Conclusions

To obtain reliable measurement data for CFD model validation, this research constructed a DV-PCB system in an environmental chamber. The experimental setup could be used to measure air velocity, air temperature and tracer gas concentration at 35 evenly distributed locations in the chamber for 9 test cases. Thermal and fluid boundary conditions were also obtained. These experimental datasets are valuable for validating numerical simulation model of a DV-PCB system.
4. CFD Modeling, Validation and Result Analysis

One of the objectives of this research project was to develop a practical guide for designing and operating the coupled DV-PCB systems. To achieve that, a validated CFD model was used to simulate many cases with various design parameters for establishing a database. This chapter illustrated the details of this CFD model and its validations by experimental data obtained in Chapter 3. With this validated CFD model, this chapter also analyzed the thermal comfort and indoor air quality in the enclosed environment with the DV-PCB coupled system using the following indices: vertical temperature gradient, draft rate, predicted mean vote, normalized contaminant concentration and age of air.

4.1 CFD Modeling

To predict the air velocity, air temperature and contaminant concentration in the room, this study developed a CFD model that employed Reynolds-Averaged Navier Stokes (RANS) equations with the Re-Normalized Group (RNG) $k - \varepsilon$ model (Yakhot et al. 1992). This turbulence model was adopted because it had been recommended for predicting indoor airflow (Zhang et al. 2007). With the use of this model, the transport equations for mean values were generalized as:

$$\rho \frac{\partial \langle \phi \rangle}{\partial t} + \rho \langle u_i \rangle \frac{\partial \langle \phi \rangle}{\partial x_i} - \frac{\partial}{\partial x_i} \left[ \Gamma_{\phi,\text{eff}} \frac{\partial \langle \phi \rangle}{\partial x_i} \right] = S_{\phi}$$

(4.1)

where $\phi$ could represent velocity components $u_i (i = 1,2,3)$, turbulent kinetic energy $k$, turbulent dissipation rate $\varepsilon$, energy $E$, or contaminant concentration $C$. The $S_{\phi}$ and $\Gamma_{\phi,\text{eff}}$ stand for the source term and effective diffusion coefficient of scalar $\phi$, respectively. The Boussinesq approximation was used to account for the variation in air density with temperature. One can refer to ANSYS Inc. (2016) for more details about this turbulence model and the Boussinesq approximation.

Proper specification of boundary conditions and appropriate establishment of mesh structure are important for accurate simulation of indoor airflow and contaminant transportation by a CFD model. In this study, non-slip boundary conditions were prescribed on surfaces of heated objects as well as walls. The surface temperatures were obtained from measurements. Each piece of the PCB was modeled as an energy sink (Fadhl et al. 2013). The corresponding heat absorption rate per unit volume was calculated from the heat removal rate and the volume of the PCB. The SF$_6$
source was assumed to have zero momentum since the release amount was minimal. Fig. 4.1 illustrates the mesh structure for this study. A combination of structured and unstructured meshes was used to discretize the computational domain. Inflation layers of structured meshes were employed to capture the relatively large temperature gradients around heated surfaces, while unstructured meshes were used to handle the complicated interior geometry. This study performed a grid-independence study with three grid resolutions: 0.89 million (coarse), 3.51 million (medium) and 8.42 million (fine). The results indicated that a mesh with 3.51 million cells was sufficient to capture the airflow features in the room. Examination of the average velocity magnitudes in a $0.2m \times 0.2m \times 0.2m$ air box beneath PCB showed the difference between results in the “coarse” case and “medium” case was 17%, and that between “medium” case and “fine” case was less than 5%. The corresponding grid size from “medium” case was thus used for further study.

![Fig. 4.1 Mesh structure in the CFD model](image)

This study employed a commercial CFD program, ANSYS Fluent 17.0 (2016), to calculate airflow and contaminant transport. The semi-implicit method for pressure-linked equations (SIMPLE) algorithm was adopted to couple velocity and pressure equations. The second-order method was
used to spatially discretize $u_t$, $k$, $\varepsilon$, $C$ and $E$. This study considered the simulation to be converged when normalized residuals were less than $10^{-6}$ for $E$ and less than $10^{-4}$ for the $u_t$, $k$, $\varepsilon$ and $C$ terms. The air velocity, air temperature and contaminant concentration results from the CFD model were then compared with the experimental data.

4.2 Validation of CFD Model

To ensure that the CFD model yielded accurate predictions that could be used for further analysis, this investigation first used experimental results to validate the model. We collected a large amount of measurement data. However, because of the space limitations in this paper, we have used the results only from Poles 1, 2 and 5 for a representative case (61 W/m², 40% of cooling load removed by two pieces of type B PCB) for the validation. The locations of these poles are shown in Fig. 3.3. Meanwhile, Fig. 4.2 illustrates the air velocity, temperature and contaminant concentration from the experiment and the simulation, with the values normalized as:

$$
\begin{align*}
    h^* &= h / H, \\
    u^* &= u / U, \\
    \theta &= (T - T_s) / (T_e - T_s), \\
    C^* &= (C - C_s) / (C_e - C_s)
\end{align*}
$$

(4.2)

where $H$, $T_s$ and $T_e$ represent room height, supply air temperature and exhaust air temperature, respectively. Here $U$ is a constant velocity of 0.2 m/s.

As shown in Fig. 4.2 (a), the air velocity in the lower part of room was generally higher than that in the upper part. This difference was due to two factors. First, the DV system supplied fresh air to the lower part of room. This was the main reason for the high air velocity along Pole 2 (close to the DV diffusers) at ground level. Second, the downward jet generated by the PCBs impinged with objects in the occupied zone and increased the local air velocity. Fig. 4.2(b) further depicts the air velocity development beneath a PCB. The mean air velocity under the PCB started at a small magnitude but increased continuously until it reached a peak. This increase can be explained by the entrainment effect that drew ambient air to the jet center (Pope 2000; Gurevich 2014). The air velocity then gradually decayed as a result of dissipation of airflow momentum and jet impingement with the objects in the occupied zone. Fig. 4.2(c) shows the normalized temperature distributions along the three poles. Although the PCBs created a downward cold jet that caused local mixing, there still existed a temperature gradient in the bulk region because of the thermal plume. Finally, as shown in Fig. 4.2(d), when PCBs were used, they could disrupt the contaminant stratification generated by DV.
Fig. 4.2 Air velocity, temperature and contaminant distributions in a DV-PCB system

(Symbols: experimental data; lines: simulation data.)

The relative errors between the simulation and experimental results for the air velocity, temperature and contaminant concentration were 26.9%, 14.4% and 20.9%, respectively. The discrepancies can be explained in part by the uncertainties in the measurement positions. The errors caused by such uncertainties may have been particularly significant in locations where the variable gradient was large, such as the SF₆ concentration along Pole 5. Besides, SF₆ transport in the room is quite sensitive to local airflow, which might lead to noticeable local discrepancy between simulation and measurement in SF₆ concentration prediction. In fact, discrepancy of this kind was also reported in
previous indoor airflow and contaminant transport simulation works (Lee et al. 2009a; Lau and Chen 2007). Furthermore, the CFD model employed a large number of approximations in discretization and turbulence modeling and these approximations could also have contributed errors. However, the overall trends in air velocity, temperature and contaminant concentration were still predicted with reasonably good accuracy. Therefore, the CFD model was considered validated and was used for further analysis.

4.3 Characteristics of Airflow Induced by PCBs

Using the validated CFD model, this study examined the airflow characteristics around a PCB, as illustrated in Fig. 4.3. The air inside the PCB was cooled and thus became denser. As a result, the air dropped and generated a downward jet. Depending on its strength, the jet may have reached the floor, or its velocity may have decayed to zero in mid-air. Meanwhile, if heat sources such as dummies were placed at a sufficiently large distance from the PCB, the thermal plumes generated by these heat sources could still have ascended without being affected by the downward jet from the PCB. Besides, the airflow patterns show that if thermal plumes draw gaseous contaminants from occupied zone into upper part, it could be recirculated downwards by PCB-induced air jet, since gaseous contaminant passively follows indoor airflow.

Fig. 4.3(b) also depicts the air temperature contour in the vicinity of the PCB. Warmer air was drawn toward the top of the PCB and was cooled significantly when it passed through the PCB. The cold downward jet discharged from the PCB entrained the ambient air, which was warmer than the air in the center of the jet. Consequently, the temperature at the jet center continued to increase along the centerline as the air in the jet core mixed with the entrained air. Since the temperature of the air jet was still lower than the ambient, the jet then cooled the air in the breathing zone through convection. With the use of the CFD model the mechanisms of PCB influencing local airflow and cooling indoor air were visualized and were thus better understood.
4.4 Thermal Comfort Analysis

In a room with a DV-PCB system, many parameters could affect thermal comfort. To study the effects of PCBs on thermal comfort in a DV system, this investigation simulated 14 cases with various parameters, as listed in Table 4.1. These cases were designed as follows. Cases 1-4 had the same high cooling load, supply air temperature and PCB type, while the overall percentage ($\eta$) of the load removed by the PCBs varied. Therefore, these four cases were used to study the cooling effect of the PCBs on indoor air and the influence of the PCB on the vertical temperature gradient, at different $\eta$. Section 4.4.1 presents the corresponding results. Cases 5-14 had the same medium cooling load and $T_{sp}$ (air temperature at H=1.1m), but used different combinations of PCB types and different $\eta$. When type A or B was used, two pieces of PCB were installed in the mid-section of the room; when type C was used, three pieces were installed in parallel in the room. (Whether it was a 2-piece or 3-piece case, the cooling load removed by each piece was the overall cooling rate of all chilled beams divided by number of PCB pieces.)

Table 4.1 Parameters of the studied cases

<table>
<thead>
<tr>
<th>Case #</th>
<th>Internal load Q (W/m²)</th>
<th>PCB type and number</th>
<th>Percentage of load removed by PCBs ($\eta$)</th>
<th>$T_s$</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>90</td>
<td>N/A</td>
<td>0%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>90</td>
<td>2B</td>
<td>40%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>90</td>
<td>2B</td>
<td>60%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>90</td>
<td>2B</td>
<td>80%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>61</td>
<td>N/A</td>
<td>0%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>61</td>
<td>2A</td>
<td>40%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>61</td>
<td>2A</td>
<td>60%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>61</td>
<td>2A</td>
<td>80%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>61</td>
<td>2B</td>
<td>40%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>61</td>
<td>2B</td>
<td>60%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

$T_{sp}$ = 21.0°C

Adjusted so that $T_{sp}$ = 23°C

Used to study the impact of PCBs on the temperature gradient under a high cooling load

Used to study the impact of PCBs on the PD distribution under a medium cooling load
To appraise the performance of the DV-PCB system, this study first employed the following indices to quantitively evaluate the resulting thermal comfort.

**Vertical Temperature Gradient**

According to the literature (Olesen et al. 1979; Ilmarinen et al. 1992), the vertical air temperature difference between head and ankle creates thermal discomfort for occupants. Therefore, this study examined the temperature difference:

\[
\Delta T_{ha\_seated} = T_{1.1m} - T_{0.1m}, \quad \Delta T_{ha\_standing} = T_{1.1m} - T_{0.1m}
\]

(4.3)

where \( \Delta T_{ha\_seated} \) and \( \Delta T_{ha\_standing} \) represent temperature gradients between head and ankle levels for a seated and standing occupant, respectively.

**Percentage Dissatisfied due to Draft (PD)**

Furthermore, unwanted cooling of the body caused by air movement leads to a draft sensation (Fanger et al. 1989). This study used the following equation to predict the percentage of people dissatisfied due to draft:

\[
PD = (34 - T)(u - 0.05)^{0.62}(3.14 + 0.37uT_u),
\]

(4.4)

where \( u \) is mean velocity, \( T \) is temperature, and \( T_u \) is the local turbulence intensity.

**Predicted mean vote (PMV) and predicted percent dissatisfied (PPD)**

PMV evaluates a 7 point scale of thermal comfort as the energy balance of body (Fanger 1970). It is calculated as:

\[
PMV = \left[0.303e^{-0.036M} + 0.028\right]L,
\]

(4.5)

where \( M \) is metabolic rate, and \( L \) is the thermal load on body, which can be calculated as:

\[
L = M - W - [(C_{sk} + R_{sk} + E_{sk}) + (C_{res} + E_{res})]
\]

(4.6)
In other words, \( L \) represents the difference between internal heat production and heat loss to the actual environment.

The calculated PMV values can be interpreted to thermal sensations using Table 4.2.

<table>
<thead>
<tr>
<th>PMV</th>
<th>Thermal Sensation</th>
</tr>
</thead>
<tbody>
<tr>
<td>+3</td>
<td>Hot</td>
</tr>
<tr>
<td>+2</td>
<td>Warm</td>
</tr>
<tr>
<td>+1</td>
<td>Slightly warm</td>
</tr>
<tr>
<td>0</td>
<td>Neutral</td>
</tr>
<tr>
<td>-1</td>
<td>Slightly Cool</td>
</tr>
<tr>
<td>-2</td>
<td>Cool</td>
</tr>
<tr>
<td>-3</td>
<td>Cold</td>
</tr>
</tbody>
</table>

PPD can further be calculated as:

\[
PPD = 100 - 95 \exp \left( -0.03353 PMV^3 - 0.2179 PMV^2 \right) \quad \% \tag{4.7}
\]

**4.4.1 Influence of PCBs on temperature gradient**

One drawback of a DV system is that it may create a large temperature gradient in the occupied zone, which is not conducive to thermal comfort. Fig. 4.4 depicts the temperature profiles of three cases in which PCBs were used to remove different amounts of cooling load in the room, and illustrates how the PCB-induced cool jet changed the temperature profile. The results were obtained from a location in close proximity to an occupant.

In Case 1 (a case with 0% load removed by the PCBs), \( \Delta T_{ha_{seated}} \) was larger than 3 K, and \( \Delta T_{ha_{standing}} \) was larger than 4 K, both of which exceeded the temperature gradient limits set by ASHRAE (2013). While the PCBs were in operation, however, they significantly reduced the temperature gradient in the occupied zone, as shown in Cases 2 and 4. When the PCBs removed a small amount of heat in a room, they could reduce the temperature in the upper part of the room, as demonstrated by a comparison of Cases 1 and 2. In these two cases, the vertical temperature profiles in breathing zone were mainly dominated by the DV system, which was why their
temperature profiles almost overlapped in lower part. When the cooling capacity of the PCBs was further increased (e.g. Case 4), they could also significantly decrease the temperature in the lower part of the room.

Fig. 4.4 Vertical temperature profiles when different percentages of cooling load were removed by PCBs

![Vertical temperature profiles](image)

Fig. 4.5 compares the $\Delta T_{ha\_seated}$ and $\Delta T_{ha\_standing}$ under different $\eta$ and $ACH$. The head-to-ankle temperature difference was inversely correlated to $\eta$ and $ACH$. When the cooling load was high, the DV-Only system was not able to maintain a sufficiently small temperature gradient in the room, even when the air change rate was as large as $ACH = 7.9$. However, adding PCBs to the DV system could solve this problem. In a coupled DV-PCB system, the value of $\Delta T_{ha}$ results from combined effect of indoor space conditions, DV system parameters and PCB system parameters. Thus, a careful design should be implemented to ensure head-to-ankle temperature difference meets the thermal comfort requirement.
4.4.2 Draft in the coupled DV-PCB system

Although PCBs can reduce the room temperature gradient, the downward cold jet could increase the local percent dissatisfied. Fig. 4.6 compares the PD distributions for Case 5 (a DV-Only system) and Case 10 (a DV-PCB system). The DV-Only system had a low overall PD in the occupied zone. The added PCBs created a high-draft region beneath with a relatively low temperature, high air velocity, and PD > 15%. However, this region was observed only under the PCB, and the global PD remained the same.

Fig. 4.7 illustrates the change in PD with $\eta$ beneath the PCB for the three sets of PCBs. With the same cooling load, the larger the $\eta$ was, the higher the overall PD became. Although the standing and sitting heights differed by 0.6 m, the PD values at these two heights were comparable. Because
jet flow in the region was still developing, the air temperature along the jet core did not change greatly. The PD at ankle level was much lower than that at standing or sitting heights, since the jet decayed significantly when it reached ankle level.

Fig. 4.7 PD at standing, sitting and ankle heights under PCBs for different PCB configurations

Fig. 4.7 shows that when other parameters were the same, the PD under the PCBs increased when \( \eta \) increased. Furthermore, the PD under the PCBs was inversely correlated with the total cross-sectional area of the PCBs. For example, at \( \eta = 40\% \) and \( H = 1.1 \text{ m} \), PD was the highest in the “2B” case and lowest in the “3C” case. The “3C” case had the largest PCB cross-sectional area and the “2B” case the smallest. These results occurred mainly because the strength of the downward jet increased with \( Q_{PCB}/A_{PCB} \) (total cooling load removed by PCB over total PCB cross-sectional area).

From indoor design perspective, Fig. 4.6 and Fig. 4.7 indicate that seats are suggested to be located at some distance away from PCB so that occupants could avoid high-draft region. Moreover, since the development of high-draft region, when obstructed by a table, is likely to propagate along table top surface, it is also not recommended to place tables away from PCB. However, as the high-draft region was restricted to be beneath PCB, PD level at most part of the room was satisfactory. Hence, in the establishment of design guideline of DV-PCB systems (detailed in Chapter 6), this research focused more on vertical temperature gradient as the thermal comfort criterion.
4.4.3 Spatial distribution of PMV and PPD

PMV is a widely used scale that quantifies thermal comfort into 7 levels based on the energy balance of body (Coleman and Stern 1997). Fig. 4.8(a) demonstrates PMV distribution (at clothing level $I_{cl} = 0.8$) in a vertical cross section in Case 4, where a very large cooling capacity of PCBs was used. Results showed that while the overall PMV level in the room was around 0, right beneath PCB there existed a band where PMV was -1. It suggests that when most region in the room gives neutral predicted thermal sensation, occupants would feel slightly cool under PCB. Fig. 4.8(b) further plots the distribution of PPD, percentage predicted dissatisfied people (Coleman and Stern 1997) for the same case. In most of the room space, the PPD level was under 10%, but the PPD magnitude could be as large as 30% beneath PCB. Results indicated that the impact of PCB on the PMV and PPD distributions was mainly in the zone right beneath PCB.

![Fig. 4.8 PMV and PPD distributions in Case 4](image)

4.4 Air Quality Analysis

The following two indices were used to appraise the air quality in a room with DV-PCB system.

Normalized contaminant concentration

The normalized contaminant concentration $C^*$ is defined as (Fleming et al. 2007):

$$C^* = \frac{C - C_s}{C_e - C_s}$$  \hspace{1cm} (4.8)
where \( C, C_e \) and \( C_s \) are the contaminant concentrations at a particular location, at the exhaust, and at the supply, respectively. When the room is in a perfectly mixed condition, \( C^* \) is equal to 1.

**Mean age of air (MAA)**

MAA (\( \tau \)) represents the average time needed for air to travel from the inlet to a specific location in the room (Li et al. 2003). This \( \tau \) is not a pre-defined variable in ANSYS Fluent, but, according to its definition, it can be obtained by solving:

\[
\rho \frac{\partial \tau}{\partial t} + \rho \langle u_i \rangle \frac{\partial \tau}{\partial x_i} - \frac{\partial}{\partial x_i} \left[ \Gamma_{\tau,\text{eff}} \frac{\partial \tau}{\partial x_i} \right] = \rho
\]  

(4.9)

where the effective diffusion coefficient is given by:

\[
\Gamma_{\tau,\text{eff}} = \rho v + v_r/0.7
\]  

(4.10)

These indices both contain information about indoor air quality, but they evaluate air quality from different perspectives: \( C^* \) indicates the contamination level of room air when contaminant is released in the room, whereas \( \tau \) is a direct measure of the freshness of room air.

Figs. 4.9(a) and 4.9(b) depict the transport of gaseous contaminant in Case 5 (DV only) and Case 10 (DV-PCB) when the contaminant was released from a single source. In Case 5, the contaminant ascended with the thermal plume, which resulted in contaminant stratification. When PCBs were used, the downward jet recirculated the contaminant downward to the lower part of the room. The airflow pathlines in Figs. 4.9(a) and (b) also also illustrate how PCB changed airflow pattern in the room. The normalized contaminant concentrations in Figs 4.9(c) and (d) very clearly show the impact of the PCBs on contaminant distribution. In the breathing zone, Case 5 exhibited much higher air quality than did Case 10. Actually, the contaminant concentration in the breathing zone in Case 10 was close to the perfectly mixed level (or the concentration at the exhaust).
Fig. 4.9 Contaminant distributions in Case 5 (a DV-Only case) and Case 10 (a DV-PCB case), where the contaminant concentration $c^*$ was normalized by the exhaust concentration in (c) and (d).

The mean age of air (MAA) is another parameter that can be used to evaluate air quality. Fig. 4.10 shows the MAA distribution on a vertical cross section for the above two cases. Case 5 exhibited a significantly smaller MAA than Case 10, since much less air recirculation existed in Case 5. Under the same air change rate, the average MAAs at breathing height (1.1 m) in Case 5 and Case 10 were 351 s and 456 s, respectively. The corresponding MAA for perfect-mixing ventilation at this air change rate was 501 s. Therefore, the PCB greatly increased the MAA in the room by enhancing indoor air mixing. As a result, the high air change efficiency of DV was negatively influenced by PCB. In the design of a DV-PCB system, it is important that sufficient fresh air is provided so that ventilation requirement could be met.

![Cross-sectional views of Case 5](image1)

(c) Cross-sectional views of Case 5

![Cross-sectional views of Case 10](image2)

(d) Cross-sectional views of Case 10

**4.5 Conclusions**

This study used a CFD model for predicting the airflow distribution and contaminant concentrations in a room with DV-PCB system. The CFD model was validated by our measured data. PCBs were found to effectively reduce the temperature gradient, especially if the PCBs were used to remove
40% or more of the cooling load. Meanwhile, the PCB-induced cold jet produced a high-draft region (PD > 15%) beneath the PCBs. It was also found that PCBs caused an airborne contaminant near the ceiling to travel downward to the occupied zone, thus disrupting the contaminant stratification created by DV. A design guide needs to be established to make sure that thermal and ventilation needs can be met by the coupled system in design.
5. Performance Evaluation and Model Development for Thermal Comfort and Air Quality

This investigation used a validated CFD model to establish a database of 70 DV-PCB cases for evaluating thermal and ventilation performance of the system under various design conditions. The database was then used for developing a mathematical model to predict thermal and ventilation performances of DV-PCB coupled systems that could be used by a designer. This section detailed how these cases were designed, what parameters were used for evaluating thermal and ventilation performances, and how the mathematical model was developed.

5.1 Indoor Space Layouts and Ranges of Design Parameters

Fig. 5.1 shows the layouts of four indoor spaces studied in this research: a classroom, an office, an industrial workshop and a restaurant. The classroom can hold 24 students and one instructor, which is an average classroom size in the US (Pianta et al. 2007; NCES 2018). The office layout consists of regular cooling loads such as computers and lights and has a seating capacity of eight people. The industrial workshop has a large floor area with high-power equipment uniformly placed on the tables. In the restaurant layout, a combination of round, square and long tables as well as partitions are included to simulate a typical dining space. In all cases, PCBs were evenly distributed near ceiling so that they can uniformly cool the rooms.
Fig. 5.1 Layouts of four indoor spaces studied in this research

(Legend: green – diffusers; blue – occupants; brown – exhausts; red – PC or other electrical devices; white – lights; grey – tables or closets)

It is critical to make sure the design parameters used in these cases are within reasonable ranges. This research conducted a literature survey to determine such ranges. Table 5.1 lists the height ranges of these indoor spaces, which were determined from New York State DOH (2009), US GSA (2018) and Yuan et al. (1999). Also listed in Table 5.1 are the ranges of cooling loads. This study split cooling loads into three categories. $Q_{oz}$ represents cooling loads from occupied zone, such as occupants and equipment in occupied zone, $Q_i$ stands for cooling loads from overhead lighting that is beyond the occupied zone, and $Q_{ex}$ represents exterior cooling loads that include transmitted solar radiation and heat from exterior surfaces and can enter both the occupied and unoccupied zones. Among these loads, ranges of $Q_i$ were determined from lighting levels recommended for different indoor spaces, based on NOAO (2015), while $Q_{oz}$ and $Q_{ex}$ were estimated according to ASHRAE (2010) and ASHRAE (2009). Air change rate in an indoor space could vary, but it needs to be larger than $\left( R_a \times A + R_p \times N \right)$ to meet the minimum outdoor air requirement (ASHRAE 2016), where $R_a$, $R_p$, $A$ and $N$ represents outdoor air rate per unit floor area, outdoor air rate per occupant, floor area and number of occupants, respectively. Previous studies (Fredriksson and Sandberg 2009) showed that PCB could remove as large as 80% of cooling load, so this research used 80% as the upper bound for $\eta$ in these cases. The details of each case are presented in Section 5.3.1.

Table 5.1 Room dimensions and ranges of parameters used for the four indoor spaces

<table>
<thead>
<tr>
<th></th>
<th>Office</th>
<th>Classroom</th>
<th>Workshop</th>
<th>Restaurant</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Floor surface area, A (m²)</td>
<td>31</td>
<td>101</td>
<td>320</td>
<td>410</td>
</tr>
<tr>
<td>---------------------------</td>
<td>-----</td>
<td>------</td>
<td>------</td>
<td>------</td>
</tr>
<tr>
<td>Height, H (m)</td>
<td>2.4 - 3.5</td>
<td>2.7 - 3.8</td>
<td>3 - 5</td>
<td>2.7 - 3.5</td>
</tr>
<tr>
<td>Loads (W/m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Occupied zone, Q_{oz}</td>
<td>5 - 50</td>
<td>5 - 25</td>
<td>10 - 35</td>
<td>5 - 25</td>
</tr>
<tr>
<td>Lighting, Q_{l}</td>
<td>0 - 12.5</td>
<td>0 - 10</td>
<td>0 - 15</td>
<td>0 - 10</td>
</tr>
<tr>
<td>Exterior, Q_{ex}</td>
<td>0 - 20</td>
<td>0 - 25</td>
<td>0 - 25</td>
<td>0 - 25</td>
</tr>
<tr>
<td>n (ACH)</td>
<td>2 - 13</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Percentage of load removed by PCB, η</td>
<td></td>
<td>0 - 80%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

5.2 Mathematical Models for $\Delta T_{ha}$ and $VE$

One objective of the current research was developing mathematical models for determining $\Delta T_{ha}$ ($= T_{1.1m} - T_{0.1m}$) and $VE(=\frac{C_p - C_s}{C_{1.1m} - C_s})$, which are important criteria for engineers in designing the DV-PCB system. Previous studies showed that in a DV system, $\Delta T_{ha}$ and $VE$ are both correlated with $Q_{oz}, Q_{l}, Q_{ex}, n,$ and $H$. In the coupled system with PCB, $\Delta T_{ha}$ and $VE$ should be expressed as:

$$\Delta T_{ha} = f\left(Q_{oz}, Q_{l}, Q_{ex}, n, H, A, \eta, \ldots\right)$$  \hspace{1cm} (5.1)

and

$$VE = g\left(Q_{oz}, Q_{l}, Q_{ex}, n, H, A, \eta, \ldots\right)$$  \hspace{1cm} (5.2)

This investigation used the database to determine the functions in Eqs. (5.1) and (5.2).

5.3 Results

This section first reported the specifications of the 70 cases in the database. With the results of these cases, this investigation examined the impact of key parameters on $\Delta T_{ha}$ and $VE$ in a DV-PCB system and developed mathematical models to correlate $\Delta T_{ha}$ and $VE$ with various design parameters. Strategies were also implemented to avoid condensation on PCB cooling coil surface.
Based on the mathematical models developed, this research developed a guide as well as a user interface for designing a DV-PCB system.

5.3.1 Case specifications and typical results

Table 5.2 lists detailed specifications of the 70 cases studied in this research, which includes 19 cases for offices, 17 cases for classrooms, 17 cases for workshops and 17 cases for restaurants. For each layout, various combinations of design parameters were used. These parameters were changed one by one among these cases, which enabled the sensitivity study of the parameters on $\Delta T_{ha}$ and $VE$. Parameters changed are marked in bold numbers in Table 5.2.

Table 5.2 Specifications of the 70 cases in the database

<table>
<thead>
<tr>
<th>Case #</th>
<th>H (m)</th>
<th>n (ACH)</th>
<th>$Q_{w}/A$ (W/m²)</th>
<th>$Q_{i}/A$ (W/m²)</th>
<th>$Q_{ev}/A$ (W/m²)</th>
<th>$Q/A$ (W/m²)</th>
<th>$\eta$ (%)</th>
<th>h (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OF.1</td>
<td>2.75</td>
<td>5</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.2</td>
<td>3.35</td>
<td>5</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.3</td>
<td>3.05</td>
<td>3</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.4</td>
<td>3.05</td>
<td>5</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.5</td>
<td>3.05</td>
<td>7</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.6</td>
<td>3.05</td>
<td>5</td>
<td>12.2</td>
<td>6.1</td>
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<td>18.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.7</td>
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<td>5</td>
<td>24.0</td>
<td>6.1</td>
<td>0</td>
<td>30.0</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.8</td>
<td>3.05</td>
<td>5</td>
<td>43.6</td>
<td>8.1</td>
<td>10.0</td>
<td>61.7</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.9</td>
<td>3.05</td>
<td>5</td>
<td>49.1</td>
<td>12.1</td>
<td>10.0</td>
<td>71.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.10</td>
<td>3.05</td>
<td>5</td>
<td>49.1</td>
<td>12.1</td>
<td>30.0</td>
<td>91.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
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<td>OF.11</td>
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<td>5</td>
<td>39.4</td>
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<td>10.0</td>
<td>57.5</td>
<td>0%</td>
<td>2.51</td>
</tr>
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<td>OF.12</td>
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<td>10.0</td>
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<td>5%</td>
<td>2.51</td>
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<td>OF.13</td>
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<td>10.0</td>
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<td>10%</td>
<td>2.51</td>
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<td>OF.14</td>
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<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>20%</td>
<td>2.51</td>
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<td>OF.15</td>
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<td>30%</td>
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<td>60%</td>
<td>2.51</td>
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<td>OF.17</td>
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<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>80%</td>
<td>2.51</td>
</tr>
<tr>
<td>OF.18</td>
<td>3.05</td>
<td>5</td>
<td>39.4</td>
<td>8.1</td>
<td>10.0</td>
<td>57.5</td>
<td>40%</td>
<td>2.31</td>
</tr>
</tbody>
</table>
### (b) Classrooms

<table>
<thead>
<tr>
<th>Case #</th>
<th>H (m)</th>
<th>n (ACH)</th>
<th>$Q_{ew}/A$ (W/m²)</th>
<th>$Q_l/A$ (W/m²)</th>
<th>$Q_{ex}/A$ (W/m²)</th>
<th>$Q_t/A$ (W/m²)</th>
<th>$\eta$ (-)</th>
<th>h (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CL.1</td>
<td>3.00</td>
<td>4</td>
<td>15.1</td>
<td>8.1</td>
<td>5.0</td>
<td>28.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>CL.2</td>
<td>3.30</td>
<td>4</td>
<td>15.1</td>
<td>8.1</td>
<td>5.0</td>
<td>28.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>CL.3</td>
<td>3.60</td>
<td>4</td>
<td>15.1</td>
<td>8.1</td>
<td>5.0</td>
<td>28.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
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<td>CL.4</td>
<td>3.30</td>
<td>3</td>
<td>15.1</td>
<td>8.1</td>
<td>5.0</td>
<td>28.2</td>
<td>40%</td>
<td>2.51</td>
</tr>
<tr>
<td>CL.5</td>
<td>3.30</td>
<td>6</td>
<td>15.1</td>
<td>8.1</td>
<td>5.0</td>
<td>28.2</td>
<td>40%</td>
<td>2.51</td>
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### (c) Workshops

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(d) Restaurants
Fig. 5.2 illustrates $\Delta T_{ha}$ and $VE$ results predicted by CFD model for several typical cases: OF.3, OF.4, OF.5, OF.11 and OF.17. These cases had different $\eta$ and $ACH$ but had the same values for other parameters. Fig. 5.2 (a) shows that at the same $ACH$, $\Delta T_{ha}$ decreases with $\eta$, which verified that PCB could reduce temperature gradient developed by DV. Meanwhile, at the same $\eta$, $\Delta T_{ha}$ decreased with $ACH$. This was because increasing airflow rate reduced overall temperature difference ($T_v - T_s$) in room and hence reduced $\Delta T_{ha}$ too.

Fig. 5.2 (b) further illustrates $VE$ profiles predicted by CFD for these cases. Since DV supplied clean air from lower part of room and exhausted air from ceiling, $VE$ was generally higher in occupied zone than in upper zone (mixing region). Besides, since PCB-induced jet generated by PCB recirculated contaminant downward, a higher $\eta$ led to a lower $VE$. On the other hand, when $\eta$ remained same, $VE$ increased with $ACH$, because increasing supply airflow rate could increase indoor airborne contaminant removal rate. Hence, in a room with DV-PCB system, $VE$ is determined by the combined effects of airflow supplied by DV system and downward jet induced by PCB.

![Fig. 5.2 $\Delta T_{ha}$ and $VE$ results predicted by CFD model for several cases](image-url)
The design parameters and boundary conditions described in Table 5.2 and the calculated $\Delta T_{ha}$ and $VE$ results formed a database, which were used for model development in Section 3.2.

5.3.2 Development of $\Delta T_{ha}$ and $VE$ models

To quantify thermal and ventilation performances of a DV-PCB system and to guide its design, it is needed to correlate $\Delta T_{ha}$ and $VE$ with various parameters using mathematical models that can be used directly in design process. Section 3.1 showed that $\Delta T_{ha}$ and $VE$ were dependent on $\eta$ and $ACH$. In addition, previous studies showed that $\Delta T_{ha}$ was also related to the distribution of cooling loads in the room. Hence, this study assumed $\Delta T_{ha}$ to be correlated with various parameters by:

$$\Delta T_{ha} = f_1(Q_{oz}, Q_{ex}, Q_{l}, h, \eta)f_2(Q_{l}, ACH, H, A, \eta)$$  \hspace{1cm} (5.3)

In Eq. (5.3), $f_2$ represents the temperature difference between air exhaust and supply in the room:

$$f_2(Q_{l}, ACH, H, A, \eta) = T_e - T_s = \frac{(1-\eta)Q_i}{\rho C_p \eta A}$$  \hspace{1cm} (5.4)

$f_1$ represents the ratio of temperature difference in occupied zone to the temperature difference between air near the floor and the ceiling in room. $f_1$ can be evaluated by cooling load distribution:

$$f_1(Q_{oz}, Q_{ex}, Q_{l}, Q_{l}, h, \eta) = \frac{\sum_{i=oz, ex, l} \alpha_i Q_i - \beta Q_{PCB}}{(1-\eta)Q_i}$$  \hspace{1cm} (5.5)

Coefficients $\alpha_i$ and $\beta$ were determined by empirical fitting using the results from the database: $\alpha_{oz} = 0.295, \alpha_l = 0.132, \alpha_{ex} = 0.185$, and $\beta = -0.203 h_0 / h$. Term $h_0 / h$ accounts for the influence of PCB height on $\beta$. When the height is infinitely large, the cooling effect of PCB on occupied zone is negligible which makes $\beta \approx 0$. A typical PCB height value, 2.6 m (Archtoolbox 2018), was used for $h_0$. Hence, the model for $\Delta T_{ha}$ was expressed as:

$$\Delta T_{ha} = \frac{0.295Q_{oz} + 0.132Q_{l} + 0.185Q_{ex} - 0.203 \left( \frac{2.6}{h} \right) Q_{PCB}}{\rho C_p \eta A}$$  \hspace{1cm} (5.6)

Eq. (5.6) indicates that different types of cooling loads contribute to $\Delta T_{ha}$ differently. Since $Q_{oz}$ represents cooling load in the occupied zone and impacts on $\Delta T_{ha}$ most directly, $\alpha_{oz}$ is the largest
among $\alpha_i$. $Q_i$ is the load outside of the occupied zone and has least impact on $\Delta T_{ha}$ so $\alpha_i$ is the smallest.

This study further developed a model for predicting ventilation effectiveness at breathing height ($H=1.1m$). Similarly, the coefficients in the mathematical expression was derived via best fit from the results in the database. The model is expressed as:

$$VE = \max \left\{ 2.83\left(1 - e^{-n/3}\right)Q_{oz} + 0.45Q_i + 0.63Q_{ex} - 0.79\left(2.6/h\right)\left(3.3/H\right)Q_{pCB},\, 1 \right\} \quad (5.7)$$

where the coefficients of $Q_{oz}$, $Q_i$ and $Q_{ex}$ were proportional to those in Eq. (5.6). Moreover, since air was supplied from DV diffusers at floor level and exhausted near ceiling, once contaminant concentration reaches mixing level at a certain height, it remains almost mixed at any height above. This could also be seen from Fig. 2. Hence, this characteristic was accounted for in Eq. (5.7).

Figs. 5.3 and 5.4 plots $\Delta T_{ha}$ and $VE$ predicted by CFD and those calculated by mathematical models, namely Eqs. (5.6) and (5.7). The coefficients in Eqs. (5.6) and (5.7) were from empirical fitting, which could partly explain the differences between CFD results and those calculated from the mathematical models. Besides, numerical errors in CFD simulations could also contribute to the discrepancies between the two sets of results. Nevertheless, the overall trend of $\Delta T_{ha}$ and $VE$ were predicted with reasonably good accuracy for these cases. Compared with the CFD method, calculation using the mathematical models requires much less modeling and computational efforts and could be easily used in design.
Fig. 5.3 Comparison of $\Delta T_{\text{int}}$ predicted by Eq. (9) and that by CFD

(C) Workshops

(d) Restaurants

Fig. 5.4 Comparison of $VE$ predicted by Eq. (10) and that by CFD

(a) Offices

(b) Classrooms

(c) Workshops

(d) Restaurants

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5.3.3 Determination of η

Eqs. (5.6) and (5.7) developed in Section 5.3.2 show that both $\Delta T_{ha}$ and $VE$ are correlated with ACH and $\eta$, so the two equations are coupled. However, plotting ACH - $\eta$ and VE - $\eta$ relationships on the same figure provides a straightforward way for determining optimal $\eta$ for a DV-PCB system.

As an example, Fig. 5.5 shows ACH - $\eta$ and VE - $\eta$ relationships generated from Eqs. (5.6) and (5.7) based on the following parameters in an indoor space: $Q_{oz}=58$ W/m$^2$, $Q_l=8$ W/m$^2$, $Q_{ex}=5$ W/m$^2$, workshop layout, $H=4.5$ m, $h=2.51$ m and desired $\Delta T_{ha}=3$ K. The two curves indicate that as $\eta$ increases, $\Delta T_{ha}$ decreases so the required ACH decreases. Meanwhile, the increase of $\eta$ leads to a decrease in $VE$ at breathing level. When a user selects a $\eta$, the required ACH and resultant $VE$ can be found directly. For instance, $\eta = 40\%$ in this case leads to $VE=1.2$ and required ACH=3.0.

![Fig. 5.5 ACH-$\eta$ and VE-$\eta$ curves for a given set of indoor space specifications](image)

This study recommends using $\eta$ which leads to $VE > 1.1$. Otherwise the resulting air quality would be close to that in mixing mode ventilation and DV-PCB system might not be economical. Therefore, Fig. 5 also includes a horizontal line $VE=1.1$, and its cross-section with VE-$\eta$ curve is marked as “critical point”. In the design, it is recommended that designer selects a $\eta$ that is smaller than $\eta_c$. 

Fig. 5.5 ACH-$\eta$ and VE-$\eta$ curves for a given set of indoor space specifications

(Q$_{oz}$=58 W/m$^2$, Q$_l$=8 W/m$^2$, Q$_{ex}$=5 W/m$^2$; Workshop layout; H=4.5m, h=2.51m)
After $\eta$ is selected for the DV-PCB system, the DV part and PCB part of the system could be designed separately.

### 5.3.4 Condensation control on PCB coil surface

Avoiding condensation on surface of PCB coils is another important objective in the design of a DV-PCB system. Hence, this study evaluated the humidity in an indoor space with this coupled system, and proposed design strategies to prevent condensation from occurring.

Fig. 5.6 shows two possible air systems that can be employed in DV-PCB: dedicated outdoor air system (DOAS) and air recirculation system. DOAS supplies 100% outdoor air to indoor space through DV, while air recirculation system mixes part of return air with outdoor air to save energy. Fig. 5.7 further shows four possible air handling processes on psychrometric chart for the two systems. When outdoor air (or mixed air) has relatively high humidity ($\omega > \omega_D$), it first needs to be dehumidified by cooling coil, as shown in processes 1 and 3. Hence, humidity of supply air becomes:

$$
\omega_l = \min[\omega_D, \omega_M]
= \min[\omega_D, x\omega_D + (1-x)\omega_E]
$$

(5.8)

where $x$ represents ratio of outdoor air in mixed air. If the air system is DOAS, $x=1.0$ and $\omega > \omega_D$. Hence, Eq. (5.8) applies to both the DOAS and air recirculation systems.

![Diagram of air handling systems](image)

(a) DOAS  (b) Air recirculation system

Fig. 5.6 Two possible air handling systems
Meanwhile, humidity at exhaust $\omega_E$ is calculated as:

$$m_{\text{air}} \omega_E = m_{\text{air}} \omega_I + S_{\text{vapor}} + m_{\text{inf}} \omega_O$$  \hspace{1cm}  (5.9)$$

where $m_{\text{air}}$ and $m_{\text{inf}}$ (<< $m_{\text{air}}$) represent mass flow rates of air from HVAC system and through infiltration respectively. $S_{\text{vapor}}$ stands for vapor generated inside indoor space, which is mainly from occupants. According to TenWholde and Pilon (2007), the average water vapor generation rate per person is 0.025 $g/s$.

With Eq. (5.8) and Eq. (5.9), $\omega_E$ could be obtained and used to obtain $T_{dp-E}$. To avoid condensation on PCB coil surface, it is critical to make sure PCB supply water temperature $T_{w,s} > T_{dp-E}$. After $T_{w,s}$ is determined, a user can find the required chilled water flow rate using data sheet provided in corresponding PCB product catalogue.

### 5.4 Determination of Required Chilled Water Flow Rate Using Mathematical Models

To determine the required chilled water flow rate ($q$) for PCB, one can refer to the corresponding product catalogue by using selected water supply temperature ($T_{w,s}$), as indicated in Section 5.3. In case a manufacturer does not provide an easy to find table, a simple mathematical model would be more desirable. ECS (2005) used a regression model to correlate $Q_{PCB}$ with the difference between room air temperature ($T_{\text{room}}$) and mean water temperature ($T_{w,m}$):

$$Q_{PCB} = k(T_{\text{room}} - T_{w,m})^n$$  \hspace{1cm}  (5.10)$$
However, the mean water temperature cannot be easily determined. Besides, this model was developed based on one piece of PCB and did not account for the variation of PCB size. In addition, Kim et al. (2018) found that chilled water flow rate \( q \) also affects \( Q_{PCB} \). Hence, this study correlates \( Q_{PCB} \) with PCB cross-sectional area \( (WL) \), difference between room air temperature, water supply temperature \( (T_{room} - T_{w,s}) \) and chilled water flow rate as follows:

\[
Q_{PCB} = k (WL)^m (T_{room} - T_{w,s})^n q^l
\]  

(5.11)

By conducting multi-variable regression using 320 performance data points provided by TROX (2017), this study obtained the following coefficients: \( m=1, n=1.07, \) and \( l=0.15 \) as well as

\[
k = 314.58 - 154.75W
\]  

(5.12)

The details of the multi-variable regression are provided in Appendix 2. Hence, Eq. (5.12) becomes:

\[
Q_{PCB} = (314.58 - 154.75W)(WL)(T_{room} - T_{w,s})^{1.07} q^{0.15}
\]  

(5.13)

In the market, the width of PCB is usually standardized by manufactures, but the length can be customized based on design needs. Therefore, it is more convenient to express Eq. (5.13) as the cooling capacity per unit length:

\[
\frac{Q_{PCB}}{L} = (314.58 - 154.75W)W(T_{room} - T_{w,s})^{1.07} q^{0.15}
\]  

(5.14)

Fig. 5.8 compares the calculated \( \frac{Q_{PCB}}{L} \) with those from TROX (2017). The agreement between the two results is good. Thus Eq. (5.14) can be used to determine required chilled water flow rate based on \( T_{w,s} \) and \( \frac{Q_{PCB}}{L} \). Fig. 5.9 further shows cooling capacity calculated by Eq. (5.14) for a PCB with \( W=0.61 \) m, at different \( (T_{room} - T_{w,s}) \). The curves clearly indicate that at the same \( q \) and \( T_{room} \), the cooling capacity significantly increases as \( T_{w,s} \) decreases. They also show that the cooling capacity grows with \( q \), although growth rate decreases as \( q \) becomes larger.
Note that coefficients $m$, $n$ and $l$ in Eq. (5.11) are sensitive to densities of coils and fins inside a PCB. The coefficients in Eq. (5.14) are based on the data from TROX (2017) and can be different for different PCB. With a different PCB, one can use the multi-variable regression method in Appendix 2 to find the corresponding coefficients.

5.5 Conclusions

This study established a database of 70 cases to investigate the thermal and ventilation performances of a DV-PCB system and developed a design guide for this system. $\Delta T_{ha}$ in a DV-PCB system was positively related to the cooling loads in the room and negatively related to $ACH$ and $\eta$. With the results from the database, a mathematical model was developed to predict $\Delta T_{ha}$ and yielded good prediction accuracy. The same database was used to develop a model that could predict reasonably accurate the ventilation effectiveness at breathing height in a DV-PCB system. At the same room height, ventilation effectiveness increases with $ACH$ but decreases with $\eta$. With developed $\Delta T_{ha}$ and VE models, $ACH$-$\eta$ and $VE$-$\eta$ curves can be generated to determine the maximum $\eta$ that can be used in a DV-PCB system.

This research also established a strategy to avoid condensation on PCB cooling coil surface by controlling PCB supply water temperature based on estimated indoor dewpoint temperature. A method was developed for obtaining the required chilled water flow rate based on PCB supply water temperature.
6. Design Guideline and Design Tool for a Coupled DV-PCB system

One important goal of this research project was to develop a practical guide for designing a coupled DV-PCB system. This chapter detailed the development of such a guide using the results obtained in the previous chapter. Moreover, since this guideline involved a lot of mathematical calculations, this study further developed a user-friendly design tool to automate the design process. The guideline and tool can give design engineers appropriate design parameters that ensures satisfying thermal comfort, good indoor quality while avoiding condensation risk on PCB cooling coil surface.

6.1 Establishment of Design Guideline

Based on the results obtained in previous sections, this investigation further proposed a step-by-step procedure for designing a DV-PCB system that provides satisfying thermal comfort and indoor air quality. This procedure also ensures that the risk of condensation on PCB cooling coil surface is avoided. Fig. 6.1 is a flowchart of the procedure.
The design process consists of the following five major steps.

1. Obtain basic characteristics and thermal conditions of the indoor space, for which the DV-PCB system is designed. These include:
   - Room dimensions: floor area and room height
   - Application of the indoor space: classroom, office, university lab, or auditorium etc.
   - Cooling loads in the room in terms of $Q_{oz}$, $Q_{ex}$ and $Q_f$.
   - Required $\Delta T_{ra}$ and temperature setpoint $T_{sp}$.
(2) Judge the applicability of the system.

As previously mentioned, some studies showed that a DV-Only system might have difficulty providing thermal comfort for an indoor space with cooling load larger than 40 W/m². But all these studies suggest thermal comfort could be easily achieved by DV-Only system if the corresponding cooling load is lower than 25 W/m². Therefore, this design guide first rates the applicability of DV-PCB system for an indoor space based on its total cooling load $Q_t$: “Not needed” for $Q_t < 25$ W/m², “Applicable” for $25$ W/m² $< Q_t < 40$ W/m² and “Recommended” for $Q_t > 40$ W/m².

It is also important that PCB can remove significant percentage of cooling load while still maintaining a satisfying VE (>1.1) at breathing height. Hence, a DV-PCB is considered applicable only if $\eta_c > 15\%$.

(3) Determine $\eta$ for the DV-PCB system.

The $ACH-\eta$ and $VE-\eta$ curves (shown in Fig. 5.5) are generated by using Eqs. (5.6) and (5.7). Designers can determine $\eta$ for the coupled system. It should be between 15% and $\eta_c$. The required $ACH$ and resulting $VE$ can then be obtained from the curves.

(4) Determine PCB design parameters.

The dewpoint temperature $T_{dp-E}$ in the indoor space is calculated by Eqs. (5.8) and (5.9). To avoid condensation, supply water temperature $T_{CWS}$ needs to be higher than $T_{dp-E}$. Designers can then find required chilled water flow rate using PCB product catalogue.

(5) Determine design parameters for DV system.

Based on $ACH$ and $VE$ calculated from Step (2), the required airflow rate $V_h$ to maintain thermal comfort is:

$$V_h = \frac{ACH \cdot H \cdot A}{3600} \quad (6.1)$$

The required air flow rate $V_f$ to meet ventilation requirement is:

$$V_f = \frac{R_p \cdot N + R_A \cdot A}{VE} \quad (6.2)$$

Hence, the design flow rate is:
\[ V = \max \{ V_h, V_f \} \]  \hspace{1cm} (6.3)

Meanwhile, the supply air temperature is:

\[ T_s = T_{sp} - \theta_f \frac{Q}{\rho C_p V} \]  \hspace{1cm} (6.4)

and \( \theta_f \) can be calculated by Mundt’s formula (1995).

With these design steps, a coupled DV-PCB system that provides good thermal comfort and air quality without condensation risk could be achieved. It should be noted that the DV-PCB system is only recommended for use in cooling condition. In heating seasons, if hot water is supplied to the beams, the heated air would move up and the DV-PCB system would not be energy effective. Hence, the DV-PCB system should operate as a DV-Only system in heating condition, and the resulting room air condition would be close to a mixed type. A perimeter heating device can also be added near exterior walls of an indoor space to provide extra heating.

**6.2 User-friendly Design Interface**

Although the five-step design guide provided a clear roadmap for designing a DV-PCB system, there were a lot of calculations involved in each step, which made the process time-consuming. To make the design more convenient for designers, this study further developed a design interface (*DV-PCB System Design Tool.xlsx*) to automate and visualize the design process. Once a designer enters required inputs, the design interface automatically updates ACH-\( \eta \) and VE-\( \eta \) curves, displays applicability of DV-PCB system and provides recommended design parameters. Below are some instructions for this design tool.

**Prerequisites**

In order to use the design tool, one should have the following two files saved in the same folder:

- *DV-PCB System Design Tool.xlsx*
- *PSYCH.XLA*

To make sure the design tool functions properly, the following steps are needed for a first-time user:

- After *DV-PCB System Design Tool.xlsx* is opened, if it shows “SECURITY WARNING: Macros have been disabled”, please click “Enable Content”.

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- Go to File \(\rightarrow\) Options\(\rightarrow\) Add-ins\(\rightarrow\) Manage Excel Add-ins\(\rightarrow\) Go, and browse for the file \textit{PSYCH.XLA}. Check “Psychrometric Functions Library” and click OK.
- In Data \(\rightarrow\) Queries & Connections\(\rightarrow\) Edit Links, please click “Change Source…” and browse for \textit{PSYCH.XLA} file. Then click “Open Source”.

Besides, the designer also needs to obtain the characteristics and thermal conditions of the indoor space, which the DV-PCB system is designed for.

**Design with the user interface**

Once the design tool is set up on computer properly and basic inputs of the indoor space are collected, one can start the designing with the tool \textit{DV-PCB System Design Tool.xlsm}. The design tool consists of five parts, which correspond to the five design steps. In all parts, red values are required inputs from designer, and black values are intermediate values calculated by the tool. Below are the explanations of the five parts.

- **PART I**
  
  In this part, a designer needs to enter outdoor design condition, HVAC air system type (DOAS, “air recirculation” or “heat recovery”) and basic thermal conditions for the indoor space. The designer should also enter temperature setpoint, and desired \(\Delta T_{hu}\) for the indoor space. Although ASHRAE (2017) recommends \(\Delta T_{hu}\) to be lower than 3 K, this tool allows users to customize required \(\Delta T_{hu}\) value in design. Once the required inputs are entered, the tool automatically generates and plots ACH – \(\eta\) and VE – \(\eta\) curves.
This tool makes sure that the coupled system can be used to remove at least 15% of cooling load while maintaining good air quality (VE>1.1). If VE is below 1.1 at η =15%, the system is considered non-applicable and a warning message would pop up:

![Warning message from design tool](image)

- **PART II**

  In this part, a designer can determine what percentage of load that he or she wants PCB to remove, by referring to curves generated in PART I. Once η value is selected, the corresponding VE and needed ACH are updated automatically in the tool.

- **PART III**
This part asks designers to select the width and length of each PCB, as well as number of pieces of PCBs that are to be used for the design. Once such data is entered, the tool calculates required sensible cooling capacity per meter, Φ, and plots the relationship between Φ and chilled water flow rate, q, at different chilled water supply temperatures. With these curves, a designer can choose a combination of chilled water supply temperature and flow rate that meet the need. It is critical that the design does not lead to condensation on PCB. Once designer selects a chilled water supply temperature, the tool shows if there is a risk of condensation in Box D64. If such a risk exists, the designer needs to select a higher chilled water supply water temperature.

- **PART IV**

Most results in this part are calculated automatically based on the inputs and intermediate calculations in prior parts. The only one item (Box D86) that the designer needs to provide is the face area of each piece of DV diffuser.

This part calculates and shows important DV system design parameters such as total airflow rate, ratio of fresh air, supply air temperature and number of diffusers needed.

---

**PART III: PCB DESIGN**

<table>
<thead>
<tr>
<th>PCB Selection</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Load to be removed by PCB (W)</td>
<td>1788.9</td>
</tr>
<tr>
<td>PCB width</td>
<td>20 inch</td>
</tr>
<tr>
<td>PCB length (m)</td>
<td>1.5</td>
</tr>
<tr>
<td>Number of pieces to be used</td>
<td>4</td>
</tr>
<tr>
<td>Required sensible cooling capacity per meter (W/m)</td>
<td>298.3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Chilled Water (CW) Temperature and Flow Rate per PCB</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>T_drain (°C)</td>
<td>9</td>
</tr>
<tr>
<td>T_con (°C)</td>
<td>15</td>
</tr>
<tr>
<td>Risk of Condensation</td>
<td>96</td>
</tr>
<tr>
<td>CW Flow Rate (mL/s)</td>
<td>125</td>
</tr>
</tbody>
</table>

**PART IV: DV SYSTEM DESIGN**

<table>
<thead>
<tr>
<th>Supply Airflow Rate</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Load to be removed by DV (W)</td>
<td>2684.9</td>
</tr>
<tr>
<td>Required V (m³/s)</td>
<td>0.327</td>
</tr>
<tr>
<td>Required V (m³/s)</td>
<td>0.145</td>
</tr>
<tr>
<td>Airflow Rate (m³/s)</td>
<td>0.327</td>
</tr>
<tr>
<td>Fresh air ratio</td>
<td>100.00%</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Supply Air Temperature</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>T_s (°C)</td>
<td>19.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Total DV Diffuser Area</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>A_total (m²)</td>
<td>1.6</td>
</tr>
<tr>
<td>Area of each diffuser A (m²)</td>
<td>0.4</td>
</tr>
<tr>
<td>Number of diffusers needed</td>
<td>5</td>
</tr>
</tbody>
</table>

---

*Fig. 6.4 Design Interface: Parts III and IV*
PART V

This section is a summary of the final design parameters for both DV part and PCB part. No extra input is needed, but it provides a list of key design parameters that can be used directly in installation or maintenance stage.

### PART V: YOUR FINAL DESIGN PARAMETERS

<table>
<thead>
<tr>
<th>PCB Parameters</th>
<th>or</th>
<th>0.51</th>
<th>(in m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width of PCB</td>
<td>20 inch</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Length of PCB</td>
<td>1.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of pieces to be used</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(T_{co}(^\circ C))</td>
<td>15.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CW Flow Rate (mL/s)</td>
<td>106</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>DV System Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Airflow Rate V (m³/s)</td>
</tr>
<tr>
<td>Fresh air ratio</td>
</tr>
<tr>
<td>(T_{co}(^\circ C))</td>
</tr>
<tr>
<td>Area of each diffuser (A (\text{m}^2))</td>
</tr>
<tr>
<td>Number of diffusers needed</td>
</tr>
</tbody>
</table>

Fig. 6.5 Design Interface: Part V

The “DesignTool” tab is the main tab that is to be used during the design. Other tabs, such as “DetailedData”, “PCBDesign” and “RoomType”, demonstrate a lot of default values or intermediate results, and only need to be accessed if the user believes the default values need to be changed.

This design tool is available in both SI and English unit versions.

### 6.3 Design Example

This section showed an example how to design a DV-PCB system for an industrial workshop with the following dimensions and indoor conditions:
The design steps are:

1. Calculate the cooling loads (in W/m²) in this indoor space:

\[ Q_{oc} = 42 W/m^2, \quad Q_i = 5 W/m^2 \quad \text{and} \quad Q_{ex} = 9 W/m^2 \]

   Enter the cooling load values, application of indoor space, temperature setpoint and required \( \Delta T_{ha} \) in PART I of the design tool.

2. Based on the generated curve, select a \( \eta \) which is to the left of critical point. For example, if \( \eta = 40\% \) is selected, the generated results are: \( VE = 1.13 \) and \( ACH = 3.03 \)

3. Determine PCB design parameters. Assume PCB width = 0.51 m (20 inch), and length = 1.5m and 4 pieces of PCBs are to be used. Based on the generated curves, the design can choose chilled water supply temperature of 13.4 °C and flow rate of 175 mL/s (per piece). The tool shows no risk of condensation based on these selections.

4. Determine DV system parameters.
   
   Most of the DV system related parameters are already determined automatically based on prior selections. Assuming each DV diffuser has an area of 0.6 m², it recommends using 10 DV diffusers in this design.

5. The final design parameters are summarized in PART V of the design tool on Table 6.1:
Table 6.1 Summarized final design parameters

<table>
<thead>
<tr>
<th>PCB Parameters</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Width of PCB</td>
<td>20 inch</td>
</tr>
<tr>
<td>Length of PCB (m)</td>
<td>1.5</td>
</tr>
<tr>
<td>Number of pieces to be used</td>
<td>10</td>
</tr>
<tr>
<td>$T_{CWS}$ ($^\circ$C)</td>
<td>13.4</td>
</tr>
<tr>
<td>CW Flow Rate (mL/s)</td>
<td>175</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>DV System Parameters</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Airflow Rate $V$ (m$^3$/s)</td>
<td>1.077</td>
</tr>
<tr>
<td>Fresh air ratio $r$</td>
<td>31.93%</td>
</tr>
<tr>
<td>$T_s$ ($^\circ$C)</td>
<td>18.5</td>
</tr>
<tr>
<td>Area of each diffuser $A_i$ (m$^2$)</td>
<td>0.6</td>
</tr>
<tr>
<td>Number of diffusers needed $N$</td>
<td>10</td>
</tr>
</tbody>
</table>

6.5 Conclusions

This study established a five-step design guideline for designing a coupled DV-PCB system. The coupled system designed by the guideline can provide acceptable thermal comfort and air quality without the risk of condensation on PCB coil. Furthermore, a design interface was developed to automate the design process. Based on the required inputs provided by designers, the design interface generates recommended design parameters automatically, which makes the design process user-friendly.
7. Energy Analysis of a DV-PCB system

In previous chapters, analysis was conducted to appraise a DV-PCB system from thermal and ventilation aspects. To obtain a more comprehensive evaluation of the system, it is also critical to study the coupled DV-PCB system from energy consumption. Hence, this chapter reported our efforts of evaluating the year-round energy consumption of a DV-PCB system in the US. This research further compared its energy consumption with a corresponding conventional DV-Only system. These results could be valuable for customers to evaluate the market value of a coupled DV-PCB system.

7.1 US Climate Zones

The Department of Energy (DOE) divided US into seven climate zones and named them as hot-humid, mixed-humid, cold, very cold, hot-dry, mixed-dry and marine.

![Fig. 7.1 US Climate zones determined by DOE (US DOE 2017)](image)

Since the climate varies significantly from one zone to another zone, the energy consumption is very different. This study selects the following five representative cities in various climate zones to perform energy analysis: Las Vegas, Chicago, Phoenix, Nashville and Miami. Since energy is related to weather data, Fig. 7.2 shows the hourly weather data (8760 data points for each city) distributions on the psychrometric chart for those cities.
Fig. 7.2 Hourly weather data distributions in various cities

7.2 Probability Matrix Method

This project employed probability matrix method to estimate the annual energy consumption of DV-PCB system. Fig. 7.3 shows that this method divided the psychrometric chart into nine sections and each section had the same air handling process. Table 7.1 further shows the probability of that weather data that would fall within each section.
Fig. 7.3 Division of psychrometric chart into nine sections

Table 7.1 Percentages of weather data that is within each section

<table>
<thead>
<tr>
<th>Section</th>
<th>Las Vegas</th>
<th>Chicago</th>
<th>Miami</th>
<th>Nashville</th>
<th>Phoenix</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.95%</td>
<td>18.37%</td>
<td>80.78%</td>
<td>34.09%</td>
<td>14.52%</td>
</tr>
<tr>
<td>2</td>
<td>0.66%</td>
<td>11.67%</td>
<td>6.72%</td>
<td>11.15%</td>
<td>1.34%</td>
</tr>
<tr>
<td>3</td>
<td>15.02%</td>
<td>0.46%</td>
<td>0.03%</td>
<td>0.54%</td>
<td>18.33%</td>
</tr>
<tr>
<td>4</td>
<td>12.10%</td>
<td>5.48%</td>
<td>4.97%</td>
<td>5.21%</td>
<td>17.03%</td>
</tr>
<tr>
<td>5</td>
<td>15.54%</td>
<td>27.99%</td>
<td>6.74%</td>
<td>24.12%</td>
<td>19.55%</td>
</tr>
<tr>
<td>6</td>
<td>7.63%</td>
<td>0.00%</td>
<td>0.00%</td>
<td>0.00%</td>
<td>3.08%</td>
</tr>
<tr>
<td>7</td>
<td>12.53%</td>
<td>0.10%</td>
<td>0.06%</td>
<td>0.59%</td>
<td>8.71%</td>
</tr>
<tr>
<td>8</td>
<td>31.56%</td>
<td>34.08%</td>
<td>0.71%</td>
<td>24.30%</td>
<td>17.43%</td>
</tr>
<tr>
<td>9</td>
<td>0.00%</td>
<td>1.86%</td>
<td>0.00%</td>
<td>0.00%</td>
<td>0.00%</td>
</tr>
</tbody>
</table>

In this study, the energy consumption from each HVAC component is determined using the following method:

\[ E_{\text{chiller}} = \sum_{i=1}^{9} P_i \cdot e_{i,\text{chiller}} \cdot N \]  
\[ E_{\text{fan}} = \sum_{i=1}^{9} P_i \cdot e_{i,\text{fan}} \cdot N \]  
\[ E_{\text{pump}} = \sum_{i=1}^{9} P_i \cdot e_{i,\text{pump}} \cdot N \]
\[ E_{\text{boiler}} = \sum_{i=1}^{9} P_i \cdot e_i,\text{boiler} \cdot N \] (7.4)

where \( P_i \) is the probability of the weather data being in section \( i \), \( e_i \) is the average energy consumption per hour in section \( i \) for different HVAC components (chiller, fan, pump or boiler) and \( N \) indicates the total number of hours for the investigation (\( N = 8760 \) in this case). According to Mierlo (1986), the difference between energy consumptions estimated using this probability matrix method and using hour-by-hour calculation method is within 5%. Therefore, this approach can be used to estimate the energy performance of DV-PCB system with good accuracy.

### 7.3 HVAC Systems and Air Handling Processes

Fig. 7.4 shows the HVAC system configuration of a DV-PCB system for the energy consumption evaluation. This system assumes that the air side and water side share one chiller. Besides, an economizer was used to save energy for this system. Since this study compares the energy performance of DV-PCB system with a conventional DV system, Fig. 7.5 also shows the system configuration of a corresponding DV system for comparison.

---

![Fig. 7.4 HVAC system configuration of DV-PCB](image)
This study used different air handling processes in different sections. Fig. 7.6 shows these processes for DV-Only system (and the DV part of DV-PCB system). The indoor temperature setpoint and relative humidity were controlled to be at 24°C and below 50%, respectively. The exhaust air temperature was assumed to be higher than the setpoint due to temperature stratification.
(6) Section 6

(7) Section 7
Fig. 7.6 Air handling processes for the nine sections in the psychrometric chart

(8) Section 8

(9) Section 9

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7.4 Comparison of Energy Consumptions Between DV-PCB System and DV-Only System

Using probability matrix method introduced in Section 7.2, this study calculated the annual energy consumptions when DV-PCB system was used in an indoor space with a floor area of 30 m² and cooling load of 60 W/m². PCB was used to remove 40% of the total cooling load. As a comparison, this research also calculated the corresponding energy consumption in the same indoor space but with a DV-Only system.

Fig. 7.7 compares the energy consumptions from chiller, fan and pump in the two systems. Although comparison shows that overall energy consumptions between these two systems are similar, the percentage of energy saved by DV-PCB system was found to be more significant (6.3% and 4.9%, respectively) in Las Vegas and Phoenix. Fig. 7.2 illustrates that the two cities have relatively larger portions of weather data located in Sections 3 and 6, where outdoor temperature is hot and dry. Fig. 7.8 further shows that on US map the two cities are located in hot-and-dry/mid-dry climate zones. This finding is also in accordance with the conclusions by Kim et al. (2019).
Fig. 7.7 Annual energy consumptions (in kWh) by DV-Only and DV-PCB systems of five investigated cities.

Note that Fig. 7.7 did not compare the energy consumptions from boilers. Since DV-PCB system is not believed to be energy efficient in heating seasons, this study assumed that the PCB part was not used for winter heating conditions and DV-PCB system thus became a DV-Only system. Hence, the results shown in this chapter reflects the energy saving potential of DV-PCB system in cooling seasons.

Fig. 7.8 Percentage of energy saved by DV-PCB system (as compared with DV-Only system) in multiple cities.

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7.5 Conclusions

This chapter reported annual energy consumption results of DV-PCB system in five cities in the US and compared them with those of DV-Only system. Overall, the energy consumption by DV-PCB system is similar as that by DV-Only system. However, the percentage of energy saving by DV-PCB system was more evident in cities in hot and dry climate regions, such as Phoenix and Las Vegas. Therefore, DV-PCB system shows higher market value when it is employed in regions with hot and dry outdoor climate.
8. Conclusions and Future Directions

8.1 Conclusions

The ASHRAE research project 1666 “Experimental Evaluation of the Thermal and Ventilation Performance of Stratified Air Distribution Systems Coupled with Passive Beams” led to the following conclusions:

(1) Existing studies focused on the DV-Only systems or PCB-only systems and investigated their performances from airflow, temperature distribution and energy consumption perspectives. There were few studies that examined the thermal and ventilation performance of coupled DV-PCB systems.

(2) This research constructed a DV-PCB system in a full-scale environmental chamber and measured the profiles of air velocity, temperature and contaminant concentration in multiple locations in the chamber for validation of the CFD model. A comparison of the simulated results and the measured data showed that the CFD model can predict indoor airflow and contaminant transport in the chamber with good accuracy.

(3) When the cooling load is high, a DV-Only system could create a high temperature gradient between head and ankle levels, which may cause discomfort. However, passive chilled beams were found to effectively reduce the temperature gradient, especially if the PCBs were used to remove 40% or more of the cooling load. The PCB-induced cold jet produced a high-draft region (PD > 15%) beneath the PCBs. The PD under the PCBs was positively correlated with the amount of heat removed by the PCBs, but inversely correlated with the total cross-sectional area of PCBs. The high-draft region was observed only under the PCBs, and the global PD level remained unaffected.

(4) The PCBs caused an airborne contaminant near the ceiling to travel downward to the occupied zone, thus disrupting the contaminant stratification created by DV. If the PCB-induced downward jet was strong enough, the contaminant concentration at breathing height could be similar to that with mixing ventilation. The PCBs also increased the mean age of air in the room.

(5) This study established a database of 70 cases to investigate the thermal and ventilation performances of a DV-PCB system and developed a design guide for this system. With the results from the database, mathematical models were developed to predict $\Delta T_{ha}$ and $\eta$. Then $ACH-\eta$ and $VE-\eta$ curves can be generated to determine the maximum $\eta$ that can be used in a DV-PCB system. Based on these curves and other results, this study developed a five-step procedure for designing a
A DV-PCB system designed based on this guide should achieve satisfying thermal comfort and air quality without condensation risk. A user-friendly design interface was further devised to automate the design process.

(6) This research compared the year-round energy consumptions of a DV-PCB system and conventional DV system at five representative cities in US. Results indicated that DV-PCB system shows better energy saving potential in dry and hot climate regions.

8.2 Future Directions

Although our results show that PCB can reduce the temperature gradient developed by DV so a DV-PCB system can remove large cooling load, PCB also recirculates part of airborne contaminant downwards to occupied zone which to some extent destroys air quality benefit of DV. Therefore, more efforts should be spent on seeking other strategies to remove high cooling loads in indoor spaces while maintaining contaminant stratification.

In addition, this study performed energy consumption analysis of DV-PCB system and demonstrated that the coupled system shows higher energy efficiency than a traditional DV system when it is used in hot and dry climate region. However, adding a PCB system also increases the first cost and operation cost of the HVAC system. Hence, it is worthwhile to perform a more comprehensive cost analysis of the DV-PCB system, such as its payback period in different climate zones. Such results can present a more complete market value of the coupled system.
References


Kim, J., Braun, J. E., & Tzempelikos, A. (2014). Energy savings potential of passive chilled beam system as a retrofit option for commercial buildings in different climates.


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Appendix 1: Comparison Between CFD Results and Experimental Data

1. Measurement Facility

Fig. A1 shows the schematic of the measurement system employed in this project. Three computers were used simultaneously during the measurements:

- **Computer 1**: Controlling the chilled water temperature and flow rate (via a building automation system) and controlling the air handling system. Fig. A2 further shows the LabView program interface for the air handling system control.
- **Computer 2**: Obtaining airflow velocity and air temperature data.
- **Computer 3**: Obtaining tracer gas concentration data via a gas analyzer.

All the computers were placed outside the chamber so that the system control and data acquisition were non-intrusive to the airflow inside test room.

![Fig. A1 Schematic of measurement system in IAQ chamber](image-url)
Fig.A2 Chamber control system

2. Dimensions of items in chamber and power of heated sources
Table A1. Dimensions of items inside chamber

<table>
<thead>
<tr>
<th>Item</th>
<th>Dimensions (m)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Δx</td>
<td>Δy</td>
</tr>
<tr>
<td>Chamber</td>
<td>5.15</td>
<td>6.08</td>
</tr>
<tr>
<td>Exhaust</td>
<td>0.44</td>
<td></td>
</tr>
<tr>
<td>Human dummy</td>
<td>0.41</td>
<td>0.41</td>
</tr>
<tr>
<td>PC</td>
<td>0.25</td>
<td>0.41</td>
</tr>
<tr>
<td>PCB A</td>
<td>0.58</td>
<td>2.52</td>
</tr>
<tr>
<td>PCB B</td>
<td>0.30</td>
<td>1.98</td>
</tr>
<tr>
<td>PCB C</td>
<td>0.43</td>
<td>2.94</td>
</tr>
<tr>
<td>Light</td>
<td>0.61</td>
<td>1.22</td>
</tr>
<tr>
<td>Table</td>
<td>1.83</td>
<td>0.61</td>
</tr>
<tr>
<td>Extra heated box</td>
<td>0.40</td>
<td>0.45</td>
</tr>
<tr>
<td>Diffuser</td>
<td>1.08</td>
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Table A2. Power of heat sources inside chamber

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<tr>
<td>Light</td>
<td>94 W each</td>
</tr>
<tr>
<td>Human dummy</td>
<td>~84 W each</td>
</tr>
<tr>
<td>PC</td>
<td>~109 W each</td>
</tr>
<tr>
<td>Extra heated box</td>
<td>220 W each</td>
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</table>

3. Measurement and Simulation Results
Case T1:

- Classroom layout, 2 pieces of type-A PCB, 30 W/m², η=40%
- 7 human dummies and 4 lights
- Tracer gas released from human dummy B

Fig. A3 Comparison of velocity profiles between measurement and simulation in Case T1
(U*=U/U_s, U_s=0.2m/s, H*=z/h, h=3.05 m. Line: simulation; Symbol: measurement.)
Fig. A4 Comparison of temperature profiles between measurement and simulation in Case T1

\[ \theta = \frac{(T - T_s)}{(T_e - T_s)}, \quad H' = \frac{z}{h}, \quad h = 3.05 \text{ m.} \]
Line: simulation; Symbol: measurement.

Fig. A5 Comparison of contaminant concentration profiles between measurement and simulation in Case T1

\[ C' = \frac{(C - C_s)}{(C_e - C_s)}, \quad H' = \frac{z}{h}, \quad h = 3.05 \text{ m.} \]
Line: simulation; Symbol: measurement.
Case T2:

- Office layout, 2 pieces of type-A PCB, 61 W/m², η=60%
- 8 human dummies, 8 PCs and 4 lights
- Tracer gas released from human dummy B

Fig. A6 Comparison of velocity profiles between measurement and simulation in Case T2

\( U^* = \frac{U}{U_s}, \quad U_s = 0.2 \text{m/s}, \quad H^* = z/h, \quad h = 3.05 \text{ m} \). Line: simulation; Symbol: measurement.
Fig. A7 Comparison of temperature profiles between measurement and simulation in Case T2
(\(\theta=(T-T_s)/(T_e-T_s)\), \(H^*=z/h\), \(h=3.05\) m. Line: simulation; Symbol: measurement.)

Fig. A8 Comparison of contaminant concentration profiles between measurement and simulation in Case T2
(\(C^*=(C-C_s)/(C_e-C_s)\), \(H^*=z/h\), \(h=3.05\) m. Line: simulation; Symbol: measurement.)
Case T3:

- Conference room layout, 2 pieces of type-A PCB, 90 W/m², \( \eta = 80\% \)
- 8 human dummies, 8 PCs, 4 extra heated box and 4 lights
- Tracer gas released from human dummy A

Fig. A9 Comparison of velocity profiles between measurement and simulation in Case T3

\( U^* = U/U_s, \ H^* = z/h, \ h = 3.05 \text{ m. Line: simulation; Symbol: measurement.} \)
Fig. A10 Comparison of temperature profiles between measurement and simulation in Case T3
\( \theta = (T - T_s)/(T_c - T_s), H' = z/h, h = 3.05 \text{ m.} \) Line: simulation; Symbol: measurement.

Fig. A11 Comparison of contaminant concentration profiles between measurement and simulation in Case T3
\( C' = (C - C_s)/(C_c - C_s), H' = z/h, h = 3.05 \text{ m.} \) Line: simulation; Symbol: measurement.
Case T4:

- Office layout, 2 pieces of type-B PCB, 30 W/m², η=80%
- 7 human dummies and 4 lights
- Tracer gas released from human dummy B

Fig. A12 Comparison of velocity profiles between measurement and simulation in Case T4
(U*=U/Us, U_s=0.2m/s, H*=z/h, h=3.05 m. Line: simulation; Symbol: measurement.)
Fig. A13 Comparison of temperature profiles between measurement and simulation in Case T4
\[ \theta = (T - T_s)/(T_e - T_s), \quad H^* = z/h, \quad h = 3.05 \text{ m}. \text{ Line: simulation; Symbol: measurement.} \]

Fig. A14 Comparison of contaminant concentration profiles between measurement and simulation in Case T4
\[ C^* = (C - C_s)/(C_e - C_s), \quad H^* = z/h, \quad h = 3.05 \text{ m}. \text{ Line: simulation; Symbol: measurement.} \]
Case T5:

- Conference room layout, 2 pieces of type-B PCB, 61 W/m², \( \eta = 40\% \)
- 8 human dummies, 8 PCs and 4 lights
- Tracer gas released from human dummy A

Fig. A15 Comparison of velocity profiles between measurement and simulation in Case T5

\( U^* = U/U_s, U_s = 0.2\text{m/s}, H^* = z/h, h = 3.05\text{ m}. \) Line: simulation; Symbol: measurement.
Fig. A16 Comparison of temperature profiles between measurement and simulation in Case T5
(θ=(T-T_s)/(T_e-T_s), H'=z/h, h=3.05 m. Line: simulation; Symbol: measurement.)

Fig. A17 Comparison of contaminant concentration profiles between measurement and simulation in Case T5
(C'=(C-C_s)/(C_e-C_s), H'=z/h, h=3.05 m. Line: simulation; Symbol: measurement.)
Case T6:

- Classroom layout, 2 pieces of type-B PCB, 90 W/m², $\eta=60\%$
- 8 human dummies, 8 PCs, 4 extra heated box and 4 lights
- Tracer gas released from human dummy B

Fig. A18 Comparison of velocity profiles between measurement and simulation in Case T6

$(U'=U/Us, H'=z/h, h=3.05 \text{ m. Line: simulation; Symbol: measurement.})$
Fig. A19 Comparison of temperature profiles between measurement and simulation in Case T6
\( \theta = (T - T_s)/(T_e - T_s), H' = z/h, h = 3.05 \, \text{m}. \) Line: simulation; Symbol: measurement.

Fig. A20 Comparison of contaminant concentration profiles between measurement and simulation in Case T6
\( C' = (C - C_s)/(C_e - C_s), H' = z/h, h = 3.05 \, \text{m}. \) Line: simulation; Symbol: measurement.
Case T7:

- Conference room layout, 3 pieces of type-A PCB, 30 W/m², 60%
- 7 human dummies and 4 lights
- Tracer gas released from human dummy A

Fig. A21 Comparison of velocity profiles between measurement and simulation in Case T7

\( U^* = U/U_s, U_s = 0.2 \text{m/s}, H^* = z/h, h = 3.05 \text{ m}. \) Line: simulation; Symbol: measurement.
Fig. A22 Comparison of temperature profiles between measurement and simulation in Case T7

( $\theta = \frac{T-T_s}{T_e-T_s}$, $H^* = \frac{z}{h}$, $h = 3.05$ m. Line: simulation; Symbol: measurement.)

Fig. A23 Comparison of contaminant concentration profiles between measurement and simulation in Case T7

( $C^* = \frac{C-C_s}{C_e-C_s}$, $H^* = \frac{z}{h}$, $h = 3.05$ m. Line: simulation; Symbol: measurement.)
Case T8:

- Classroom layout, 3 pieces of type-A PCB, 61 W/m², η=80%
- 8 human dummies, 8 PCs and 4 lights
- Tracer gas released from human dummy B

Fig. A24 Comparison of velocity profiles between measurement and simulation in Case T8

\((U^* = U/U_s, U_s = 0.2 \text{m/s}, H^* = z/h, h = 3.05 \text{ m}. \text{Line: simulation; Symbol: measurement.})\)
Fig. A25 Comparison of temperature profiles between measurement and simulation in Case E8
(\( \theta = (T-T_s)/(T_c-T_s) \), \( H^* = z/h \), \( h = 3.05 \) m. Line: simulation; Symbol: measurement.)

Fig. A26 Comparison of contaminant concentration profiles between measurement and simulation in Case T8
(\( C^* = (C-C_s)/(C_C-C_s) \), \( H^* = z/h \), \( h = 3.05 \) m. Line: simulation; Symbol: measurement.)
Case T9:

- Classroom layout, 3 pieces of type-C PCB, 90 W/m², η=40%
- 8 human dummies, 8 PCs, 4 extra heated box and 4 lights
- Tracer gas released from human dummy B

Fig. A27 Comparison of velocity profiles between measurement and simulation in Case T9
(U' = U/U₀, U₀=0.2 m/s, H'=z/h, h=3.05 m. Line: simulation; Symbol: measurement.)
Fig. A28 Comparison of temperature profiles between measurement and simulation in Case T9
\( \theta = (T-T_s)/(T_e-T_s), H^* = z/h, h=3.05 \text{ m. Line: simulation; Symbol: measurement.} \)

Fig. A29 Comparison of contaminant concentration profiles between measurement and simulation in Case T9
\( C^* = (C-C_s)/(C_e-C_s), H^* = z/h, h=3.05 \text{ m. Line: simulation; Symbol: measurement.} \)
Living Lab Cases

Fig. A30 Photo of living lab

Fig. A31 3D model of the living lab
Living Lab Case 1: A DV-Only Case

- Internal cooling load: 5 real occupants, 1 dummy, lights (600W), PCs, monitors
- External load: 200W through wall
- Total: 1800W
- PCB not in operation
- CO₂ released on top of dummy 3B (at 1.1 m) as tracer gas
- Presented results are from red dot location in left figure.

Fig. A32 Comparison of airflow velocity, air temperature contaminant concentration profiles between measurement and simulation in Living Lab Case 1 (\(U^* = U/U_s\), \(U_s=0.2\) m/s, \(\theta = (T-T_s)/(T_e-T_s)\), \(C^* = (C-C_s)/(C_e-C_s)\), \(H^* = z/h\), \(h=4.5\) m. Line: simulation; Symbol: measurement.)
Living Lab Case 2: A DV-PCB Case

- Internal cooling load: 1 real occupant, 8 dummies, 4 extra heat boxes (220W each), lights (600W), PCs, monitors
- Total heat generation: 3000W
- PCB was used to remove 60% of the total heat generation
- CO₂ released on top of dummies 3B and 4C (both at 1.1 m) as tracer gas
- Presented results are from red dot location (under a PCB) in left figure.

Fig. A33 Comparison of airflow velocity, air temperature contaminant concentration profiles between measurement and simulation in Living Lab Case 2 \( (U^* = U/U_s, \theta = (T-T_s)/(T_e-T_s), C^* = (C-C_s)/(C_e-C_s), H^* = z/h, h = 4.5 \text{ m}) \). Line: simulation; Symbol: measurement.
Appendix 2: MATLAB code for obtaining coefficients in Eq. (5.13)

```matlab
% Read data points
q1 = xlsread('PCB_Cooling_Data.xlsx', 1, 'B2:B81');
q2 = xlsread('PCB_Cooling_Data.xlsx', 1, 'B83:B162');
q3 = xlsread('PCB_Cooling_Data.xlsx', 1, 'B165:B244');
q4 = xlsread('PCB_Cooling_Data.xlsx', 1, 'B248:B327');

Delta_T1 = xlsread('PCB_Cooling_Data.xlsx', 1, 'C2:C81');
Delta_T2 = xlsread('PCB_Cooling_Data.xlsx', 1, 'C83:C162');
Delta_T3 = xlsread('PCB_Cooling_Data.xlsx', 1, 'C165:C244');
Delta_T4 = xlsread('PCB_Cooling_Data.xlsx', 1, 'C248:C327');

Q1 = xlsread('PCB_Cooling_Data.xlsx', 1, 'E2:E81');
Q2 = xlsread('PCB_Cooling_Data.xlsx', 1, 'E83:E162');
Q3 = xlsread('PCB_Cooling_Data.xlsx', 1, 'E165:E244');
Q4 = xlsread('PCB_Cooling_Data.xlsx', 1, 'E248:E327');

ln_q1 = log(q1);
ln_q2 = log(q2);
ln_q3 = log(q3);
ln_q4 = log(q4);

ln_Delta_T1 = log(Delta_T1);
ln_Delta_T2 = log(Delta_T2);
ln_Delta_T3 = log(Delta_T3);
ln_Delta_T4 = log(Delta_T4);

ln_Q1 = log(Q1);
ln_Q2 = log(Q2);
ln_Q3 = log(Q3);
ln_Q4 = log(Q4);

X1 = [ones(size(ln_q1)) ln_q1 ln_Delta_T1];
b1 = regress (ln_Q1, X1);

X2 = [ones(size(ln_q2)) ln_q2 ln_Delta_T2];
b2 = regress (ln_Q2, X2);
```
X3 = [ones(size(ln_q3)) ln_q3 ln_Delta_T3];
b3 = regress (ln_Q3, X3);

X4 = [ones(size(ln_q4)) ln_q4 ln_Delta_T4];
b4 = regress (ln_Q4, X4);

n = mean([b1(3) b2(3) b3(3) b4(3)]);
l = mean([b1(2) b2(2) b3(2) b4(2)]);

W = [0.6096; 0.508; 0.4064; 0.3048]
K = exp([mean(ln_Q1 - l*ln_q1 - n*ln_Delta_T1);
mean(ln_Q2 - l*ln_q2 - n*ln_Delta_T2);
mean(ln_Q3 - l*ln_q3 - n*ln_Delta_T3);
mean(ln_Q4 - l*ln_q4 - n*ln_Delta_T4)])
p = polyfit(W,K,1)
PART I: BASIC INPUTS

Outdoor Condition

- Outdoor Temperature (°C): 24.4
- Outdoor Relative Humidity: 60%
- HVAC Air System: 2048

Design Space Geometry

- Room Type: Classroom
- Room Size: 120
- Number of Spaces: 8

Space Cooling Loads (W/m²)

- Loads from exterior: 0.50
- Loads from occupants: 20
- Loads from lighting: 3.3
- Total load: 23.8

Air Temperatures Input:

- Required Tc (°C): 2
- Temperature Setpoint (°C): 18.5
- Relative Humidity: 3.5

PART II: DETERMINE η

Input Percentage

- Percentage to be removed by PCB: 80%
- Percentage to be removed by DV: 20%

PART III: PCB DESIGN

PCB Selection

- Load to be removed by PCB (W): 18784.8
- PCB width (m): 0.51
- Number of pieces to be used: 2
- Required sensible cooling capacity per meter (W/m): 296.5

Chilled Water (CW)

- Temperature and Flow Rate per PCB
  - Tcw (°C): 18.5
  - Flow Rate (m³/s): 0.144

PART IV: DV SYSTEM DESIGN

Supply Air Temperature

- Tc (°C): 19.5

Total DV Diffuser Area

- Area of each diffuser (m²): 0.0318
- Number of diffusers needed: 3

PART V: YOUR FINAL DESIGN PARAMETERS

PCB Parameters

- Width of PCB (m): 20
- Length of PCB (m): 1.5
- Number of pieces to be used: 4
- Cork Weight (m²): 125

DV System Parameters

- Air Flow Rate (m³/s): 0.218
- Fresh air ratio: 0.6
- Tc (°C): 19.5
- Area of each diffuser (m²): 0.0318
- Number of diffusers needed: 2

Instructions for first-time users:

- Download PSYCH.XLA from https://wcec.ucdavis.edu/resources/software-resource-applications/
- After DV-PCB System Design Tool.xlsm is opened, if it shows "SECURITY WARNING: Macros have been disabled", please click "Enable Content".
- Go to File -> Options -> Add-ins -> Manage Excel Add-ins -> Go, and browse for the file PSYCH.XLA. Check "Psychrometric Functions Library" and click OK.
- In Data -> Queries & Connections -> Edit Links, please click "Change Source..." and browse for PSYCH.XLA. Then click "Open Source".

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Design Space Geometry:

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<th>Area (m²)</th>
<th>Height (m)</th>
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<tbody>
<tr>
<td>99</td>
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Enter Cooling Loads (W/m²):

- Occupants and Equipment: 24
- Overhead lighting: 8.1
- Loads from exterior: 1
- Total loads: 43.1

Required Temperature

| Required ΔT in °C | 2 |

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<th>α1</th>
<th>α2</th>
<th>α3</th>
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ACH 10

Required ACH and calculated Ventilation Effectiveness

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Curves based on:
- Beam width = 24 inch (0.61m)

Curves based on:
- Beam width = 20 inch (0.51m)

Curves based on:
- Beam width = 16 inch (0.41m)

Curves based on:
- Beam width = 14 inch (0.36m)
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<tr>
<td>Computer Lab</td>
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<tr>
<td>Restaurant</td>
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HVAC System

DOAS

Heat recovery

Air recirculation

Exhaust → Fan

Outside Air

Cooling Coil

Heating Coil

Indoor Space

Exhaust → Fan

Outside Air

Cooling Coil

Heating Coil

Indoor Space

Exhaust → Fan

Outside Air

Cooling Coil

Heating Coil

Indoor Space