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SUMMARY

A novel ventilation system which introduces bench exhausts and radiant cooling panels is proposed. The benefits of using bench exhausts and radiant cooling panel system in removing heat from equipment intensive laboratory are quantitatively evaluated using CFD simulations. This study resulted in a recommendation for a ventilation strategy using a combination of ceiling exhausts, bench exhausts and ceiling radiant cooling panels that appears to give the best thermal condition and energy saving in the laboratory. The system offers the potential for application in new and existing research laboratories where a large portion of the space cooling load in a laboratory is a result of the heat produced by the research equipment on the bench.

INTRODUCTION

Laboratories are usually equipment intensive. The ventilation flow rate required to cool these laboratories is higher than in a less equipment intensive zone of the building. A large portion of the space cooling load in a laboratory is a result of the heat produced by the research equipment on the bench. If the heat can be captured at the source, its impact on space cooling load and the resulting HVAC requirements and cooling cost will be reduced.

Traditionally, HVAC systems are designed as All-Air Systems. These systems achieve the tasks of ventilating and cooling a building by convection only, which means that air is used to ventilate the buildings in order to maintain a high level of indoor air quality as well as provide thermal comfort in the buildings. An alternative is to separate the tasks of ventilation and thermal space conditioning through a combination of radiation and convection inside the building by using the forced air to fulfill the ventilation requirements and radiant cooling panels to provide most of the cooling. This is because radiant cooling can be more energy-efficient than air-based systems [1-2]. It requires less parasitic energy (pump and fan energy) to remove heat from a space. Since the walls are radiantly cooled, the air temperature can be warmer to achieve the same level of thermal comfort. The warmer air temperature results in lower energy loss to the outdoors. In addition, the radiant cooling system also reduces noise and drafts of air-based HVAC systems. The preferred installation of the radiant cooling panel is ceiling mounted, as this reduces air stratification and facilitates collection of condensation. To investigate the possibility
of further reduction in cooling cost, the combination of benchtop exhaust system and ceiling mounted radiant cooling panels needs to be considered. The purpose of this study is

- to assess the performances of bench exhaust system and ceiling mounted radiant cooling panels in achieving required thermal comfort with reduced ventilation flow rate.
- to evaluate the saving of annual cooling cost by using bench exhausts and cooling panel at reduced ventilation flow rate.

**METHODOLOGY**

Computational Fluid Dynamics (CFD) uses numerical procedure to solve the conservation equations, Equation (1), that govern the airflow and heat transfer in a space. It is a very powerful and efficient methodology to investigate temperature and flow field in a room where there are many parameters involved [3-4]. Therefore, CFD is employed in this study [5].

\[
\frac{\partial}{\partial t}( \rho \varphi ) + \text{div} ( \rho \vec{V} \varphi - \Gamma_{\varphi} \text{grad} \varphi ) = S_{\varphi} \tag{1}
\]

Where:
- \( \rho \) = density
- \( \vec{V} \) = velocity vector
- \( \varphi \) = dependent variable
- \( \Gamma_{\varphi} \) = exchange coefficient (laminar + turbulent)
- \( S_{\varphi} \) = source or sink term

The airflow in a ventilated laboratory is turbulent. In this study, the turbulence is simulated with the k-\( \varepsilon \) model [6-7] that represents the most appropriate choice because of its extensive use in other research applications, such as predicting mixing rate of a jet flow and modeling airflow in urban open space [8-9].

The Predicted Percentage Dissatisfied (PPD) introduced by Fanger and given in the ASHRAE guide [10] is a widely used index in assessing the thermal comfort [11]. This index can be estimated by equations that are based on an empirical investigation of how people react to environments. As each individual has a different perception of the climate produced in a building, any given climate is unlikely to be considered satisfactory by all. Therefore, 80% occupant satisfaction is considered good, or a PPD of less than 20% is good. The equations implemented in the study shown below are taken from Fanger’s equations for Predicted Mean Vote (PMV) and PPD as given in BS EN ISO 7730: 1995.

**Definitions**

\[
\text{PMV} = (0.303e^{-0.036M} + 0.028)(M - W) - 3.05 \times 10^{-3}[5733 - 6.99(M - W) - p_a] - 0.42[(M - W) - 58.15]1.7 \times 10^{-5}M(5867 - p_a) - 0.0014M(34 - t_a) - 3.96 \times 10^{-8}f_{cl}[t_{cl} + 273]^4 - (t_r + 273)^4]
+ f_{cl}h_c(t_{cl} - t_a)
\tag{2}
\]

where
- \( t_{cl} = 35.7 \times 0.028(M - W) - I_{cl}(3.96 \times 10^{-8}f_{cl}[t_{cl} + 273]^4 - (t_r + 273)^4] + f_{cl}h_c(t_{cl} - t_a) \)
- \( h_c = 2.38(t_{cl} - t_a)^{0.25} \) or \( h_c = 12.1V^{0.5} \) whichever is the greater
- \( f_{cl} = 1.00 + 1.29I_{cl} \) for \( I_{cl} \leq 0.078 \text{ m}^2\text{KW}^{-1} \) or \( f_{cl} = 1.05 + 0.645I_{cl} \) for \( I_{cl} > 0.078 \text{ m}^2\text{KW}^{-1} \)
PPD = 100 - 95e^{-n} \tag{3}
where \ n = 0.03353PMV^4 + 0.2179PMV^2

List of Symbols

PMV = Predicted Mean Vote  
PPD = Predicted Percentage Dissatisfied  
M = Metabolic rate (W/m^2 of the body area)  
W = External work (W/m^2 of the body area ( = 0 in most cases))  
Icl = thermal resistance of clothing (m^2KW^-1)  
fcl = Ratio of clothed surface area to nude surface area  
ta = Air temperature (^\circ C)  
tr = Mean radiant temperature (^\circ C)  
v = Air velocity relative to the body (ms^-1)  
pa = Partial water vapor pressure in Pa  
hc = Convective heat transfer coefficient (Wm^-2K)  
tcl = Clothing surface Temperature (^\circ C)

CASE DESCRIPTION

A generic laboratory with a conventional air distribution system, shown in Figure 1, was developed as the baseline laboratory model. The same laboratory space was then modeled with the bench exhaust ventilation scheme at different exhaust flow rates and cooling panels of different arrangements for more than 30 cases among which 16 cases were presented in this paper as listed in Table 1. The bench exhausts used in this study were continuous slots along the length of the benches, mounted beneath shelves of the bench. The cooling panels were flushed on the ceiling above the bench top and aisle in as shown in Figure 1. One set that was mounted above the bench top, called bench panels (light blue colored in Figure 1), included 3 panels. The central panel was 0.6m wide, and the two against the side walls were 0.3m wide. The other set, called aisle panels, was mounted above the two aisles. These cooling panels were maintained at 13.9°C temperature. The total heat generation from the bench devices was 5808W. The lighting heat sources were 2275W. The sensible heat from each occupant was assumed to be 80W. Solar loading from south-facing windows on the external wall was divided as 1160W transmitted into the room and 1273W absorbed by the glass and the external wall section. The supply temperature was 11.1°C for all cases.
Table 1. Case description and results

<table>
<thead>
<tr>
<th></th>
<th>Cooling panels</th>
<th>ACH</th>
<th>Bench exh. CFM</th>
<th>Ceiling exh. CFM</th>
<th>Average Alt T (°C)</th>
<th>Average PPD (%)</th>
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<tr>
<td>Baseline</td>
<td>-</td>
<td>13</td>
<td>0</td>
<td>-1750</td>
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<td>12.8</td>
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<td>24.0</td>
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<tr>
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<td>23.6</td>
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<tr>
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RESULTS

In order to evaluate the performance of bench exhausts and cooling panels, the occupied zone is divided into two, the walking zone and the bench zone, as highlighted in a) and b) of Figure 2, respectively. The walking zone covers the regions of aisles and the doorways from the floor to...
1.8m above, and the bench zone includes the volumes from the top of the bench to 1.8m from the floor.

Let’s first discuss the effects of bench exhaust on thermal comfort. The average PPD and temperature in the two occupied zones are summarized in Table 1. They are also graphically presented in Figures 3 to demonstrate the trends. Table 1 shows that the PPD in the occupied zone is 12.8% in the baseline case with 13 ACH. When using bench exhausts, the PPD in the walking zone drops to 7.6% in Case 1 even at reduced supply flow rate of 8 ACH. When further reducing the supply flow rate to 6 ACH (see Case 5), the average air temperature in occupied zones increases by 2.3°C, and PPD, especially in bench zone becomes significantly higher. This indicates that, when using bench exhausts, the supply flow rate can be reduced from 13 ACH to 8 ACH to achieve similar level of thermal comfort in the lab.
Secondly, we would like to evaluate the performance of cooling panels. Comparing the cases of same flow rate with and without cooling panel, Cases 1 through 5 for example, shows that the cooling panels reduce the average air temperature in the occupied zones by about 0.8°C to 1.4°C. When cooling panels are used, sensible heat is removed from the room by both ventilation and radiation. Through radiative heat transfer, the heat emitted by occupants is absorbed by the cooling panel. Therefore, even the average air temperature in the occupied zone is reduced by only 1°C, the thermal comfort can be improved significantly as seen in Table 1 or Figure 3. It is especially so when the supply flow rate is lower, for example Case 13 and Case 16. Table 1 shows that at 8 ACH, the PPD in the walking zone increases slightly if the two sets of cooling panel are used together. This is because the air temperature is already slightly lower than the desired temperature of 23.5°C (74.3°F), cooler temperature of surrounding surfaces can have negative impact on thermal comfort. With aisle panels, the air temperature is, in general, lower in both walking and bench zones than with bench panels since the total surface area of aisle panels is larger. The average PPD in the two occupied zones at 6 ACH all drop below 20% with any cooling panel arrangements. At 5 ACH, however, it requires both bench and aisle panels working together to bring PPD below 20% in occupied zones. At 4 ACH, even combination of the two sets of cooling panels cannot bring the PPD to lower than 20% unless more cooling panels are installed.

**OPERATING COST REDUCTION**

The ventilation flow rate required for equipment-intensive laboratories to be thermally comfortable can be as high as 13 ACH, referring to the baseline case. With the proposed bench exhausts, the ventilation rate can be reduced to 8 ACH (Case 1) to achieve similar level of thermal comfort. The combination of bench exhausts and ceiling mounted radiant cooling panels can further bring the ventilation flow rate down to 5 ACH as seen in Case 12. Figure 4 illustrates the annual cooling costs of these three cases estimated for a typical lab located in Washington DC area.

![Annual Cooling Cost](image)

*Figure 4 Saving in annual cooling cost of a lab*
Comparing 13 ACH and 8 ACH, both without cooling panels, the total saving in annual cooling operating cost by utilizing bench exhausts alone is around 29%. Comparing the cooling costs of 8 ACH without cooling panel and 5 ACH with cooling panels, the saving by using cooling panels alone is about 28%. The total saving for annual cooling season when using bench exhausts together with radiant cooling panels is around 49%. The following conditions/assumptions are used in this cost calculation.

- The outdoor condition is taken from weather data in Washington DC.
- The cooling season is considered to be 4489 hours annually.
- The percentage of outdoor air is 70% for 13 ACH case and 100% for other two cases.
- Supply air temperature is 11.1°C (52°F), and the desired room temperature is 23°C (73.5°F) which is used as return air temperature in the calculation.
- Cooling load per CFM is considered to be the difference in air enthalpy when entering and leaving the HVAC system. Perfect duct insulation is assumed.
- The ventilation flow rate (CFM) in Table 1 is the flow rate required during the peak load of the day. The average cooling load of a day is assumed to be 64.3% of the day’s peak load. So is the ventilation flow rate used in the cost calculation.
- The cost of electricity is 0.1$/KWH, fuel is 8.0$/100Btu; chilled water generation efficiency is 1.0KW/TON; fan efficiency is 68%.

DISCUSSION

The following conclusions are drawn from this study:

- For laboratories with high bench heat sources, using bench exhausts in conjunction with ceiling exhausts improves the thermal condition in the occupied zone at even relatively low total ventilation flow rates.
- Without cooling panels, the case of 8 ACH with 30% to ceiling exhausts and 70% to bench exhausts (Case 1) appears to give the best thermal condition in the laboratory.
- With bench and aisle radiant cooling panels, the ventilation flow rate can be reduced to 5 ACH while maintaining the PPD in occupied zone below 20%. The cost reduction of annual cooling associated with using bench exhausts and radiant cooling panels is about 49% for a typical lab located in Washington DC, USA.

REFERENCES