EXPERIMENTAL STUDY ON INDOOR THERMAL ENVIRONMENT AND VENTILATION PERFORMANCE OF FLOOR-SUPPLY DISPLACEMENT VENTILATION SYSTEM

Takashi AKIMOTO*, Tatsuo NOBE**, Shin-ichi TANABE*** and Ken-ichi KIMURA****

秋元孝之，野部達夫，田辺新一，木村建一

Results of laboratory measurements on the thermal performance of the floor-supply displacement ventilation system in comparison with a displacement ventilation system with sidewall mounted diffuser and a ceiling based distribution system are described. The experiments were performed in a controlled chamber configured to resemble a modern office space with modular workstation furniture and partitions. In addition to detailed measurements of temperature, air velocity, and tracer gas concentration, a skin temperature controlled thermal manikin was used to evaluate the non-uniformity of thermal environments produced by these systems. Thermal stratification was observed in both of the displacement ventilation systems which produced greater air temperature differences than the ceiling based system. Under the conditions of the appropriate supply air volume against indoor heat load the displacement ventilation systems could be operated to maintain acceptable thermal comfort and high ventilation efficiency in the occupant zone of the space, at the same time taking advantage of the temperature stratification to achieve energy savings in air conditioning.

keywords: ventilation efficiency, thermal comfort, displacement ventilation

1. INTRODUCTION

The displacement ventilation system, that realizes high ventilation efficiency, has been one of the main topics in the HVAC field in recent years. This system has been gaining popularity in Scandinavia not only for large spaces with high ceilings, such as auditoriums and assembly halls, but also for medium sized office buildings. In this system, supply air of a temperature several degrees below room temperature is introduced at very low velocity through air supply devices of large area near floor level and returned at ceiling level. The system can achieve considerably higher ventilation efficiency in comparison with what is theoretically possible with the complete mixing system1. Displacement ventilation is characterized by two distinct horizontal zones with different airflow patterns. The lower zone contains cooler and fresher supply air. Where the air moves upward above heat sources, ambient air is entrained into the air movement. The upper zone is relatively well mixed and contains warmer air and greater concentrations of space contaminants2. However, thermal stratification may cause cold discomfort for the legs and feet in conjunction with warm discomfort at the head level. Actually, some examples of serious local discomfort due to draft and vertical temperature difference were reported in Melikov and Nielsen3 and Melikov et al.4 Wyon and Sandberg5 used a thermal manikin to predict comfort due to displacement ventilation and proposed some equations to assess indoor environment.

The horizontal discharge from a low side wall position has been regarded as the most common air supply configuration of the displacement ventilation system. However, when we have to address the need for greater supply volume without increasing the risk of draft, this could not be considered the most suitable arrangement. Recent publications have described an office displacement ventilation system design, what we call the floor-supply displacement ventilation system, that supplies air through an air permeable carpet over a raised access floor6,7, thus significantly increasing the area of air supply.

The purpose of this study is to identify the indoor thermal environment and ventilation performance of a floor-supply displacement ventilation system in comparison with a displacement ventilation system with sidewall mounted diffuser and a ceiling based distribution system by experiment in a
simulated office space. In addition to detailed measurements of temperature, air velocity, and tracer gas concentration, a skin temperature controlled thermal manikin was used to evaluate the non-uniformity of thermal environments produced by these systems.

2. EXPERIMENTAL METHODS

2.1 Experimental Chamber

All experiments were performed in an office environmental test chamber of Ochanomizu University in June 1995. The chamber was designed to resemble a modern office space with modular workstation furniture and partitions as shown in Figure 1. The floor except for the circumferential area of 150-200 mm width was fully covered with the air permeable carpet over a raised access floor, producing a 300 mm high subfloor plenum. Figure 2 presents the concept of the floor-supply displacement ventilation system. The air permeable tile carpet was composed of carpet fiber, a meshed intermediate layer, and a foundation layer with a total thickness of 7 mm. The air flow resistance of the carpet was about 14 Pa when the face air velocity of supply air was 0.01 m/s (calculated using measured air volume through the carpet). The underfloor space was used as a pressurized plenum chamber for air supply, to one side of which a supply duct was connected to realize the floor-supply displacement ventilation system. For the displacement ventilation system with the sidewall mounted diffuser, an air supply unit was placed at the corner of the chamber connected to a supply duct. For the conventional ceiling based system, supply air was diffused through an inlet on the ceiling connected to a supply duct. For all systems, air is exhausted at the ceiling level through two return air grilles. Four electric heaters on each desk were provided to simulate heat load sources for experiments.

![Experimental chamber](image)

**Figure 1** Experimental chamber

![Floor-supply displacement ventilation system](image)

**Figure 2** Floor-supply displacement ventilation system

2.2 Measurement Methods

Measurement methods are presented in Table 1. Room air temperatures were measured at 0 m, 0.1 m, 0.3 m, 0.6 m, 1.1 m, 1.6 m, 2.0 m, 2.3 m, 2.5 m, and 2.6 m above floor level at six points (poles 1-6). Air velocities were measured at 0 m, 0.1 m, 0.6 m, 1.1 m, 1.8 m, and 2.4 m above floor level.
at four points (points A-D). Local mean age of air was observed at 0.1 m, 0.6 m, 1.2 m, 1.8 m, and 2.4 m above floor level at point D for the vertical distribution. At the same time, that was measured at 1.2 m above floor level at points A, B, C, and D to see the horizontal distribution. The skin temperature controlled thermal manikin was used to evaluate the non-uniform thermal environments. Local mean age of air was obtained using a tracer gas (SF₆) decay method. Two or three sets of multi-gas monitors and multi-point sampler/dosers (B&K, 1302, 1303) were used to measure the concentration of tracer gas. For each test, local mean age of air was calculated using measured concentration. Local Mean Age of Air (TN) for tracer decay method is defined here as follows.

\[ T_{p} = \int_{0}^{\infty} \frac{C_{p}(t)}{C_{p}(0)} \, dt \]

where \( C_{p}(t) \) = concentration of tracer gas at point p at time t \([m^3/m^3]\)

\( C_{p}(0) \) = initial concentration of tracer gas for tracer decay method \([m^3/m^3]\)

Table 1 Measurement methods

<table>
<thead>
<tr>
<th>Temperature - measured using thermocouples</th>
<th>Vertical distribution at 11 heights</th>
</tr>
</thead>
<tbody>
<tr>
<td>underfloor: -0.1 m, 0 m / indoor: 0 m, 0.1 m, 0.3 m, 0.6 m, 1.1 m, 1.6 m, 2.0 m, 2.3 m, 2.5 m, 2.6 m</td>
<td>at 6 points (pole 1-6) = 66 points</td>
</tr>
<tr>
<td>Air inlet</td>
<td>2 points</td>
</tr>
<tr>
<td>Air outlet</td>
<td>2 points</td>
</tr>
<tr>
<td>Wall surface</td>
<td>4 points</td>
</tr>
<tr>
<td>Ceiling plenum</td>
<td>2 points</td>
</tr>
<tr>
<td>Electric heater surface</td>
<td>4 points</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Air Velocity - measured using Indoor climate analyzer</th>
<th>Vertical distribution at 6 heights</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 m, 0.1 m, 0.6 m, 1.1 m, 1.8 m, 2.4 m at 4 points (point A-D) = 24 points</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tracer Gas - measured using Gas monitor</th>
<th>Vertical distribution at 5 heights</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1 m, 0.6 m, 1.2 m, 1.8 m, 2.4 m at point D</td>
<td></td>
</tr>
<tr>
<td>Horizontal distribution at 1 height</td>
<td></td>
</tr>
<tr>
<td>1.2 m at 4 points (point A-D)</td>
<td></td>
</tr>
</tbody>
</table>

Table 2 Experimental conditions

<table>
<thead>
<tr>
<th>Case</th>
<th>System</th>
<th>Supply air volume</th>
<th>Heat load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>F-1</td>
<td>Floor-supply displacement ventilation system</td>
<td>800 m³/h¹</td>
<td>1600 W (50 W/m²)</td>
</tr>
<tr>
<td>F-2</td>
<td>Floor-supply displacement ventilation system</td>
<td>800 m³/h²</td>
<td>800 W (25 W/m²)</td>
</tr>
<tr>
<td>F-3</td>
<td>Floor-supply displacement ventilation system</td>
<td>400 m³/h³</td>
<td>800 W (25 W/m²)</td>
</tr>
<tr>
<td>D-1</td>
<td>Displacement ventilation system with sidewall mounted diffuser</td>
<td>800 m³/h</td>
<td>1600 W (50 W/m²)</td>
</tr>
<tr>
<td>D-2</td>
<td>Displacement ventilation system with sidewall mounted diffuser</td>
<td>800 m³/h</td>
<td>800 W (25 W/m²)</td>
</tr>
<tr>
<td>D-3</td>
<td>Displacement ventilation system with sidewall mounted diffuser</td>
<td>400 m³/h</td>
<td>800 W (25 W/m²)</td>
</tr>
<tr>
<td>C-1</td>
<td>Ceiling based air distribution system</td>
<td>800 m³/h</td>
<td>1600 W (50 W/m²)</td>
</tr>
<tr>
<td>C-2</td>
<td>Ceiling based air distribution system</td>
<td>800 m³/h</td>
<td>800 W (25 W/m²)</td>
</tr>
<tr>
<td>C-3</td>
<td>Ceiling based air distribution system</td>
<td>400 m³/h</td>
<td>800 W (25 W/m²)</td>
</tr>
</tbody>
</table>

¹ 9.6 room air changes per hour  ᵇ 4.8 room air changes per hour
3. RESULTS
3.1 Temperature Distribution

Vertical air temperature distribution is presented in Figures 3, 4, and 5. Data shown in the figures are averaged values measured at poles 1, 2, 3, and 6. Because in Cases D-1, D-2, and D-3 of the displacement ventilation system with sidewall mounted diffuser, the data measured at poles 4 and 5 were pretty much influenced by the direct supply air flow from the air supply unit set quite close to these poles, which were not considered to be suitable for the occupant space. Thermal stratification was observed as greater air temperature differences between lower and higher parts of the room for both of the floor-supply displacement ventilation system (Cases F-1, F-2, and F-3) and the displacement ventilation system with sidewall mounted diffuser (Cases C-1, C-2, and C-3) in comparison to the ceiling based system (Cases C-1, C-2, and C-3). There were more or less one degree temperature differences between both sides of the air permeable tile carpet of the floor-supply displacement ventilation system.

For vertical air temperature differences in the occupied space, air temperature differences between the 0.1-m and 1.1-m levels or 1.6-m levels are presented in Figure 6. They are averaged values at poles 1, 2, 3, and 6, as described before. A large vertical temperature difference that may cause local thermal discomfort was observed in several cases for both of the displacement ventilation systems. The floor-supply displacement ventilation system showed greater temperature differences compared to that of the displacement ventilation system with sidewall mounted diffuser for 800 m³/h supply air volume with 25 W/m² heat load and 400 m³/h supply air volume with 25 W/m² heat load. ISO Standard 7730(1) recommends that the vertical air temperature difference should be less than 3°C between the 0.1-m and 1.1-m levels, and ASHRAE Standard 55-1992(2) specifies less than 3°C between the

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*Figure 3* Vertical air temperature distribution: 800 m³/h, 50 W/m²

*Figure 4* Vertical air temperature distribution: 800 m³/h, 25 W/m²

*Figure 5* Vertical air temperature distribution: 400 m³/h, 25 W/m²

*Figure 6* Air temperature differences in occupied space: deviation from 0.1 m-level temperature
0.1-m and 1.7-m levels. Among the displacement ventilation systems, only two cases of a large supply air volume with a small heat load (Cases F-2 and D-2) could meet the qualification of the ASHRAE standard. The limits of local discomfort recommended in the standards are based on conventional complete mixing systems. However, care must be taken to choose appropriate supply air volumes in applying the displacement ventilation systems to the actual space to prevent local discomfort.

In Table 3, measured heat load, supply air temperature, return air temperature, and heat balance are shown for each case.

<table>
<thead>
<tr>
<th>Case</th>
<th>Measured heat load</th>
<th>Supply air temperature</th>
<th>Return air temperature</th>
<th>Heat balance</th>
</tr>
</thead>
<tbody>
<tr>
<td>F-1</td>
<td>1538 W</td>
<td>21.5°C</td>
<td>27.0°C</td>
<td>1450 W</td>
</tr>
<tr>
<td>F-2</td>
<td>973 W</td>
<td>22.7°C</td>
<td>27.0°C</td>
<td>1140 W</td>
</tr>
<tr>
<td>F-3</td>
<td>865 W</td>
<td>21.0°C</td>
<td>26.7°C</td>
<td>750 W</td>
</tr>
<tr>
<td>D-1</td>
<td>1584 W</td>
<td>19.9°C</td>
<td>26.2°C</td>
<td>1900 W</td>
</tr>
<tr>
<td>D-2</td>
<td>765 W</td>
<td>23.3°C</td>
<td>26.0°C</td>
<td>740 W</td>
</tr>
<tr>
<td>D-3</td>
<td>755 W</td>
<td>18.6°C</td>
<td>25.8°C</td>
<td>950 W</td>
</tr>
<tr>
<td>C-1</td>
<td>1642 W</td>
<td>20.3°C</td>
<td>25.7°C</td>
<td>1430 W</td>
</tr>
<tr>
<td>C-2</td>
<td>764 W</td>
<td>(21.6°C)</td>
<td>24.5°C</td>
<td>(764 W)</td>
</tr>
<tr>
<td>C-3</td>
<td>753 W</td>
<td>17.2°C</td>
<td>24.5°C</td>
<td>960 W</td>
</tr>
</tbody>
</table>

* Estimated using supply air volume and measured heat load because of imperfect measurement of supply air temperature

3.2 Air Velocity Distribution

Vertical air velocity distribution is presented in Figure 7 for 800 m³/h supply air volume with 50 W/m² heat load and in Figure 8 for 400 m³/h supply air volume with 25 W/m² heat load respectively. Plotted points are the averaged values of four measurement points at the same vertical height. The floor-supply displacement ventilation system (Cases F-1 and F-3) produced an uniform low air velocity (less than 0.1 m/s) at each measurement height. In comparison to that, a rather high air velocity near the floor (0.1-m level) was observed for the displacement ventilation system with sidewall mounted diffuser (Cases D-1 and D-3). The ceiling based system (Cases C-1 and C-3), however, showed a non-uniform distribution and a much higher air velocity near the ceiling. It is expected that the floor-supply displacement ventilation system will meet the need for a greater air supply volume without increasing the risk of draft discomfort.

![Figure 7](image1.png) ![Figure 8](image2.png)

**Figure 7** Vertical air velocity distribution: 800 m³/h, 50 W/m²  
**Figure 8** Vertical air velocity distribution: 400 m³/h, 25 W/m²

3.3 Measurements with Thermal Manikin

Figure 9 presents the measured skin surface temperatures at 16 parts of the thermal manikin for 800 m³/h supply air volume with 50 W/m² heat load. The thermal manikin was sitting in one of workstations (point D) during the experiments. Plotted points are the averages of measured values for 10 minutes. It was observed that measured skin surface temperatures of both of the displacement ventilation systems were slightly lower except in the area of the 'pelvis' than those of the ceiling based system. However, this is not considered too low to cause local thermal discomfort.
3.4 Local Mean Age of Air

Vertical distribution of local mean age of air is presented in Figure 10 for 800 m$^3$/h supply air volume with 50 W/m$^2$ heat load and in Figure 11 for 400 m$^3$/h supply air volume with 25 W/m$^2$ heat load respectively. In most cases in both of the displacement ventilation systems, the effect of piston flow is evident in the lower part of the room (Cases F-1, D-1, and F-3). In the displacement ventilation system with sidewall mounted diffuser for 400 m$^3$/h supply air volume with 25 W/m$^2$ heat load (Case D-3), however, local mean age of air in the lower part of the room was relatively greater. It could be explained that there was salient vertical down-draft along the walls due to the effect of gravity in the cooled air near the surface for this particular case. This downward flow along the wall compensated the surplus air that was not extracted from the room through ceiling air outlets. Since the data were only measured at point D, further investigation will be required to clarify the exact reason of this observation. Horizontal distribution of local mean age of air at 1.2 m above floor level was quite uniform in each cases, incidentally. For the ceiling based system, there were a little vertical differences in local mean age of air (Cases C-1 and C-3).

3.5 Contaminant removal

Time series of tracer gas concentrations measured at three different heights (0.1 m, 1.2 m, and 2.4 m) at point D are presented in Figures 12 for 800 m$^3$/h supply air volume with 50 W/m$^2$ heat load. Tracer gas was dosed at the mouth of the thermal manikin sitting in one of the workstations (point D). The displacement ventilation systems hardly had the room air mixed and contaminants were exhausted to the upper space even from the results with SF$_6$ measurements. It is observed that tracer gas concentrations at 1.2-m level of the floor-supply displacement ventilation system (Case F-1) were quite low, almost less than five ppm. However, the other plots at 0.1 m and 2.4 m showed higher concentrations and were unstable. Presumably, this instability was considered due to the use of SF$_6$, known as 'cold' gas, whose density is greater than the air, mixing itself with the air below the dosing point. If there was no close thermal flow, SF$_6$ could have spread in all directions resulting in high or low local concentrations. The other kind of mixed gas which
Density and temperature are similar to the exhalation should be used to investigate the influence of the human respiration exactly in future experiments. For the displacement ventilation system with sidewall mounted diffuser (Case D-1), tracer gas concentrations at 1.2-m level were not so low as the floor-supply displacement ventilation system. It could be explained that relatively higher air velocity below 1.2-m level influenced SF6 not to go down to the floor with its temperature in this case. Also, possibly existing downward thermal flow near the measurement point, as mentioned before, might have caused higher concentration. The concentrations at three different heights of the ceiling based system (Case C-1) were relatively uniform and almost all of them fell within the range between 10 and 20 ppm.

4. DISCUSSION

In most of the environmental chambers, the entire floor is used for air supply to realize high ventilation performance with low air velocity. The floor-supply displacement ventilation system produces a suitable environment in a room with supply air through a perforated raised floor with air-permeable carpet in the same manner. When a high air volume is required to get rid of heavy loads, for example, the air supplied from a very large area does not cause high velocities above the floor, thus making it possible to reduce the risk of draft in the occupied zone. Besides, there is still the advantage of cooling energy savings from thermal stratification with high ventilation performance even under the conditions where air volume is high. Furthermore, air supply from the entire floor allows for improved flexibility, whereas the floor-supply system with independent diffusers requires the space immediately above the diffuser grille to be wide open.

It is considered that more or less one degree temperature difference between both sides of the air-permeable tile carpet of the floor-supply displacement ventilation system as observed in Figures 3, 4, and 5 are one of the advantages of this system, because this system could have a part of indoor heat load compensated not only by convection but also by conduction heat transfer. Furthermore, one could have the supply air temperature higher because of the radiation effect caused by the carpet surface temperature slightly lower than the air temperature of the occupied space. Fitzner\textsuperscript{13} reported that when a cooled ceiling is applied to the displacement ventilation system, the wall temperature becomes lower than the air temperature along the entire wall and the downward flow from the ceiling reaches the floor. It is considered that the floor-supply displacement ventilation system would make the wall temperature lower, as well. In this case, the correct position of a temperature sensor is important.

5. CONCLUSIONS
Measurements have been made on the performance of a floor-supply displacement ventilation system in comparison with a displacement ventilation system with sidewall mounted diffuser and a ceiling based distribution system to obtain temperature and air velocity distributions, and local mean age of air in an environmental chamber simulating an office space. The experiments using thermal manikin were also performed. The major conclusions are as follows:

1. Thermal stratification was observed as greater air temperature differences in both of the displacement ventilation systems than in the ceiling based system.
2. Only the cases of a large supply air volume (800 m³/h) with a small heat load (25 W/m²) with the displacement ventilation systems could meet the ASHRAE standard in terms of vertical temperature difference.
3. The floor-supply displacement ventilation system showed greater vertical air temperature differences than the displacement ventilation system with sidewall mounted diffuser except for 800 m³/h supply air volume with 50 W/m² heat load.
4. The floor-supply displacement ventilation system produced an uniform low air velocity at each measurement height. Compared to that, rather high air velocity near the floor was observed for the displacement ventilation system with sidewall mounted diffuser. The ceiling based system showed non-uniform distribution and its air velocity near the ceiling was much higher than the other systems.
5. Measured skin surface temperatures of the thermal manikin with both of the displacement ventilation systems were slightly lower except in the area of the ‘pelvis’ than in the case of the ceiling based system.
6. Local mean age of air of the floor-supply displacement ventilation system was lower than the other systems especially in the lower part of the room.
7. The floor-supply displacement ventilation system could be operated to share a part of indoor heat load not only by convection but also by conduction heat transfer through the carpet floor.

ACKNOWLEDGMENTS
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REFERENCES
和文要約
1. はじめに
高効率の換気を可能とする置換換気による空調方式は、空調設備を分野で注目される技術の1つである。この方式は、これまで北欧においてオーディトリィム等の高効率の空気清浄機を用いて、オフィスに広く適用されてきた。この方式の特徴は、温度成層を解消することであり、その一般的な方法は、壁面下部から水平に空気を吹出するという吹出置換換気方式である。この場合、足元部の低温と頭部高さの高気温による局所不快感を引き起こす可能性がある。これに対して著者らは、ドライタイプの危険性が少なく、かつ大風量に対応する方式として、通気的なカーペットとOAフロアを経気流路とするという全面吹出置換換気方式を提案し、一部報告した。

本研究では、オフィスを模倣した実験室でこの全面床吹出置換換気方式と天井吹出空調方式を比較して室内熱環境、換気性能を評価することを目的としている。

2. 実験方法
全ての実験は、1995年6月にお茶の水女子大学の環境試験室にて行った。この試験室はワークスペース、倉庫、パーキングを有し、実際のオフィス空間を模擬したデザインとなっている。OAフロアの上には、床全面に通気的なカーペットを敷設した。床下空間は、全面床吹出置換換気の給気チャンバーとして用いた。壁吹出置換換気の吹出ノズルは、試験室のコーナー部に設置した。また、天井吹出空調の給気は天井吹出口（1ヶ所）より行った。すべてのシステムにおいて、換気は天井換気口（2ヶ所）に行われる。熱負荷として机上の電気ヒーター台を利用した。空気温度は6点（ポール1・6）の床上高さ0.1m、0.3m、0.6m、1.1m、1.6m、2.0m、2.3m、2.5m、2.6mの位置で計測した。気流は4点（A・D）の床上高さ0.1m、0.3m、0.6m、1.1m、1.6mの位置で計測した。局所空気温度はD点の床上高さ0.1m、0.3m、0.6m、1.2m、1.6m、2.4mの位置で計測した。また、A、B、C点の床上高さ1.2mの位置で計測した。熱環境の不均一を評価するために皮膚温度制御型のサーマルマネキンを用いた。気流はトレーサガス（SF6）の濃度変化により導出した。室内への給気は一定風量で制御した。給気温度はD点の床上高さ1.0m、1.2mの位置で25℃になるように制御した。給気温度と熱負荷の組合せは、各実験システムにおいて「800m³/h、25W/m²」、「800m³/h、25W/m²」、「400m³/h、25W/m²」の条件とした。

3. 結果
全面床吹出置換換気と壁吹出置換換気を用いた場合には、天井吹出空調を用いた場合と比べて室内下部と上部との間に大きな温度差が生じる。温度成層が確認された。全面床吹出置換換気を用いた場合、通気性カーペットの両面1℃前後の温度差があった。「800m³/h、25W/m²」、「400m³/h、25W/m²」の条件において、全面床吹出置換換気を用いた方が壁吹出置換換気を用いた場合よりも約1℃高い温度差が生じていた。全面床吹出置換換気と壁吹出置換換気を用いた場合、「800m³/h、25W/m²」の条件下でのみASHRAE 55-1992の上下温度差の推奨値（床上部0.1m・1.7m間で3℃）の条件を満たした。

全面床吹出置換換気を用いた場合の向上図に各計測点における気温は0.1m/s以下の低風速であった。これと比較して、壁吹出置換換気を用いた場合には、床下部においてかなり風速が高くなっただけでなく、天井吹出空調を用いた場合は天井近傍の風速が他の高温の計測値と比べて非常に高くなり、不均一な上下分野となった。

実験中、サーマルマネキンを点Dの位置に設置した。全面床吹出置換換気と壁吹出置換換気を用いた場合、サーマルマネキンの胸以下の表面温度は、天井吹出空調を用いた場合に対し、やや低かった。全面床吹出置換換気と壁吹出置換換気を用いた場合、ほとんどの条件で屋部下部におけるアストロ・フローレクサに顯著であったが、「400m³/h、25W/m²」の条件下で壁吹出置換換気を用いた場合には屋部下部における局所空気温度が相対的に高くなった。これは壁面内の空気冷気による下部空気の温度が上昇したものと考えられるが、この現象を解明するためにはさらなる検討を要する。天井吹出空調を用いた場合は向上図にしたがって局所空気温度の差がわずかしかみられなかった。

トレーサガスは点Dに設置したサーマルマネキンの口元から吹出した。全面床吹出置換換気を用いた場合の床上1.2mの高温のトレーサガス濃度値はほぼ5ppm以下であり非常に低かった。しかし床上0.1mと2.4mの高温の高さより高い濃度を示し、変動した。おそらくこの条件はSF6がコールドガスであるためドースを小上りの空気と混合したためと考えられる。壁吹出置換換気を用いた場合には床上1.2mの高温のトレーサガス濃度はあまり低くはなかった。これはこの高さより下部の比较的高い風速や存在したであろう下部空気流れに影響を受けると考えられる。全面床吹出空調を用いた場合、上下3点のトレーサガス濃度値は10-20ppmとなり比較的均一であった。

4. ディスカッション
全面床吹出置換換気は床面からの吹出しを行うことにより、高熱負荷処理のために大風量を吹出す場合でも床上の風速を低く抑えることができる。このシステムを用いた場合の適性のカーペット両面の0℃前後の温度差は、床下熱負荷の一部を対流だけでなく熱伝導により処理していことは示している。

5. 結論
全面床吹出置換換気と壁吹出置換換気を用いた場合には、天井吹出空調を用いた場合と比べて室内下部と上部との間に大きな温度差がみられ、温度成層が確認された。全面床吹出置換換気と壁吹出置換換気を用いた場合、「800m³/h、25W/m²」の条件下でのみASHRAE 55-1992の上下温度差の推奨値の条件を満たした。「800m³/h、50W/m²」の条件以外では、全面床吹出置換換気を用いた場合の方が壁吹出置換換気を用いた場合よりも床下部の温度分布が大きかった。全面床吹出置換換気を用いた場合は床上方向の各計測点における気温は0.1m/s以下の低風速であった。これと比較して、壁吹出置換換気を用いた場合には、床下部においてかなり風速が高くなった。全面床吹出置換換気と壁吹出置換換気を用いた場合、サーマルマネキンの胸以下の表面温度は、天井吹出空調を用いた場合よりやや低かった。全面床吹出置換換気を用いた場合、他のシステムと比較して、特に床下部における局所空気温度が低くなった。

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