EXPERIMENTAL STUDY ON INDOOR THERMAL ENVIRONMENT OF FLOOR-SUPPLY DISPLACEMENT VENTILATION SYSTEM UNDER VARIOUS HEAT LOAD CONDITIONS

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1. INTRODUCTION

Energy conservation and comfort in life are regarded as contradictory to each other in many aspects of our modern society. In the field of air-conditioning, the problem with a large amount of energy consumption to obtain thermal comfort has been discussed for a long time. The displacement ventilation system was developed as one of the most promising systems to realize high ventilation efficiency to provide acceptable thermal comfort. However, the vertical temperature difference and draft remain a critical issue in Japan, because higher cooling load with the displacement ventilation system might be caused by hot and humid summer. The horizontal discharge from a low side wall position has been regarded as a common air supply configuration of the displacement ventilation system. When we have to address the need for a greater supply air volume without increasing the risk of draft, however, this could not be considered as a suitable arrangement. We have developed a unique office displacement ventilation system, what we call the floor-supply displacement ventilation system, that supplies air through an air permeable carpet over a raised access floor1,2,3, thus significantly increasing the area of air supply. On the other hand, when designing this system, we must consider not only the total amount but the vertical and horizontal position of indoor heat sources which might influence the thermal stratification. A comparison was made among different ventilation systems in Tanabe and Kimura with regard to ventilation efficiency and thermal comfort4.

The purpose of this study is to identify the indoor thermal environment yielded by the floor-supply displacement ventilation system in a controlled chamber simulating a typical office space where numerous heat-generating equipment such as personal computers and copying machines are scattered throughout the room under various conditions. The authors conducted a set of experiments altering the supply air volume, the amount of heat load, and the position of heat sources.

2. EXPERIMENTAL METHODS

2.1 Experimental Chamber

All experiments were performed in a large climatic chamber designed to resemble a single span of an office building as shown in Figure 1, during the summer season in 1996. The floor is completely covered with the air permeable carpet over a raised access floor, producing a 120 mm high subfloor plenum.
Figure 2 illustrates 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. Using the measured volume through the carpet, the air flow resistance of the carpet was calculated to be about 14 Pa when the face air velocity of supply air was 0.01 m/s. The underfloor space was used as a pressurized plenum chamber of the supply air fed from a supply duct connected to one end side as shown in Figure 1. Air is extracted through the outlets in the lighting fixtures into the ceiling plenum space used as a return chamber. This double-chambered structure allows for the conditioned compartment equipped with an artificial sunlight outside the window to simulate the summer and winter conditions for the perimeter zone.

![Diagram of experimental chamber](image)

**Figure 1** Experimental chamber

### 2.3 Measurement Methods

Three types of measurements were conducted under the steady-state conditions, utilizing an instrumental cart devised to measure seven thermal parameters in eight locations as shown in Table 1, thermocouples for temperatures at different points as shown in Table 2, and a skin temperature controlled thermal manikin3 (three locations). The thermal manikin was used to evaluate the non-uniform thermal environments. For the cart measurements, locations were changed for the conditions with scattered heat sources (1.0 m, 1.5 m, 3.0 m, 4.5 m, 6.0 m, 9.0 m, 12.0 m, 15.0 m from the window) and clustered heat sources (1.0 m, 1.5 m, 3.0 m, 4.5 m, 6.0 m, 7.0 m, 8.0 m, 9.0 m from the window). The thermal manikin was set on an office chair facing the window. See Figure 1 for measurement locations.

#### Table 1 Measured items and points of instrumental cart

<table>
<thead>
<tr>
<th>Item</th>
<th>Instrument</th>
<th>Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Temperature</td>
<td>Thermocouple</td>
<td>0.1, 0.6, 1.1, 1.6 m</td>
</tr>
<tr>
<td>Globe Temperature</td>
<td>Small Globe Thermometer</td>
<td>0.1, 0.6, 1.1, 1.6 m</td>
</tr>
<tr>
<td>Air Velocity</td>
<td>Indoor Climate Analyzer</td>
<td>0.1, 1.1, 1.6 m</td>
</tr>
<tr>
<td>Solar Radiation</td>
<td>Solar Meter</td>
<td>1.3 m</td>
</tr>
<tr>
<td>Humidity</td>
<td>Relative Humidity Sensor</td>
<td>0.6 m</td>
</tr>
<tr>
<td>Equivalent Temperature</td>
<td>Comfort Meter</td>
<td>0.6 m</td>
</tr>
<tr>
<td>Radiant Temperature</td>
<td>Indoor Climate Analyzer</td>
<td>1.1 m / 6 sides</td>
</tr>
</tbody>
</table>

#### Table 2 Measured points of temperature distribution

<table>
<thead>
<tr>
<th>Vertical Distribution</th>
<th>Height (0.0, 0.1, 0.6, 1.1, 1.7, 2.2, 2.7, 3.0 m) × 7 points</th>
</tr>
</thead>
<tbody>
<tr>
<td>Under Floor</td>
<td>42 points</td>
</tr>
<tr>
<td>Wall Surface</td>
<td>7 points</td>
</tr>
<tr>
<td>Window Surface</td>
<td>15 points</td>
</tr>
<tr>
<td>Room Air Outlet</td>
<td>9 points</td>
</tr>
<tr>
<td>Outer Chamber</td>
<td>6 points</td>
</tr>
<tr>
<td>Air-Conditioning Unit</td>
<td>7 points</td>
</tr>
</tbody>
</table>

#### 2.3 Experimental Conditions

Incandescent light bulbs covered with aluminum cylinders were used as heat sources (100W×10, 20), and were either scattered in two rows or clustered together in one location, 8.0 m apart from the window, as shown in Figure 1. Supply air volume was kept constant during the period of each experiment. Supply air temperature was controlled to maintain 25.5°C room air temperature at a representative point near the entrance, 1.1 m above the floor level. Experiments were conducted on 12 conditions, altering the supply air volume (1890 m³/h, 1350 m³/h, 810 m³/h), the amount of heat load (26W/m², 36W/m²), the height of heat sources (high, middle, low), and the arrangement of heat sources (scattered, clustered). In addition, two cases of typical summer and winter conditions were reproduced to investigate the thermal environment near the window. In a summer condition, the artificial sunlight was utilized to acquire 580 W/m² of radiation on the surface of an assembly of Venetian blind set inside the window and the air temperature in the conditioned compartment outside the window was controlled to be 32°C at that time. In a winter condition, the Venetian blinds were not used and the air temperature outside the window was...
controlled to be 0°C. Table 3 presents details of experimental conditions. The air temperature of all of the guarded spaces surrounding the test chamber was steadily controlled throughout the experiments to prevent undesirable heat exchange with the outside.

Table 3 Experimental conditions

<table>
<thead>
<tr>
<th>Legend</th>
<th>Heat Load</th>
<th>Height of Heat Source</th>
<th>Arrangement of Heat Source</th>
<th>Perimeter Load</th>
<th>Supply Air Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>26HS-7</td>
<td>26 W/m²</td>
<td>High (2.0 m)</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26MS-7</td>
<td>26 W/m²</td>
<td>Middle (1.0 m)</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26LS-7</td>
<td>26 W/m²</td>
<td>Low (0.2 m)</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26HC-7</td>
<td>26 W/m²</td>
<td>High</td>
<td>Clustered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26MC-7</td>
<td>26 W/m²</td>
<td>Middle</td>
<td>Clustered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26LC-7</td>
<td>26 W/m²</td>
<td>Low</td>
<td>Clustered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>26MS-5</td>
<td>26 W/m²</td>
<td>Middle</td>
<td>Scattered</td>
<td>-</td>
<td>1350 m³/h</td>
</tr>
<tr>
<td>26MS-3</td>
<td>26 W/m²</td>
<td>Middle</td>
<td>Scattered</td>
<td>-</td>
<td>810 m³/h</td>
</tr>
<tr>
<td>36HS-7</td>
<td>36 W/m²</td>
<td>High</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>36MS-7</td>
<td>36 W/m²</td>
<td>Middle</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>36LS-7</td>
<td>36 W/m²</td>
<td>Low</td>
<td>Scattered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>36MC-7</td>
<td>36 W/m²</td>
<td>Middle</td>
<td>Clustered</td>
<td>-</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>SMR</td>
<td>26 W/m²</td>
<td>Middle</td>
<td>Scattered</td>
<td>Artificial sunlight</td>
<td>1890 m³/h</td>
</tr>
<tr>
<td>WTR</td>
<td>26 W/m²</td>
<td>Middle</td>
<td>Scattered</td>
<td>Cold window</td>
<td>1890 m³/h</td>
</tr>
</tbody>
</table>

"1" 7 room air changes per hour "2" 5 room air changes per hour "3" 3 room air changes per hour

3. RESULTS

3.1 Heat Balance

Ceiling material of the experimental chamber was made of 15 mm thick rock wool board. The radiation from the floor could affect the thermal environment in the ceiling plenum, depending on the thermal performance and temperature of ceiling material. In this experiment, the temperature difference between the air temperature at the ceiling air outlet and in the ceiling plenum was quite small. For instance, the mean value of measured air temperatures at the ceiling outlet and in the ceiling plenum were 26.7°C and 26.8°C when the amount of heat load was 26 W/m². When the amount of heat load was 36 W/m², they were 27.0°C and 27.0°C respectively. Therefore we did not take into account the heat transmission through ceiling material in particular. Figure 3 illustrates the fact that extracted heat calculated from the temperature difference of supply and exhausted air agreed well with the given heat load. Also, the heat load was reduced down to 50-60% before the supply air entered the occupied space. This means that the supply air penetration through the floor panels caused the whole floor to cool down, which in turn gave rise to a considerable heat transmission between the occupied space and the underfloor plenum. One could have the supply air temperature raised higher making use of the advantage in this floor-supply system due to the heat transmission through the floor panels.

3.2 Temperature Distribution

Figure 4 presents the vertical air temperature profiles for 1890 m³/h supply air volume in the case of 36 W/m² of scattered heat load with the height of the heat sources altered. All of the measured values at seven locations are plotted in the figure, and the profiles are almost identical for each condition. This means the thermal stratification was formed in the experimental chamber uniformly in each case. A change in the temperature gradient was observed at the height of the heat sources. The air warmed at the height of heat sources was stratified in the upper part of the experimental chamber. This proved the fact that the lower the height of the heat sources, the greater the vertical air temperature difference in the occupied zone. This effect became more significant when the

![Figure 3 Heat balance of experiments](image-url)
heat load was increased. However, the vertical temperature difference between 0.1m and 1.1m was 2.1°C at the severest condition for the case with 36 W/m² of scattered heat load at low position and 1890 m³/h of supply air (36LS-7), while that of others were kept below 1.5°C. The floor cooling effect of this floor-supply system as mentioned earlier was found quite obvious.

The vertical temperature profiles with different supply air volumes are shown in Figure 5. The vertical temperature difference increased when the supply air volume was reduced. The system could not meet the heat discharge requirements at 810 m³/h, causing a temperature rise throughout the chamber. In designing this floor-supply system, it is important to select the appropriate air volume against the heat load condition. In other words, if an air handling unit could not make the supply air temperature lower than a certain degree because of its performance limit, one has to think about increasing the supply air volume first to compensate the heat load appropriately.

Figure 6 presents the non-dimensional vertical air temperature distribution to compare several test conditions with different supply/exhaust air temperatures. In this process, the room air temperature is given in non-dimensional form as the temperature difference between the room air and the supply air, (T-Ts), divided by the temperature difference between in the extract and in the supply (Te-Ts). Supply air temperature was defined as the air temperature at the air supply duct between an air handling unit and the underfloor plenum. Extract air temperature was defined as the air temperature at the ceiling air outlet. If shown in Figure 6 means the surface air temperature of the air permeable carpet. The equation of non-dimensional temperature is shown as follows.

\[
\text{(non-dimensional temperature)} = \frac{(T-T_s)}{(T_e-T_s)}
\]

where

\[
T : \text{room air temperature [°C]}
\]
\[
T_s : \text{supply air temperature [°C]}
\]
\[
T_e : \text{extract air temperature [°C]}
\]

For the conditions where heat sources were scattered at the height of 1.0 m or 2.0 m with 1890 m³/h supply air volume (26HS-7, 26HS-7, 36HS-7, 36MS-7), non-dimensional temperature at the surface of the air permeable carpet was about 0.5. In addition to that, it was found that all of these temperatures, at the floor surface, at the 1.1 m height, and extract air fell on the same straight line. Supply air temperature was lowered in accordance with a decrease of
supply air volume (26MS-5, 26MS-3). This caused non-dimensional temperature at the surface of the air permeable carpet to be relatively higher, but, temperature at the three heights also fell on the same straight line. When the height of the scattered heat sources was low (26LS-7, 36LS-7), a substantial temperature increase at the lower part of the experimental chamber could not make the temperatures at the three heights fall on the straight line.

The local vertical air temperature profiles were no longer identical when the heat sources were clustered together at one location, 8 m from the window. As shown in Figure 7, the air temperature arose from 0.3 to 0.5°C near the heat sources, but the temperature gradient was maintained to be nearly constant throughout the room. Also, the vertical temperature difference was kept below 1°C even at the location nearest to the heat sources. Supposedly, these results owe much to the strong tendency of air to spread horizontally. Scattered heat source case showed a slight increase of air temperature at the center of the experimental chamber, also. It is considered that the sensor located near the entrance, representative point for the supply air temperature control, could not exactly sense a slight increase in the air temperature at the center of the experimental chamber.

Vertical air temperature distribution near the window for summer and winter conditions are presented in Figure 8. When artificial sunlight was brought in to simulate a typical summer condition of August (SMR), the thermal stratification was destroyed near the window, even though the blinds were lowered. In a winter condition where the temperature of the exterior compartment outside the window was controlled at 0°C (WTR), the effect of cold draft was observed in the perimeter zone, but the thermal stratification at 0.6 m above the floor level was kept undisturbed. To maintain the thermally comfortable condition at the perimeter zone for this floor-supply system, an effective perimeter conditioning system should be used together as well as for other air-conditioning systems.

3.3 Thermal Comfort

Air velocities for all the conditions except the case of winter (WTR) were kept under 0.1 m/s, well below the standard of ISO-77309 and ASHRAE 55-9210. For the case of winter (WTR), the cold draft caused an increase of air velocity below the window up to 0.15 m/s. Figures 9, 10, and 11 present estimated PMV values for scattered heat source conditions. All the PMV values based on 1.1 met and 0.6 clo fell between 0 and 0.5 except for the cases of summer (SMR) and winter (WTR). Careful measurements were conducted to investigate occupancy satisfaction. In the present experiment, a skin temperature controlled thermal manikin was used for this purpose. A large number of data were collected for further analyses. A strong influence of radiation and ambient air temperature on the skin temperature of a thermal manikin can be confirmed in Figure 12.

4. DISCUSSION

In the previous study, the authors reported results of experiments comparing the floor-supply displacement ventilation system with a sidewall mounted diffuser9. The results showed that the floor-supply displacement ventilation system formed greater temperature differences than the displacement ventilation system with a 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 experimented in a relatively small experimental chamber (5.1 m x 6.3 m x 2.6 m). The floor-supply system actually has the advantage of the floor-cooling effect as shown in the present study, however, the size of floor area, the amount and locations of heat load must be deliberated by taken into account when designing the system. The normalized minimum temperature in the floor region TF close to the floor, and the temperature distribution are strongly dependent on the type of heat sources in the room and dependent on airflow rate and heat load in a displacement ventilated room10. In the present study, it is possible to say that ‘high’ or ‘middle’ height of heat sources give a temperature distribution with relatively low temperatures in the occupied zone corresponding to a high system effectiveness.

The ratio of radiation to convection is an important parameter for the temperature distribution. A high level of this ratio will shift the temperature distribution curves to the higher side because it will increase the amount of heat supplied to the floor9. In the present study, light bulbs covered with aluminum cylinders were used. The aluminum cylinders itself have relatively low emission rates, but, the vertical openings yielded locally high emission rate parts facing the ceiling and the floor of the experimental chamber. Most of heat sources in a realistic office situation, however, have more uniform and higher emission rates even if the total amount of heat load is the same. Furthermore the surface temperature of the heat sources could affect to change the temperature distribution.
Skistad\(^9\) mentioned that the air temperature measured close to the floor appears to be approximately half way between the supply air temperature and the extract air temperature using the displacement ventilation system with sidewall mounted diffuser for rooms with conventional height (2.5 m - 3.5 m) and normal heat loadings. In the present study, under the conditions where heat sources were scattered at a height of 1.0 m or 2.0 m with 1890 m\(^3\)/h supply air volume (26HS-7, 26MS-7, 36HS-7, 36MS-7), non-dimensional air temperature at the surface of the air permeable carpet was about 0.5. When heat sources were scattered at a height of 1.0 m with 1350 m\(^3\)/h supply air volume (26MS-5) made it about 0.6. These results followed Skistad's rule quite well. Non-dimensional temperature at the surface of the air permeable carpet was about 0.8 under the condition of '26MS-3' since the system could not meet the heat discharge requirement. The authors showed a non-dimensional minimum temperature of 0.2 for several heat sources\(^9\) where supply air temperature was defined as the air temperature in the perforated floor, not the air temperature at the duct between an air handling unit and the underfloor plenum. It is quite simple and useful to use a non-dimensional temperature for expressing the characteristics of air distributions. However, especially for this floor-supply system, one has to care how supply air temperature was defined to normalize temperature when comparing air distributions.

It is often assumed that the temperature gradient in the room is linear from floor to ceiling for the rooms with displacement ventilation system\(^9\). In the present study, however, when the height of the scattered heat sources were low (26LS-7, 36LS-7), a large temperature increase at the lower part of the experimental chamber could not make a linear temperature gradient. There must be a specific rule for the floor-supply displacement ventilation system different from the displacement ventilation system with a sidewall mounted diffuser. Further investigation would be necessary to clarify the relationship among the supply air conditions, the heat source conditions, and the profile of vertical air distribution.

5. CONCLUSIONS

Measurements were made on the indoor thermal environment of the floor-supply displacement ventilation system in a controlled chamber altering the supply air volume, the amount of heat load, and the position of heat sources. Experiments using thermal manikin were also performed. The major conclusions are summarized as follows:
1. Heat transmission between the occupied space and underfloor plenum due to the floor cooling effect of the floor-supply displacement ventilation system kept the temperature gradient substantially low in the occupied zone. 50-60 percent of the total heat load was treated by the heat transmission through the floor panels.

2. The lower the height of the heat sources, the greater the vertical air temperature difference in the occupied zone. An increase of heat load or a decrease of supply air volume also caused a greater vertical air temperature difference in the occupied zone. The vertical temperature difference between 0.1 m and 1.1 m was $2.1 \degree C$ at the severest condition for the case with 36 W/m² of scattered heat load and 1890 m³/h of supply air.

3. A slight air temperature rise of 0.3-0.5°C was observed near the clustered heat sources, but the temperature gradient was kept nearly constant below 1°C throughout the room.

4. The thermal stratification near the window was destroyed in a summer condition. An effect of cold draft was observed in a winter condition, but the thermal stratification above 0.6 m was kept undisturbed.

5. For the case of winter, the cold draft caused an increase of air velocity below the window up to 0.15 m/s. All the estimated PMV values for scattered heat source conditions based on 1.1 met and 0.6 clo fell between 0 and 0.5 except for the case of summer and winter.

6. A strong influence of radiation and ambient air temperature on the skin temperature of a thermal manikin was observed to be significant.

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REFERENCES


1. 室温に

全面床吹出置換空調システムは置換空気システムの一種である。換気効率が高く、熱的快適性と省エネルギー性の点から注目されているが、温度成層を形成するために上下温度差が問題となる場合がある。垂直温度分布には室内熱負荷全体だけでなく窓熱源位置も影響を与えると考えられる。本研究では、2A化により内遮熱負荷が局所の状態で配置されているオフィス空間全面床吹出置換空調方式を適用した場合の室内熱環境および人体の熱的快適性を実験により評価することを目的としている。

2. 実験

実験は1996年の夏季に4棟、幅6m、床面高さ2.7mのオフィス空間1スパン分を模擬した環境試験室にて行われた。内部チャンバーフローパラメータ・サイクル・多孔・フロントを数値であり、建物内部プランナシートを空調モニタ内部で実際に敷設し室内にほぼ均一に給気される。内部チャンバーからの排気は天井に配したダブルTバーの排気スリットを通じて行う。天井ベンチチャネル方式とする。また、窓の外側の光熱室は夏季および冬季の外気を再現するための空調システムおよび人工太陽が備わっている。室内熱環境の設定は14条件について1）移動測定カート、2）熱電対による温度分布、3）サーマルマネキンによる測定が行われた。移動測定カートを7項目の環境要素を同時に測定可能なカートで、負荷換算条件については窓面からの距離1.0、0.5、3.0、6.0、9.0、12.0、15.0、18.0m、負荷集中条件については1.0、0.5、3.0、6.0、7.0、8.0、9.0mの地点にて測定を行った。また、窓内上部温度分布は7地点、高さ3地点にて実験中ロガーにより記録された。サーマルマネキンによる皮膚温度の測定は窓面を正面として検座位にて3地点を行った。

3. 室内負荷

室内負荷は照明負荷（1000×18）とヒーターで暖めた電球を用いた室内負荷（1000×10、20）により25W/m²、35W/m²とした。また、熱源の高さ0.2、1.0、2.0、0.2m、配置（分散、集中）、給気気流（1900、1350、810m³/h）、エアロマネキン（無、有・夏季、冬季）を変化させることによって14条件について測定を行った。負荷の配置は、熱源配設時は模擬負荷を2列にそして屋上に配置し、窓面は窓面より5mの地点にて集中させた。夏季に想定した条件では外気温度を32℃、ガラス内側のブラインドが閉じた放熱気流を550m³/h、冬季は外気温度を0℃とした。給気気流は一定し、室内的代表点（出入り口側の壁面近く、高さ1.5m）で空気温度25.5℃となるように吹出し温度を自動制御した。

4. 実験

実験は35W/m²、25W/m²の2種類の内部負荷で行われたが、空調機の給気から窓面吹出までの空気温度差により算出された除去熱量はこの無負荷を通じ一致した。窓面に給気される空気で窓室内側床表面までに全処理熱量の50～80％が処理され、これは床全体が冷されて床下への熱流が発生したためと考えられる。負荷を分散させ、高さを変化させた場合、上下温度分布より屋外全体を通じてほど一様に温度成層が広がっていることが分かった。また、負荷熱量により温度勾配の不連続点が変化が見られた。これ

5. まとめ

本方式では床パネルによる床冷却効果により居住者における温度快適性が小さくなった。熱源の温度低下および屋上に空調機が最大の温度低下が見られたが温度分布が全室内を通してほぼ一定に保たれた。夏季条件では窓面近傍の床が冷え、室内の温度成層が乱れた。冬季条件では窓面近傍にコールドドライフールは確認されなかった。高さ1.5m以上の温度成層が保たれていた。サーマルマネキンの皮膚温度と周辺空気温度および放射の影響が強く見られた。

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