Thermal Comfort Performance of a suspended ceiling system with non-perforated tiles as diffuse ventilation air inlet

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Abstract
This paper reports on an experimental study of the ability of a suspended ceiling system solution with non-perforated tiles to remove excessive heat loads from a single-person room without causing thermal discomfort when functioning as a diffuse ceiling inlet. Thermal comfort performance was evaluated for four different heat load intensities (10, 30, 60 and 100 W/m²). For all four scenarios all criteria for indoor climate category A were fulfilled except the criterion regarding maximum mean air velocity where the scenario 60 and 100 W/m² only fulfilled class B and C, respectively. However, the draught rates for all scenarios did not exceed the criteria for class A. For all scenarios the room air flow pattern appeared to be a room-size vortex in longitudinal direction generated by the heat sources in the room. The air movement was upward close to the heat sources and downward to the floor at the wall opposite to the heat sources. This flow pattern became more pronounced with increasing heat loads. The results support findings from previous studies.

Keywords - Ventilation; Diffuse ceiling; Indoor thermal comfort

1. Introduction
Diffuse ceiling ventilation (DIFCV) is an air distribution system where fresh air is supplied to the room from a pressurized space above a suspended ceiling (plenum). Research suggests DIFCV as a promising candidate for fulfilling future energy and indoor climate requirements in certain room settings compared to conventional ventilation systems [1-10]. However, more evidence on the performance of this rather novel approach to ventilation of buildings for humans is desirable. The purpose of this paper is to report on a study of the ability of a suspended ceiling system solution with non-perforated tiles to remove excessive heat loads from a room without causing thermal discomfort when functioning as a diffuse ceiling inlet. The room air distribution and flow pattern was also analyzed. The study was
conducted as a full-scale experiment in a climate chamber equipped as a single-person office.

2. Method

2.1 Test chamber

The experiments were executed in a test facility located at Aarhus University, Denmark. Fig. 1 shows the experimental setup.

![Fig. 1. Illustration of test chamber and heat load configuration with dimensions (mm). The room height was 2.800 mm to the suspended ceiling. Only wall to the right is facing outside.](image)

The south-facing wall was external but additional insulation material was fitted before experiments were conducted. The remaining walls were made of plasterboard elements fitted with 70 mm insulation material and floor and roof consisted of concrete slabs. The acoustic ceiling was suspended approximately 1.2 m from the upper concrete deck resulting in a room height of 2.8 m. The applied acoustic ceiling system consisted of Ecophon FocusTM Dg-tiles with 600 x 600 x 20 mm dimensions made of glass wool tiles with a nominal density of 100 kg/m³. The tiles were fitted in the Ecophon Connect profile system (reversed T-profiles) which also includes a finishing strip between the ceiling and the wall. The ventilation of the test chamber was established using a fully controllable mechanical ventilation system. The supply duct inlet was located above the suspended ceiling at the northern wall parallel to the centerline of the test chamber. The exhaust was a 600 x 600 mm perforated plate which replaced one of the acoustic ceiling tiles. Three types of heat loads were applied. A thermal manikin (80 W) and a laptop (35 W) were used to simulate a working station for a single-person office. Furthermore, heating blankets with controllable effect outputs were used to maintain the overall heat balance of the test chamber while representing solar radiation on the floor deck.
2.2 Test scenarios

Four test scenarios with varying heat load intensity were investigated, see Table 1. The room air temperature ($T_a$) was kept constant at 25 °C in all scenarios corresponding to the expected indoor temperature in a (cooling) summer situation [11]. The temperature difference between supply air and air temperature in the occupied zone ($\Delta T$) was kept constant at 10 K and the air flow rate was varied to remove excess heat depending on the heat load intensity of the scenario. The reason for this setup is that a $\Delta T$ of 10 K is difficult to provide without creating thermal discomfort. However, diffuse ceiling ventilation is known for its ability to operate with high $\Delta T$ without causing thermal discomfort [12]. It is therefore interesting to keep $\Delta T$ constant at this critical level and then investigate the ability of the diffuse ceiling system to remove excess heat loads as a function of the air flow rate without causing thermal discomfort.

The mean value of all air temperature measurements conducted in the occupied zone was used as an expression of $T_a$. It was not possible to determine the supply air temperature of the room ($T_i$), i.e. the air penetrating the suspended ceiling system, due to the unknown variations in air flow and air temperatures across the ceiling surface. Instead $T_i$ was set to be the supply air temperature to the plenum.

Table 1. List of test conditions for the four scenarios.

<table>
<thead>
<tr>
<th>Test name</th>
<th>Heat load [W/m²]</th>
<th>$\Delta T=T_a-T_i$ [K]</th>
<th>Air flow rate [m³/h]</th>
<th>Air change rate [h⁻¹]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T.1</td>
<td>10 (150 W)</td>
<td>10</td>
<td>62</td>
<td>1.5</td>
</tr>
<tr>
<td>T.2</td>
<td>30 (450 W)</td>
<td>10</td>
<td>182</td>
<td>4.3</td>
</tr>
<tr>
<td>T.3</td>
<td>60 (900 W)</td>
<td>10</td>
<td>336</td>
<td>8.0</td>
</tr>
<tr>
<td>T.4</td>
<td>100 (1500 W)</td>
<td>10</td>
<td>620</td>
<td>14.8</td>
</tr>
</tbody>
</table>

2.3 Evaluation of thermal conditions

The quality of the thermal indoor environment was evaluated in accordance with the criteria of CEN 1752 category A [11] for operative temperature (24.5 ± 1.0 °C), maximum mean air velocity (0.18 m/s), draught rate (< 15 %), cool ceiling (< 14 °C), floor surface temperature (19<$T_{floor}$<29 °C) and vertical air temperature difference between 1.1 m and 0.1 m from floor (<2 °C).

2.4 Measurements

Supply and exhaust flow rates were balanced and controlled by two Trox 200 TVR VAV controllers placed in the supply and the outtake duct, respectively.
Measurements of vertical air velocity and air temperature profiles were conducted under steady-state conditions using Dantec 54T21 anemometers. Seven anemometers at different heights were located at five different locations in the test chamber (A-E in Fig. 2).

Temperature measurements were conducted using shielded thermocouples of Type K. Air temperatures were measured at three different locations (A, C and E in Fig. 2). The plenum air temperature, supply and exhaust air temperatures were also measured. Furthermore, the floor surface temperature was measured at point C (Fig. 2) together with a range of points on the other room surfaces to ensure stationary room conditions.

![Fig. 2. Placement of air velocity and temperature measurements. Left: Plan drawing. Right: Section showing moveable measuring columns.](image)

Additionally, measurements of the operative temperature were conducted using a Dantec 54T38 comfort probe to investigate the operative temperature at the working station (E in Fig. 3). The measurements were conducted at a height of 1.1 m above ground representing a sitting person. Finally, measurements of draught rates were conducted using combined air velocity and temperature measurements. This was done at three locations (A, C and E in Fig. 2) in heights above ground representing ankle and neck for both a sitting and a standing person (0.1 m, 1.1 m and 1.7 m).

### 3. Results

#### 3.1 Thermal comfort

The mean operative temperature for all four scenarios (Table 2) was stable and within the comfort range (24.5 ± 1.0 °C). The difference in mean operative temperature in the four scenarios was small but with a slight increase with increasing heat loads.

<table>
<thead>
<tr>
<th>Test name</th>
<th>T.1</th>
<th>T.2</th>
<th>T.3</th>
<th>T.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operative temperature [°C]</td>
<td>25.20</td>
<td>25.29</td>
<td>25.30</td>
<td>25.47</td>
</tr>
<tr>
<td>Standard error [°C]</td>
<td>0.01</td>
<td>0.01</td>
<td>0.02</td>
<td>0.02</td>
</tr>
</tbody>
</table>
The surface temperatures on the lower side of the suspended ceiling facing the room (Fig. 3) fulfilled requirements (< 14 °C) and were decreasing with increasing heat load. This was probably due to the increasing flow rate across the ceiling.

![Ceiling surface temperatures for all scenarios also listing lowest temperatures.](image)

The floor surface temperatures (Table 3) show that the thermal requirement (19<T_floor<29 °C) was fulfilled in all scenarios. Furthermore, there is no relation between floor surface temperature and the size of the heat load.

Table 3. Floor surface mean temperatures for all scenarios.

<table>
<thead>
<tr>
<th>Test name</th>
<th>T.1</th>
<th>T.2</th>
<th>T.3</th>
<th>T.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floor surface temperature [°C]</td>
<td>24.49</td>
<td>24.86</td>
<td>24.30</td>
<td>24.78</td>
</tr>
</tbody>
</table>

The vertical air temperature difference between 1.1 m and 0.1 m was slightly increasing with increased heat loads (Table 4). However, the temperature increase was small in all scenarios and they all fulfill the requirement (<2 °C).

Table 4. Maximum vertical air temperature difference (1.1 m and 0.1m) for all scenarios.

<table>
<thead>
<tr>
<th>Test name</th>
<th>T.1</th>
<th>T.2</th>
<th>T.3</th>
<th>T.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum vertical air temperature difference (0.1 and 1.1 m) [°C]</td>
<td>0.16</td>
<td>0.21</td>
<td>0.35</td>
<td>0.50</td>
</tr>
</tbody>
</table>

The calculated draught rates (Fig. 4) of the four test scenarios fulfilled the requirements (< 15 %). Increasing heat loads led to increasing draught rates. Independently of the size of the heat load the largest draught rates were registered at 0.1 m above ground at measurement point A and C, i.e. opposite to where the heat loads was located. For all scenarios the lowest draught rates were at measurement point E, i.e. near the working station. In point E the largest draught rates were recorded at a height of 1.7 m. The somewhat
lower draught rates at 0.1 m and 1.1 m in point E was most likely a consequence of the near presence of the heat loads which resulted in locally higher air temperatures and geometrical obscuration of the air flow.

Fig. 4. Draught rates for all scenarios.

Fig. 5 depicts the maximum mean air velocity in the occupied zone registered during the experiments. There is a tendency for increasing maximum mean air velocities with increasing heat loads. The class A requirement was only fulfilled for scenario T.1 and T.2. Scenario T.3 fulfils requirements for indoor climate category B while T.4 exceeds requirements for category C.

Fig. 5. Maximum registered mean air velocities in occupied zone for all scenarios using all measurement points in the occupied zone.

3.2 Room air distribution

Data from the experiments is investigated to achieve a better understanding of the room air distribution and movement using this ceiling solution as a diffuse ceiling inlet.

The vertical air velocity profiles for all scenarios are depicted in Fig. 6. The velocity variations in the occupied zone at the working station
(measurement point E) are acceptable in terms of comfort. Somewhat high air velocities are registered near the ceiling in all five measurement points for all four scenarios, and just above floor level at measurement point A, B and C. Acceptable air velocities are recorded in the occupied zone of the room for measurement point A, B, C and D. In general, Fig. 6 indicate an upward flowing air movement at the heat sources at point E (due to their convective heat plume) leading to a room size vortex in longitudinal direction of the room with downward flowing air in the part of the room opposite to the heat sources in point E.

Fig. 6. Vertical air velocity profiles for all scenarios.

Table 5 shows limited vertical temperature gradients for all scenarios. This suggests a high degree of mixing of the room air.

<table>
<thead>
<tr>
<th>Test name</th>
<th>T.1</th>
<th>T.2</th>
<th>T.3</th>
<th>T.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum vertical air temperature gradient in occupied zone [°C/m]</td>
<td>0.12</td>
<td>0.13</td>
<td>0.21</td>
<td>0.27</td>
</tr>
<tr>
<td>Maximum vertical air temperature gradient in whole room [°C/m]</td>
<td>0.11</td>
<td>0.09</td>
<td>0.09</td>
<td>-0.01</td>
</tr>
</tbody>
</table>

The vertical temperature profiles for the measurement plan in Fig. 2 for all scenarios are depicted on Fig. 7. The temperature in measurement point E was in general approximately 0.5 °C higher than in the other measurement points due to the nearby presence of the heat sources. Measurements at 0.1 m above ground at measurement point E were omitted from the results due to unreliable measuring results caused by fluctuating heat fluxes in the near zone of the heating blankets. Instead the temperature profile at this location was estimated using linear regression based on values from 1.1 m and 0.6 m. There is a tendency for decreased air temperatures closest to the ceiling in scenario T.4 (which has the largest supply flow rate) most likely as a
This further supports the notion that supply air from plenum to the room primarily enters the occupied zone in the part of the room opposite to the heat sources in point E.

Fig. 7. Vertical air temperature profiles for all scenarios.

Fig. 8 shows that the difference between ceiling surface temperature and air temperature in the room increases with increasing heat load and supply flow rates. However, the variations in surface temperatures did not cause any significant changes in the vertical air temperature profiles in the room. No clear tendencies were found between the floor surface temperature and the heat load. A plausible explanation for this could be the near influence of the heating blankets on the floor and/or variations in the uncontrollable heat flux into the concrete deck.

Fig. 8. Vertical temperature profile of section C including ceiling and floor surface temperatures.

4. Conclusion

This paper reports on experimental investigations of a suspended ceiling system solution with non-perforated tiles used as a diffuse ceiling inlet system. The purpose of the investigation was to evaluate the system’s ability
to remove excessive heat loads from an occupied space without causing thermal discomfort. The room air distribution and flow pattern was also analyzed.

The investigations were carried out using full-scale experiments in a climate chamber equipped as a single-person office using four different heat load intensities (10, 30, 60 and 100 W/m²). For all four scenarios all criteria for indoor climate category A were fulfilled except the criterion regarding maximum mean air velocity. This criterion was only fulfilled for the 10 W/m² and 30 W/m² scenarios. The scenario with 60 W/m² fulfilled the maximum mean air velocity for category B while the scenario with 100 W/m² exceeded the criteria for category C. However, even though high mean air velocities were recorded none of the measured draught rates did exceed the draught rate criteria for class A. For all scenarios the room air flow pattern appeared to be a room-size vortex in longitudinal direction generated by the heat sources in the room. The air movement was upward close to the heat sources and downward to the floor at the wall opposite to the heat sources. This flow pattern became more pronounced with increasing heat loads.

The results support findings from previous studies, and as such the study adds to the growing evidence of the field. However, further investigations are needed to identify modifications of the ceiling product that may improve the performance of the ceiling in terms of indoor climate and comfort at high heat loads. Future work would also be to investigate the ceiling system for different office setups and room geometries.

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References