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AN EXPERIMENTAL STUDY OF DIFFUSE CEILING VENTILATION WITH SYMMETRICAL AND ASYMMETRICAL HEAT LOAD DISTRIBUTIONS

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SUMMARY

Strength and location of heat loads have a significant effect on air distribution. Symmetrical and asymmetrical heat load distributions were studied at 40 W/floor-m^2 (C40) and at 80 W/floor-m^2 (C80) with diffuse ceiling ventilation. Experiments were carried out in a test chamber (5.5x3.8x3.2 m^3, LxWxH) by conducting experiments with omnidirectional anemometers. In the symmetrical case (SC), cylindrical heat sources were located evenly on the floor. In the asymmetrical case (AC), double office layout including a seated-dummy with a laptop and a monitor was conducted next to warm window panels in the perimeter area. The results show that AC produces stronger circulating airflow pattern that SC from perimeter area to opposite corridor side, because heat load was equally distributed in SC. In AC, the air temperature was higher (0.5°C, 0.9°C) and the air speed was lower (0.02 m/s, 0.04 m/s) in the perimeter area than in the corridor side regarding the seated person zone at C40 and C80, respectively. In SC, significant horizontal temperature difference was not obtained between the perimeter and corridor sides. Thermal environment is classified as the category B defined by the EN ISO 7730:2005 for common heat load conditions and category C for peak load conditions.

Keywords: thermal environment, heat load, buoyancy flows, airflow interaction, diffuse ceiling ventilation

1 INTRODUCTION

Several studies have shown that high performance ventilation is essential for good indoor environmental quality (Seppänen, 2008). Furthermore, ventilation is essential for well-being of occupants (Mundt et al., 2004). In addition, thermal conditions have been shown to affect the performance of users on office work and schoolwork (Wargocki and Wyon, 2017). Consequently, air distribution is one of the major factors for health, comfort and performance (Müller et al., 2013). However, controlling room airflows can be challenging. The buoyancy flows and air jets may usually provide an airflow field that involves spatial and temporal gradients that are difficult to predict. Therefore, local thermal discomfort is a common complaint in modern offices. The main objective is to investigate the effects of high heat load on thermal conditions and draught with diffuse ceiling ventilation, in which an even supply of perforated suspended ceiling penetrates supply air down to occupied zone instead of local supply openings (Zhang et al., 2014). The diffuse ceiling ventilation is reported to be able to handle high heat loads without significant draught (Nielsen, 2011), hence disturbing only little the buoyancy flows from heat sources. Novelty of this study comes from detailed analysis of time and spatial averaged dataset records to discover effects on the occupied zone with symmetrical and asymmetrical heat load setup.

2 METHODS

Internal dimensions of test chamber were length 5.5 m, width 3.8 m and height 3.2 m. Thus, the floor area of the test chamber was 21 m^2. The conditions of two test cases are shown in Table 1.

The symmetrical set-up consisted of 12 cylindrical heat sources, which have a diameter of 0.4 m and a height of 1.1 m (Figure 1a and Figure 2a). Every cylinder had a similar heat power such that total heat load was either at 40 W/floor-m^2 or at 80 W/floor-m^2. The asymmetrical set-up consisted of a double office layout. Heat sources of a workstation were a seated test dummy (90±5 W), a laptop (48±3 W) and a monitor (35±2 W). Workstation was located 0.6 m from the heated window panels in a perimeter area (Figure 1b and 2b). Lights (116±6 W) were at the suspended ceiling at height 3.2 m. Heating foil (420±21 W, 5x1 m^2, LxW) was installed on the floor 0.8 m from the window wall. At the peak load condition, a cubic heat source (103±5 W) of 0.4x0.4x0.4 m^3 was located under the table of the workstation (Figure 2, rectangle near the location 13). The surface temperature of the window panel was adjusted to be between 30-40°C such that the target heat load levels were achieved.

Table 1. The test cases.

Test cases	Case 40 (C40)	Case 80 (C80)
Heat load [W/floor-m^2]	40±2	80±4
Exhaust air temperature [°C]	26±0.5	26±0.5
Supply airflow rate [l/s,m^2]	3.6±0.2	7.3±0.4
Supply air temperature [°C]	17±0.1	17±0.1

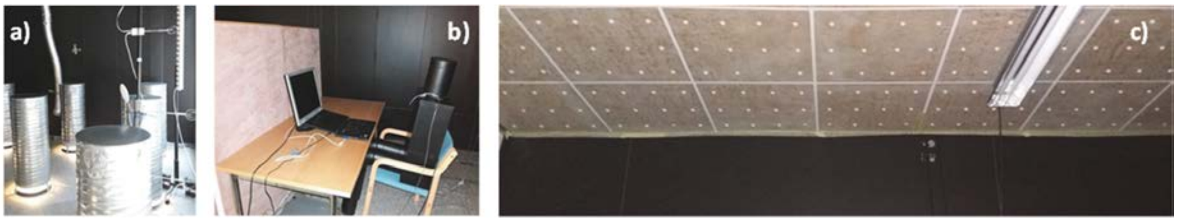

Figure 1. Test chamber: a) symmetrical setup, b) asymmetrical setup and c) diffuse ceiling inlet with the perforation rate of 0.50±0.02 %.

Supply air was discharged through the diffused ceiling straight down to the occupied zone. The perforation rate of ceiling was 0.50±0.02 % and the nozzle diameter was 14 mm (Figure 1c). The outermost row near each wall was sealed. In the symmetrical heat load set-up, the air was extracted at 0.4 m level above the floor (Figure 2a, circle at length 5 m, width 2.4 m). The diameter of exhaust duct was 0.16 m. In the asymmetrical set-up, the air was extracted correspondingly at the height of 3.2 m (Figure 2b, circle near location 11).

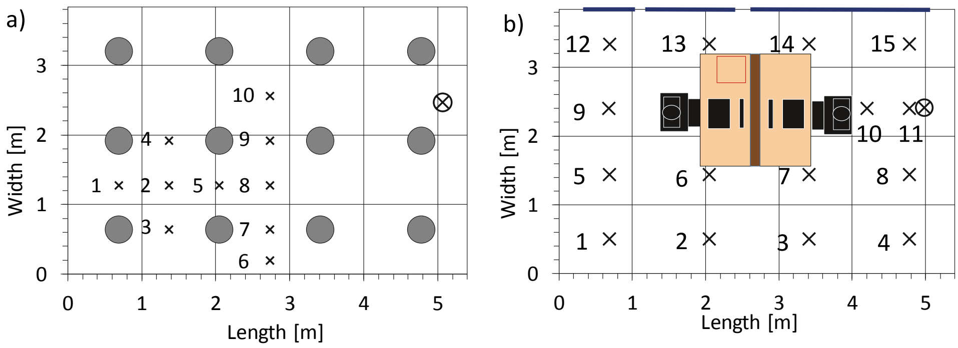

Figure 2. The measuring locations and the heat load distribution: a) the symmetrical setup and b) the asymmetrical setup.

The anemometers (Table 2) were installed at the heights of 0.1 m, 0.6 m, 1.1 m, 1.4 m, 1.7 m, 2.3 m and 2.9 m recommended in the standard EN ISO 7726:2001 (CEN, 2001).

Table 2. Measuring equipment.

Variable	Meter	Model	Accuracy
Air temperature Air speed (height 0.1-1.1 m)	Omnidirectional anemometer	Dantec dynamics Vivo Draught 20T31	Air speed (v) ±0.01 m/s±0.025v Air temperature ±0.15°C
Air temperature Air speed (upper zone)	Omnidirectional anemometer	Sensor electronic SensoAnemo 5100SF	Air speed (v) ±0.02 m/s±0.015v Air temperature ±0.2°C

In the symmetrical set-up, the sampling rate was 1 Hz in the occupied zone from 0.1 m to 1.1 m, and 0.5 Hz in the upper zone from 1.4 m to 2.9 m. In the asymmetrical set-up, the sampling rate was 10 Hz and 0.5 Hz, respectively. The measuring intervals were 0.5 h and 1 h in the symmetrical and asymmetrical set-ups.

The considered variables were air temperature, air speed and draught rate (CEN, 2005) that is expressed as

$$DR = \left(34 - t_{a,l}\right)\left(\overline{U}_{a,l} - 0.05\right)^{0.62}\left(0.37 \cdot \overline{U}_{a,l} \cdot Tu + 3.14\right) \quad (1)$$

where $t_{a,l}$ [°C] is the local air temperature, $\overline{U}_{a,l}$ [m/s] is the local mean air velocity and Tu [%] is the local turbulence intensity. Turbulence intensity is defined as

$$Tu = \frac{U_{\mathrm{SD}}}{\overline{U}} \times 100 \quad (2)$$

where U_{SD} is the standard deviation of air speeds and $\overline{U}$ is the mean air speed.

3 RESULTS

3.1 Air temperature

In the symmetrical heat load set-up (SC), a range of mean air temperatures was 0.9°C at the heat load of 40 W/floor-m^2 (C40) in the seated person zone below the height of 1.1 m. The corresponding range was higher i.e. 1.3°C at 80 W/floor-m^2 (C80). In the asymmetrical set-up (AC), the corresponding range was 1.7°C at C40 and 2.0°C at C80. Hence, the range of air temperatures was about 0.8°C larger in AC than in SC. In SC, the standard deviation (std) of mean air temperatures was 0.2°C and 0.3°C at C40 and C80. In AC, the corresponding deviation was little bit higher i.e. 0.3°C and 0.5°C, respectively. Furthermore, in the asymmetrical case, the air temperature was on average 0.5°C and 0.9°C higher in the perimeter area than in the corridor side, respectively (Figure 3). In SC, a significant temperature difference was not obtained between the corridor side and the other locations of the room, because the heat load was equally distributed on the floor. In addition, the vertical air temperature difference was small with both heat load set-ups. In SC, the average temperature increased 0.3°C in the seated person zone and decreased 0.1°C in the upper zone. In AC, the corresponding differences were 0.3°C and −0.3°C, respectively.

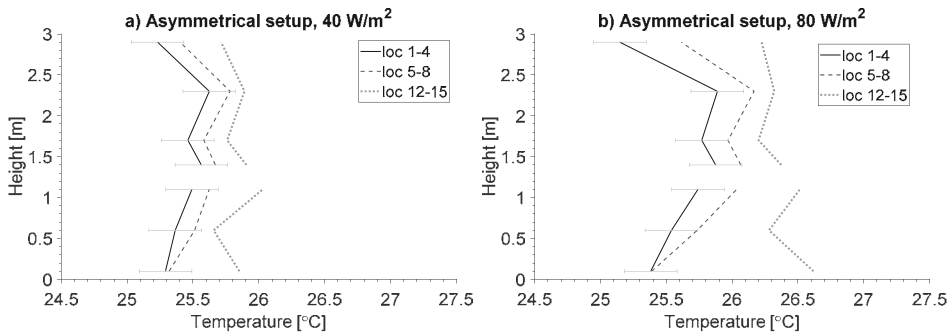

Figure 3. The air temperature between the window side (location 12-15), the corridor side (location 1-4) and the middle (location 5-8) in the asymmetrical heat load setup: a) at 40 W/floor-m^2 and b) at 80 W/floor-m^2. Error bars indicate the uncertainty in measurement.

3.2 Air speed

The mean air speed and the deviation of air speeds increased with heat load (Figure 4). In the occupied zone at height 0.1-1.1 m, the average of mean air speeds was 0.12±0.03 m/s (±std) in SC and 0.12±0.05 m/s in AC. Thus, the air average speed was at a same level in both setups. Furthermore, the deviation of mean air speeds was slightly smaller in SC than in AC (Figure 4). Compared the average values, the air speed increased with AC near the floor and also near the ceiling level. That indicates a large-scale circulating airflow pattern that was also observed using a marker smoke visualization. It should be noted that the lowest average air speed was obtained at the height of 1.1 m that is at a head level of seated person (Figure 4).

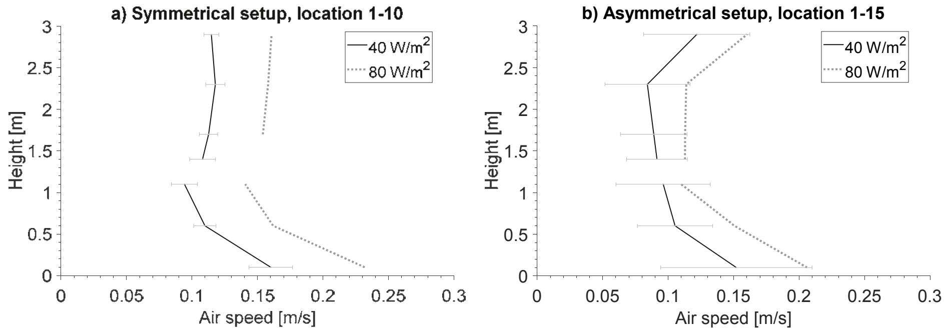

Figure 4. The average air speed at 40 W/floor-m² and 80 W/floor-m²: a) symmetrical setup, b) asymmetrical setup. Error bars indicate the standard deviation of air speeds.

In SC as the horizontal temperature differences indicate, the significant air speed difference was not obtained between the perimeter and the corridor sides. In AC, the average air speed was 0.02 m/s and 0.04 m/s in the corridor side than in the window side at C40 and C80, respectively. The lowest air speeds were at the heights of 0.6-2.3 m in the corridor side and in the middle of the room. On the contrary, the air speeds were mainly highest at those heights in the window side (Figure 5). However, the absolute maximum air speed located in the middle at the height of 0.1 m. The reason for that was the return flow from corridor wall side.

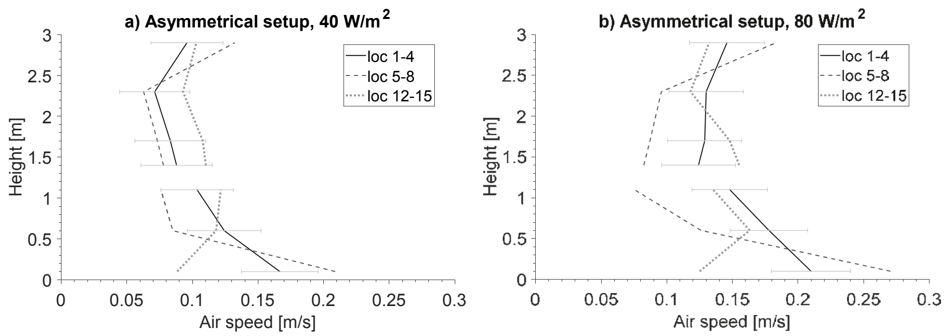

Figure 5. The air speed difference between the window side (location 12-15), the corridor side (location 1-4) and the middle (location 5-8) in the asymmetrical heat load setup: a) at 40 W/floor-m², b) at 80 W/floor-m². Error bars indicate the uncertainty in measurement.

3.3 Draught rate

Average draught rates were greater with SC than AC (Figure 6). In AC, a local maximum of draught rates (*DR*) were between 10-20 % at 40 W/m² and slightly over 20 % with 80 W/m² (Figure 7). Consequently, the results indicate category B defined by EN ISO 7730:2005 (CEN, 2005) for a common heat load conditions and category C for a peak load conditions. Furthermore, the average draught rate was higher in the corridor than in the window side, because the air speed was higher and the air temperature was lower in the corridor side than in the window side. In those cases, the average turbulence intensity was at the same level in both sides.

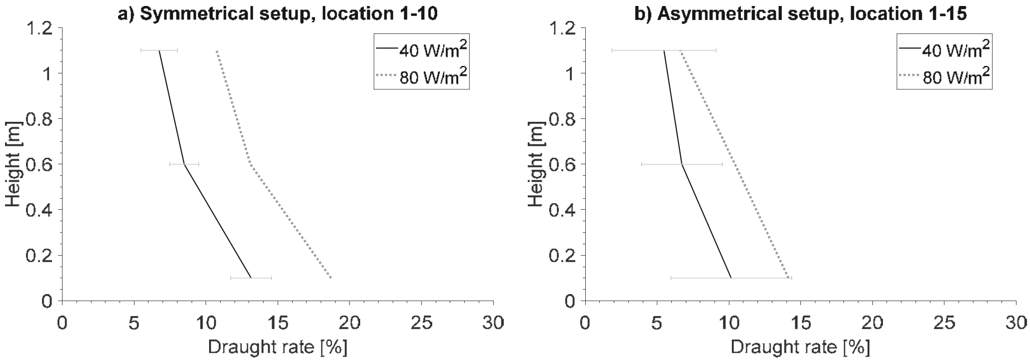

Figure 6. The average draught rate at 40 W/floor-m^2 and 80 W/floor-m^2: a) symmetrical setup, b) asymmetrical setup. Error bars indicate the standard deviation of draught rates.

a) Asymmetrical setup, 40 W/m^2

9	9	9	6
14		9	11
18	14	9	16
13	13	10	10

b) Asymmetrical setup, 80 W/m^2

10	12	9	10
18		12	19
16	21	17	20
18	17	14	16

Figure 7. The category of thermal environment below the height of 1.1 m in the asymmetrical setup: a) at 40 W/floor-m^2 and b) at 80 W/floor-m^2 based on the maximum local draught rate defined by EN ISO 7730:2005. The grey-color scale: light grey is category A (DR<10 %), medium grey is category B (DR<20 %) and dark grey is category C (DR<30 %). Maximum draught rate is shown at the corresponding location (Figure 2b).

4 DISCUSSION

The results show that the air speed levels increase with heat load. This is because the supply airflow rate will increase when the heat load will increase, thus the indoor airflow motion increase due to larger buoyancy flows and supply airflow. The study shows that the asymmetrical heat load setup (AC) provides a large-scale circulating airflow pattern from the heat sources to the other side of the room. This is because buoyancy forces accelerate the airflow upwards and turn the main flow towards corridor side. In AC, the air temperature was on average 0.5°C and 0.9°C higher in the perimeter area than in the corridor side at C40 and C80, respectively. That is because the heat sources heated up the surrounding air in the perimeter area. Consequently, the supply air decreased the air temperature at the lower level in the corridor side. Furthermore, the average air speed increased both near the floor and near the ceiling zone due to circulating airflow pattern. In addition, the average air speed was higher in the corridor side than in the perimeter area. In SC, significant air temperature or air speed differences were not obtained between the corridor side and the other locations of the room, because the heat load was equally distributed on the floor. The deviation of mean air speeds and mean air temperatures increase with heat load, most probably because the gradients may increase with heat load. The buoyancy flows dominate the air distribution with diffuse ceiling ventilation, regardless of the change in heat load distribution. The study indicates that it is difficult to achieve the category A defined by EN ISO 7730:2005 (CEN, 2005) even with a common heat load conditions and diffuse ceiling ventilation. However, a reasonable uncertainty in measurement can be around 5 % p.p. (Melikov et al., 2007), thus uncertainty in category exists correspondingly. A risk of discomfort can effectively only be minimized by reducing heat sources. The results are valid only with the given experimental set-ups, and therefore, further studies are recommended with advanced cooling systems and heat load layouts.

5 CONCLUSIONS

The results indicate that heat loads should be reduced in modern indoor environments. Otherwise, the risk of draught will increase that has been shown to have an effect on sensation with thermal environment.

AC provided a large-scale circulating airflow pattern from the perimeter area to the opposite corridor side. Furthermore, the air temperature was 0.5°C and 0.9°C higher and the air speed was 0.02 m/s and 0.04 m/s lower in the perimeter area than in the corridor side regarding the seated person zone at C40 and C80, respectively. SC provided rather uniform thermal conditions and therefore, the significant air temperature or air speed differences were not obtained between the corridor side and the other locations of the room, because of the evenly distributed heat load.

One alternative method could be to direct the buoyancy flows immediately into the exhaust at the ceiling zone or dampen the flows locally. In addition, the properly designed internal architecture may improve thermal conditions and draught discomfort such that the airflow patterns are reasonable at workstations.

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