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THE EFFECT OF THE STRENGTH OF ASYMMETRIC HEAT LOAD ON INDOOR AIR QUALITY WITH DIFFUSE CEILING VENTILATION

Weixin Zhao*, Sami Lestinen, Simo Kilpeläinen, Risto Kosonen, Juha Jokisalo
Aalto University, Espoo, Finland
**Corresponding email: weixin.zhao@aalto.fi*

SUMMARY

Indoor air quality has a significant effect on performance of building occupants. In this paper, indoor air quality was studied when the heat load was located asymmetrically with diffuse ceiling ventilation. The test chamber was furnished like an office room with two laptops and two monitors, a heated manikin and a heated dummy. The highest heat load was 53 W/m² and reduced to 27 W/m² in three cases. The outdoor air was supplied through diffuse ceiling. The air flow rate used was 4.8 l/s· m², 3.9 l/s·m² and 2.0 l/s· m² in the cases of 53 W/ m², 41 W/ m² and 27 W/ m². In term of the air change efficiency, the value of efficiency varied between 50% and 54.7%. Those values of air change efficiency are slightly better than the standard mixing ventilation. The observations indicate that the highest value of the mean age of air (67.7 minutes) is for 27 W/ m² and the lowest value of the mean age of air (21.6 minutes) is for 53 W/ m². The convection flows have a significant influence on air distribution. However, air change efficiency was above 50% in every measured location and the performance of the diffused ceiling ventilation is like fully mixed ventilation.

Keywords: diffuse ceiling ventilation, age of air, air change efficiency, indoor air quality

1 INTRODUCTION

It is recognized that sedentary workers represent the majority of the labour market worldwide. They spend 90% of their time indoors and expect to have a good indoor environment that could satisfy their need for indoor air quality. The primary purpose for ventilation is to supply clean air to occupied zone and improve the indoor air quality by extract contaminants and create a health indoor environment (Mundt et al., 2004). In many cases, ventilation is also used for cooling especially in public buildings.

The earlier studies have shown that ventilation is essential for good indoor quality. In turn, improving indoor air quality has been shown to affect significantly on learning and productivity (Wargocki and Wyon, 2017). With excellent designing ventilation and air distribution, it is possible to guarantee excellent indoor conditions (Seppänen, 2008; Haverinen-Shaughnessy et al., 2015).

Diffuse ceiling ventilation is an innovative ventilation concept where the suspended ceiling serves as an air diffuser to supply fresh air into the room. Because of the large opening area, air is delivered to the room with very low velocity. Hviid and Terkildsen (2012) stated that air distribution system of diffuse ceiling ventilation can handle high heat load without significant draught, because the air flow is determined by the heat sources and not by momentum flow from the supply openings.

Fan et al. (2013) reported that the ventilation effectiveness by diffuse ceiling ventilation is between 0.9 and 1 in the breathing zone. These results correspond well with the studies by (Nielsen et al., 2010; Chodor and Taradajko, 2013; Nielsen and Jakubowska, 2009). Hviid et al. (2013) investigated the air exchange efficiency and the result was in agreement with the finding of good mixing with diffuse ceiling ventilation. Also, there was no evidence of any stagnant zone or short-circuiting ventilation in the room based on the experiment and simulation. Kristensen et al. (2017) reported that ventilation effectiveness was comparable to conventional mixing ventilation.

The air distribution patterns in rooms with diffuse ceiling inlets may be controlled either by buoyancy flows from heat sources or by momentum flow depending on the air change rate. In the case of an air distribution pattern controlled by momentum flow a high air change rate is needed (e.g. 50 - 100 h^{-1}), which creates piston-flow. For air change rates ranges from 1 to 15 h^{-1}, the air distribution pattern in the room is controlled by the buoyancy flow generated by heat source and this could be defined as mixing ventilation.

In this paper, the effect of three asymmetric heat loads levels on indoor air quality was analysed in a typical office layout.

2 METHODS

2.1 Test room

The measurements were carried out in a test room of the internal dimensions of 5.5 m length, 3.8 m width, 3.2 m height. A double office-layout with a thermal-manikin and a sitting dummy was investigated near heated window panels in the perimeter area. Both workstations equipped with a laptop and a monitor. The manikin, dummy, laptop, monitor and lighting were simulated as internal heat loads. The layout was modelled to simulate an asymmetrical heat load distribution.

Two workstations were located 0.6 m from the heated window panels in the middle of the room. Lights were installed in the middle of the workstations at the ceiling. A laptop and a monitor were located on the both tables. Between the workstations, there was a partition wall. Fig 1 shows the workplace set-up with the thermal manikin and the seated dummy.

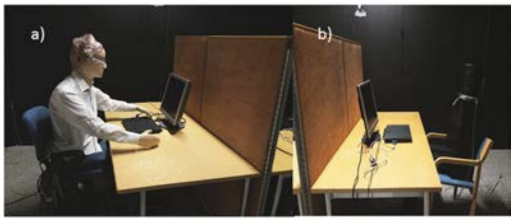

Figure 1. Workstation set-up: a) thermal manikin and b) seated dummy with laptop and monitor

2.2 The experimental set-up

Figure 2 shows a sketch of the test room with all the measurement locations (TL1-TL15). The air-flow field was measured by conducting omnidirectional anemometers for air temperature and air velocity. Gas-analysator is used to measure and log concentration values of CO_2 at locations (IL1-IL5). IL1 and IL2 were close to breathing zone, IL3 was close to the heated window, IL4 was mirrored location with respect to IL3, and IL5 was close to exhaust location.

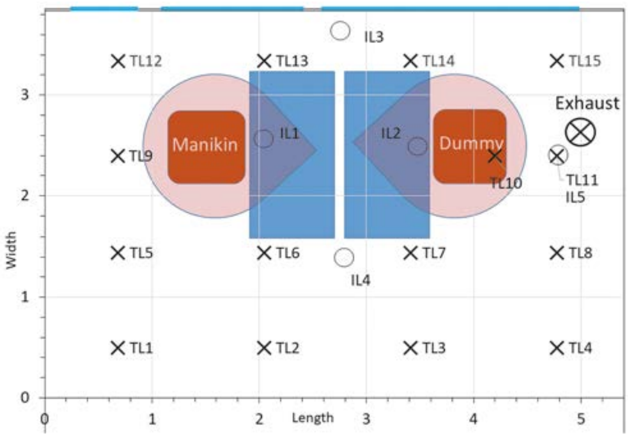

Figure 2. Test chamber layout seen during the tracer gas measurement

CO_2 was chosen as tracer gas as it is a real contaminant in occupied spaces, easy to measure and behaves like air. Gas-analysator is able to measure concentration values and the measurement interval was 1 minute at each location. For all the chosen locations, the measured height was 1.1 m (occupied zone), except for the exhaust location, that was at the suspended ceiling at height 3.2 m.

In the occupied zone of the test chamber, air temperature and air velocity were measured in three heights (0.1 m, 0.6 m, 1.1 m) above the floor level.

The goal of the ventilation system was to maintain an internal set location temperature of 26±0.5°C in the occupied zone at the high of 1.1 m from the floor. Consequently, the supply air flow rate was adjusted for providing sufficient cooling power in responding to heat load change. The supply air temperature in the room is set to 17°C.

All the measurement instruments that will be used during the experiments are summarized in Table 1.

Table 1. The measuring instruments and visualization equipment

Variable	Meter-type	Model	Accuracy
Temperature Air velocity	Hot-sphere anemometer	Dantec dynamics Vivo Draught 20T31	Air velocity: Range 0.05-1.0 m/s ±0.01 m/s±0.025 v_{means} Temperature: Range 0-45°C ±0.15°C
Tracer gas concentration	gas-analysator	Bruel Kjaer Multi-gas Monitor Gas Analyzer Type 1302	Detection threshold:10^{-3}ppt-1ppt Range: upper detection limit=100000 times the lower detection limit

The supply air was distributed through a suspended ceiling down to the room. Supply air is released inside the upper plenum chamber by a six meters long ductwork and then flows slowly into the room through perforations in the suspended ceiling. In Fig. 3, the diffuse ceiling panel has dimensions of 600x600x20 mm and is made of glass-wool-plate elements. Each hole has a diameter of 14 mm and the overall degree of perforation was approximately 0.5%. The diffuse ceiling ventilation can produce rather equal supply air distribution with low pressure drop, which forces the airflow through the suspended ceiling down to the occupied zone.

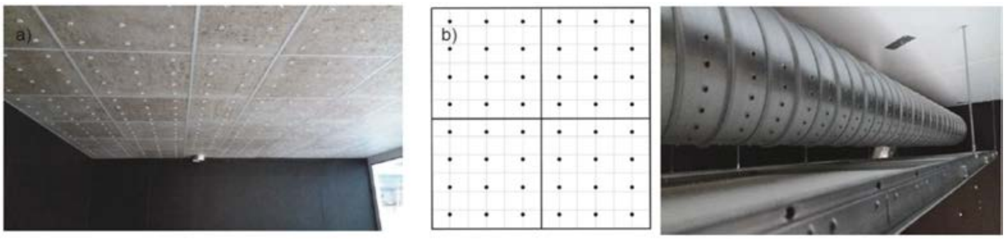

Fig .3 (a) Diffuse ceiling inlet structure, (b) hole density as 0.5% of the entire roof area,(c) duct diffuser located in the plenum chamber

2.3 Test cases

In the test chamber, the solar heat load was simulated using warm window panels that were heated up to 30-35°C to have typical window temperatures in practice. In the study, three heat load levels were measured. The measured specific heat loads were: 27 W/ floor-m^2, 41 W/ floor-m^2 and 53 W/ floor-m^2.

The parameters of the test cases are summarized in Table 2.

Table 3 shows the used heat loads in different measurement cases. Occupancy, computers and lighting have a constant value during all the measurement. The heat flow of the window panels has to be adjusted to cover the cooling load required.

Table 2. Heat load per floor area, air flow rate and temperatures

Case	Heat load per floor area(W/m²)	Air flow rate(l/s·m²)	Supply air temperature (°C)	Indoor air temperature (°C)
1	53	4.8	17	26
2	41	3.9	17	26
3	27	2.0	17	26

Table 3. Used heat loads in the tests

Case	Dummy (W)	Manikin equipment (W)	Manikin (W)	2 laptops (W)	2 monitors (W)	2 lights (W)	Window plates (W)	Total heat load (W)	Total heat flux (W/m²)
1	85	67	83	96	70	116	584	1099	53
2	85	67	83	96	70	116	340	857	41
3	85	67	83	96	70	116	56	573	27

The concept of age of air is introduced by Sandberg (Sandberg, 1981). The age of air in a space can be considered as a measure of the freshness of the environment and it is defined as the time that has elapsed since the air entered the space through an opening (Etheridge, 2015).

The ventilation efficiency within the building was also investigated using the air change efficiency. The air change efficiency is a measure of how quickly the air in a room can be replaced with the clean air compared to the shortest possible time it takes to replace the air with the same ventilation airflow in the room (Etheridge, 1996).

The definition of the air change efficiency can be explained as the ratio between the theoretically shortest possible air change time τ_n and of the average time it takes to replace the air in the room $\overline{\tau_r}$, Equation (1)

$$\epsilon^a = \frac{\tau_n}{\overline{\tau_r}} \cdot 100\% \ . \tag{1}$$

The actual air change time $\overline{\tau_r}$ can be derived from the room mean age of air $\langle\overline{\tau}\rangle$ by Equation (2)

$$\overline{\tau_r} = 2\langle\overline{\tau}\rangle \ . \tag{2}$$

where ϵ^a is air change efficiency, $\langle\overline{\tau}\rangle$ is mean age of air, $\overline{\tau_r}$ is actual air change time. So air change efficiency also is the ratio between τ_n and τ_r, Equation (1) becomes

$$\epsilon^a = \frac{\tau_n}{2\langle\overline{\tau}\rangle} \cdot 100\% \ . \tag{3}$$

3 RESULT

3.1 Age of air

From the tracer-gas experiments, the air change efficiency of the room and the mean age of air were calculated. The mean age of air for five measurement locations is illustrated in Figure 3. The highest value of 67.7 minutes was obtained for the heat load of 27 W/m².

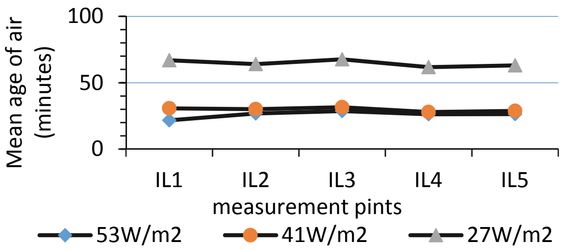

Figure 3. Mean age of air for three heat loads

The difference between the cases of 53 W/m² and 41 W/m² was insignificant. At those heat loads, the age of air reduced to 22 and 32minutes, respectively. It should be noted that the differences of mean age of air between the locations were very small for each case, which indicates good mixing was achieved in the room space.

It could be expected that with the lowest heat load conditions, the supply air flow rate is not dominating the air movement. When the hear load increased, the flow structure changed and the convection flow dominates and that causes well-mixing in the room space.

3.2 Air change efficiency

The effect of the heat load on air change efficiency is shown in Figure 4. The average room air temperature is presented in Figure 5, respectively. The average of air change efficiency increased when the heat load increased from 27 W/m² to 53 W/m². The value of the average air change efficiency varied between 50% and 55 % with all cases.

The air exchange efficiency showed the same trend than the mean age of air. The highest air exchange efficiency was achieved at the highest heat load when the air flow rate is also highest. However, the difference of air exchange efficiency between the cases of 53 W/m² and 41 W/m² is insignificant. The air change efficiency values were 50% in every measurement location in the case of 27 W/m².

For IL1 and IL2 at the breathing zone, the air change efficiency was at the same level than in the other locations. Thus, with the asymmetric load located, the air change efficiency is the same in all locations of the room spaces.

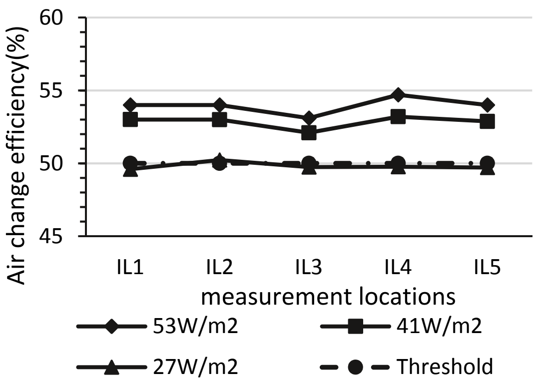

Figure 4. Air change efficiency

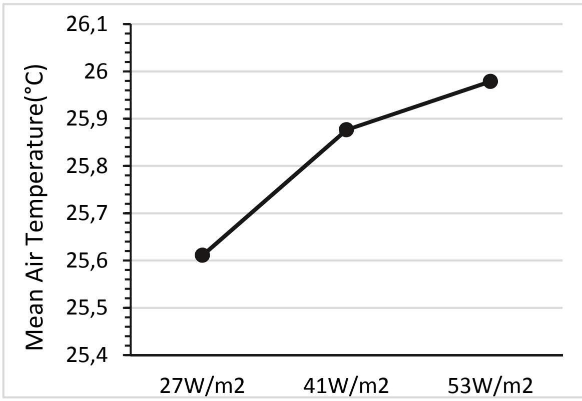

Figure 5. The average room air temperature

4 DISCUSSION

The air change efficiency of diffuse ceiling ventilation was quite close to mixing ventilation. The measured air change efficiency values did not much change when the heat load levels varied between 27 and 53 W/m^2. It should be also noted that the air change efficiency is the same level over the whole room space and asymmetric load distribution does not have any effect on air change efficiency. However, the amount of the supply air flow rate has effect on age of air. Still when the load is higher than 41 W/m², the increase of air flow rate does not change anymore age of air.

Diffuse ceiling ventilation seems to be a good solution for providing a good indoor air quality by avoiding the formation of stagnation zones and with a good air replacement in all the locations of the room. The results indicate that there is a specific threshold value of heat loads where the flow structure changed and also the age of air changed. Further investigation should be conducted on the different layout and different size of office installed with the diffuse ceiling ventilation. The different boundary conditions also should be studies in future.

5 CONCLUSIONS

The purpose of this study was to investigate the indoor air quality of a diffuse ceiling ventilation in asymmetric load conditions. With three different heat load levels, air change efficiency was above 50% in all measured locations. This indicates that the condition of fully mixing ventilation was achieved in all location of the room. The asymmetric load distribution does not have any effect on air change efficiency. With higher heat loads levels, the required higher of the supply air flow rate reduced the age of air. However, when the heat load changed from 41 W/m^2 to 53 W/m^2, the age of air is not thereafter improved.

6 ACKNOWLEDGEMENTS

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