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Krajík, Michal; Simone, Angela; Tomasi, Roberta; Olesen, Bjarne W.

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EVALUATION OF THE INDOOR ENVIRONMENT IN AN OFFICE ROOM EQUIPPED BY DISPLACEMENT VENTILATION AND RADIANT FLOOR COOLING

Michal KRAJČÍK[1,*], Angela SIMONE[2], Roberta TOMASI[3] and Bjarne W. OLESEN[2]

[1]Faculty of Civil Engineering, Slovak University of Technology, Bratislava, Slovakia
[2]ICIEE, BYG, Technical University of Denmark, Kgs. Lyngby, Denmark
[3]Department of Industrial Engineering, University of Padova, Padova, Italy

*Corresponding email: michal.krajcik@stuba.sk

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SUMMARY

The effects of displacement ventilation combined with a floor cooling system on indoor climate in a simulated office room were examined by experimental measurements. The aims of the study were to verify and extend the results of previous measurements and to investigate the system functionality at low air flow rates. Vertical air temperature, operative temperature, air velocity profiles and equivalent temperatures were used to characterize thermal comfort. Contaminant removal effectiveness and local air change index were measured in order to characterize ventilation effectiveness in the occupied zone. Vertical air temperature differences of 5-6 K for a sitting person occurred, except for one case when the temperature of the simulated window was decreased. In the occupant´s breathing zone the ventilation effectiveness was always better than in other points in the room and even at the lowest nominal air change rate of 1.5 h^{-1} it was significantly better than 1.

INTRODUCTION

Displacement ventilation may be a suitable choice for spaces where the requirements on the air quality are tight and the internal heat gains are high due to the presence of workers and office equipment, being the source of pollution at the same time (Skistad et al., 2004). With the aim to reduce artificial light during the winter season while remaining within the standard limits on lighting, large window areas are often adopted, resulting in higher cooling demands in summer due to the presence of solar radiation. Although the supplied ventilation air can partially decrease the cooling need, high amounts of supplied cold air may result in high velocity and turbulent layer of cold air, and consequently in draught (Melikov and Nielsen, 1989). In order to remove the peaks of cooling load and to avoid draught, displacement ventilation can be combined with a low exergy radiant floor cooling system. Due to the low temperature difference between the room air and the surface temperatures required, radiant systems work well with renewable sources and can therefore be considered as a high energy efficiency solution (Olesen and Mattarolo, 2009; Babiak et al., 2010). Mundt (1995) summarized results of many experiments in rooms with displacement ventilation and concluded that in a typical displacement ventilated room the increase in temperature for the supply air at the floor level is due to the radiation from the warmer ceiling to the cooler floor and the subsequent convection heat transport to the air at the floor level. However, in the

experiments with displacement ventilation combined with floor cooling by Causone et al. (2010), the measured vertical air temperature differences were generally higher than limits in the standards and higher than could be expected according to the 50%-rule (for the 50%-rule see Skistad et al., 2004), while the draught rating was up to 15 % in the occupied zone. The high thermal gradients were explained by the fact that the heat transmitted by radiation from the warmer ceiling to the floor was directly removed by the cold floor, and it was therefore not further transmitted to the supply air at floor level by convection. The ability to remove a contaminant from a simulated active pollution source at the breathing level was very good at higher supply air flow rates (above 3 liters per second per square meter), whereas the improvement was low comparing with a mixing ventilation system, when the supply air flow rate was as low as 2.1 liters per second per square meter.
In the present study, thermal comfort and ventilation effectiveness in a small office equipped with a semicircular wall DV unit and a radiant floor cooling system were studied under summer conditions. The aims of the study were to:

1) verify the results of previous measurements;
2) extend the results to different boundary conditions, using additional IAQ indicators;
3) investigate functionality of the system at air flow rates lower than typical for a single displacement ventilation system (down to 1.5 air changes per hour).

Different air flow rates, supply air temperatures and floor temperatures were tested. Contaminant removal effectiveness (CRE) was measured to study the ability of the combined system to remove pollution from a simulated active contaminant source. Local air change index (ACE) was obtained by tracer gas age-of-air measurements in order to investigate the distribution of the fresh air to the occupants.

METHODOLOGIES

The experimental measurements were carried in an experimental chamber (Tomasi et al., 2013), representing a room in a low energy building during realistic summer conditions, typical e.g. for the Mediterranean area in Europe. Five radiant hydronic panels with a total area of 8 m^2 were located on one of the walls to simulate a warm window surface. The room was equipped with a thermal manikin and a heated thermal dummy simulating two seated occupants, two desk lamps and two heated metal boxes simulating office equipment such as computers or similar. The arrangement of the experimental room and the location of displacement ventilation and air exhaust devices can be seen in Figure 1. A view of the room is shown on the left side from the position of manikin´s head.

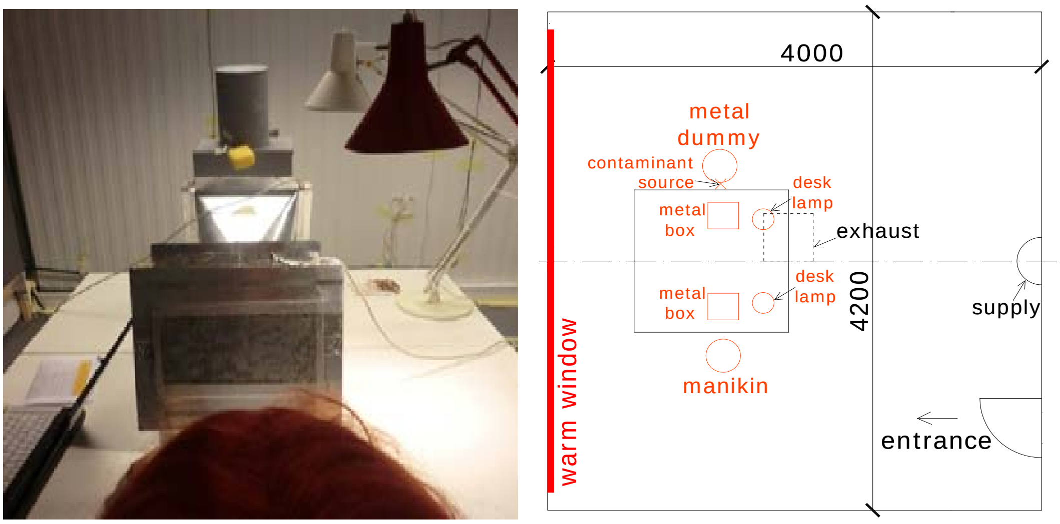

Figure 1. Left: View of the experimental room. Right: Layout of the experimental room.

The thermal insulation of the manikin clothing was estimated to 0.6 clo (including the chair) according to ISO 9920 (2007). Freon (R134a) was used as the tracer gas in the ACE measurements and CO_2 was used as the tracer gas in the CRE measurements. The ventilation effectiveness indicators are described in detail by Mundt et al. (2004). A perforated table tennis ball covered with a sponge material was used as the contaminant source, attached to the heated thermal dummy at the height of 1.1 m above the floor, simulating e.g. a body odour from the occupant. The measuring instruments are described in detail by Tomasi et al. (2013). The air was supplied by a semi-circular displacement ATD at the floor level, and exhausted at the ceiling. The cases investigated represent different combinations of supply air temperatures, air flow rates and floor temperatures, when the air flow rate and temperature of the supply air were kept constant and the cooling floor was adjusted in order to keep the reference room air temperature. The experimental conditions are listed in Table 1. In case 5 it was not possible to reach the required temperature at the reference point and the heat load by the window had to be decreased. The heat gains simulated in the experiments represent various combinations of U-values, solar irradiation and solar heat gains coefficient (SHGC). The U-value of 1.2 W/(m^2.K) was considered in all cases. Assuming a SHGC equal to 0.24, the heat load by the window corresponds to a rather high solar irradiation (about 200 W/m^2), representing the solar irradiance typical for the Mediterranean area in Europe. The external air temperature of 30 °C for the others were considered.

Table 1. Experimental conditions during tests with displacement ventilation.

Case	Nominal ACH (h^{-1})	Calculated heat gain by the window (W/m^2)	Internal heat gains (W/m^2)	Room air temperature at the reference point (°C)	Supply air temperature (°C)	Temperature of the simulated window (°C)	Floor temperature (°C)
case 1	3.2	27.4	31	26	15.7	33.2	20.1
case 2	4.5	27	31	26.2	16.1	33.3	25.4
case 3	3	26.7	31	26.2	22.3	33.2	19.7
case 4	2.1	27	31	26.2	20.1	33.3	19.8
case 5	2.1	13	31	26	24	29.4	20
case 6	1.5	25.5	31	26.4	16.5	33.1	19.4

In the air change efficiency measurements three channels were used, sampling in sequence, with the sampling time about 45 seconds for each. To obtain data on measurement uncertainty, additional measurements were performed with two fans mixing the air in the experimental chamber. Two cases with complete mixing were performed (cases 2 and 3) at two measurement points for each. Comparison with the expected true value of 1 at complete mixing provides information on the uncertainty of the measurements. Moreover, two cases were repeated (case 2 at two points and case 5 at four points). The measurement positions of the thermal environment and ventilation effectiveness indicators are shown in Figure 2. The layout of the chamber was considered symmetrical and the measuring points were therefore located only in the half of the room.

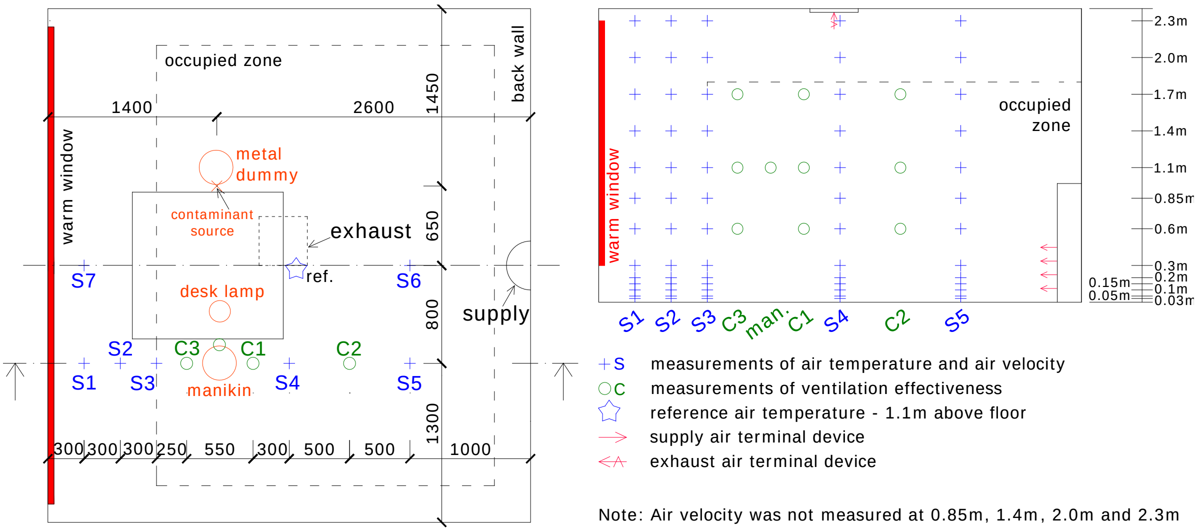

Figure 2. Plan and cross-section of the experimental chamber during tests with displacement ventilation.

RESULTS AND DISCUSSION

Thermal environment

Vertical air temperature differences of 5-6 K occurred for a sitting person, except for case 5, when the temperature of the simulated window was decreased by 1.7 K (Table 2). The vertical air temperature profiles are visualized in Figure 3.

Table 2. Experimental conditions and parameters of the thermal environment.

Case	Nominal ACH (h^{-1})	Heat gain by the window (W/m^2)	Internal heat gains (W/m^2)	Supply air temp. minus room temp. (°C)	Room temp. minus window temp. (°C)	Floor temp. minus room temp. (°C)	Vertical air temp. difference in occupied zone 1.1m-0.1m (°C)	Air velocity in occupied zone at 0.1m (m/s)
case 1	3.2	27	31	-10.3	7.2	-5.9	5.8	0.13
case 2	4.5	27	31	-10.1	7.1	-0.8	5.4	0.18
case 3	3.0	27	31	-3.9	7.0	-6.5	n/a	0.07
case 4	2.1	27	31	-6.1	7.1	-6.4	5.0	0.09
case 5	2.1	13	31	-2.0	3.4	-6.0	3.3	0.07
case 6	1.5	26	31	-9.9	6.7	-7.0	6.3	0.09

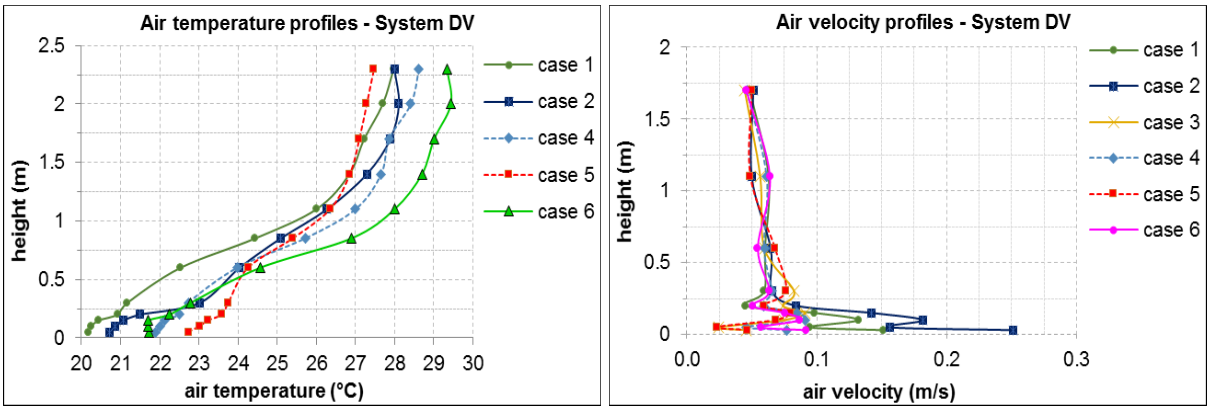

Figure 3. Left: Average air temperature profiles in the occupied zone. Right: Average air velocity profiles in the occupied zone.

Figure 3 shows that the vertical differences were higher than recommended by the standard EN ISO 7730 (2005). At such a high thermal stratification the thermal perception of the occupants differs from that in a homogeneous thermal environment; moreover, it may be difficult to keep the desired temperature at the reference point, as showed by the experiments. No serious risk of discomfort due to draught occurred close to the occupied zone due to the relatively low ventilation rates and appropriate choice of the supply air diffuser, except for case 2 at the highest air flow rate, when the air velocity was elevated at the floor level. The difference between the equivalent temperature on the head and on the feet confirms the high vertical air temperature differences, when it ranged from 5-6 K for case 2 to about 8 K for case 6 (Figure 4) with the lowest floor temperature and the smallest nominal air change rate.

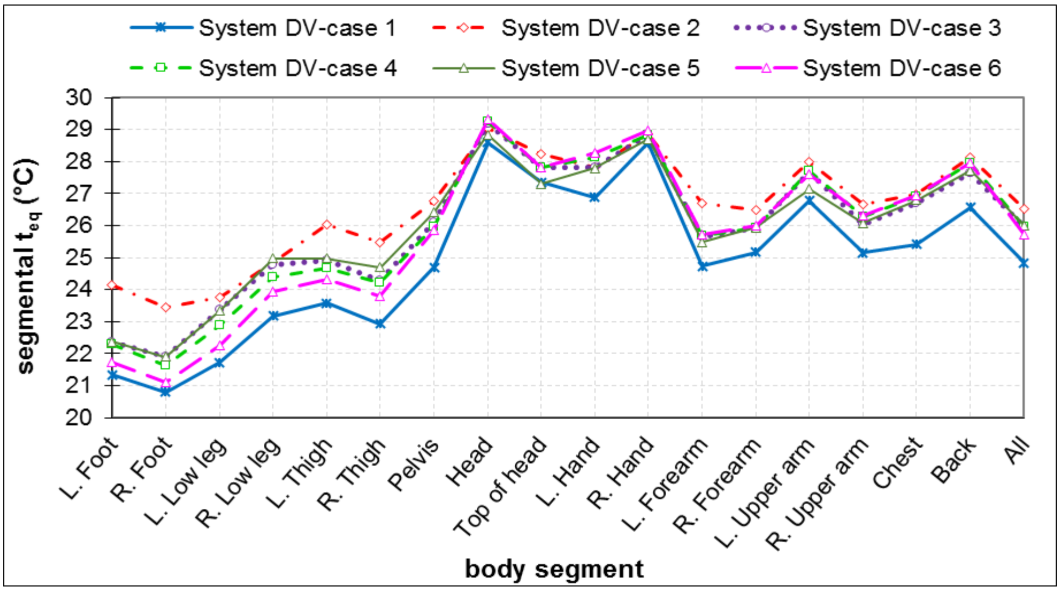

Figure 4. Equivalent temperature of different body segments of the thermal manikin.

Contaminant removal effectiveness

The results of CRE are listed in Table 3. Even at a nominal air change rate as low as 1.5 h^{-1} the average CRE in the occupied zone and also the personal exposure index are more than 1, thus better than at complete mixing.

Table 3. Experimental conditions and ventilation effectiveness for System DV.

Case	Nominal ACH (h^{-1})	Heat gain by the window (W/m^2)	Internal heat gains (W/m^2)	Personal exposure index[a]	CRE C1 at 1.1m	CRE Occupied zone[b]	ACE Inhalation zone	ACE C1 at 1.1m	ACE C1 at 1.7m	ACE C2 at 1.1m
case 1	3.2	27	31	7.62±0.42	5.57	4.10	n/a	1.27	1.04	n/a
case 2	4.5	27	31	18.58±2.06/ 19.90±2.11*	16.22/ 10.86*	5.82/ 5.00*	1.59	1.80/ 1.62*	1.07/ 1.08*	0.92
case 3	3.0	27	31	12.60±2.65	4.39	2.07	1.99	1.91	1.05	0.94
case 4	2.1	27	31	6.79±0.11	2.89	1.73	1.82	1.55	1.05	1.03
case 5	2.1	13	31	6.49±0.56	2.28	1.47	2.10/ 1.91*	1.14/ 1.15*	1.11/ 1.07*	1.17/ 1.15*
case 6	1.5	26	31	3.01±0.11	1.45	1.34	1.49	0.99	1.06	1.07

* repeated measurement

a) Mean of three measurements at one position (dummy´s breathing zone) ± 95 % confidence limit.

b) Mean value from one measurement at six positions (C1, C2 and C3 at 1.1m and 1.7m).

The vertical CRE profiles are visualized in Figure 5 for each case (average from points C1, C2 and C3). At 0.6 m above the floor the CRE is very high due to the fresh air supply at the floor level, whereas at 1.7 m the CRE is close to 1 and it is equal for all cases, regardless of the nominal air change rate. At 1.1 m above the floor, representing the inhalation zone of a sitting person, the CRE was always more than 1, even the smallest air flow rate of 1.5 h^{-1}.

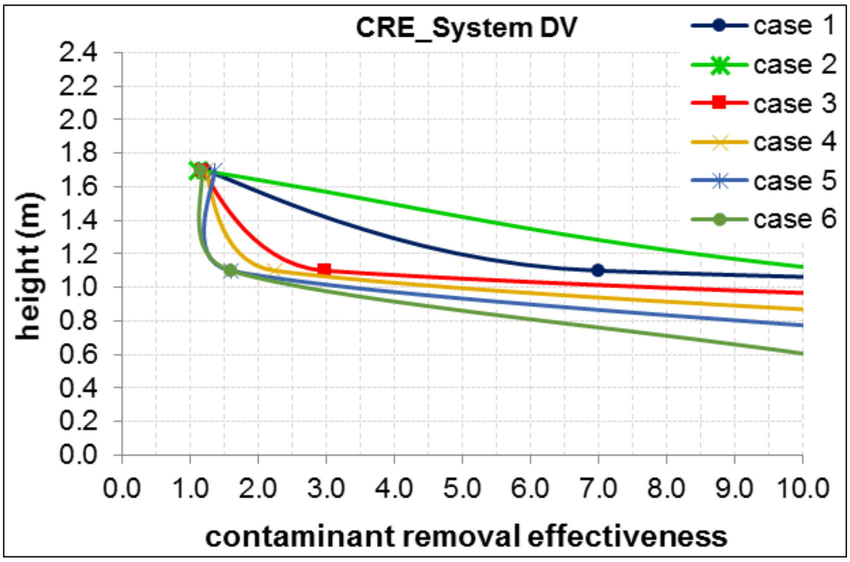

Figure 5. Profiles of contaminant removal effectiveness (CRE).

Figure 6 shows personal exposure index (CRE at the breathing zone of the occupant, simulated by thermal manikin) as a function of the nominal air change rate. The linear regression line in the figure has a coefficient of determination of 0.87, meaning that the nominal air change rate is able to explain 87 % of the variation seen in the personal exposure index of a sitting person. At the same height, similar trend can be observed for the position C1 between the manikin and the supply diffuser. In the position C3, between the manikin and the warm window, the trend was not clearly visible.

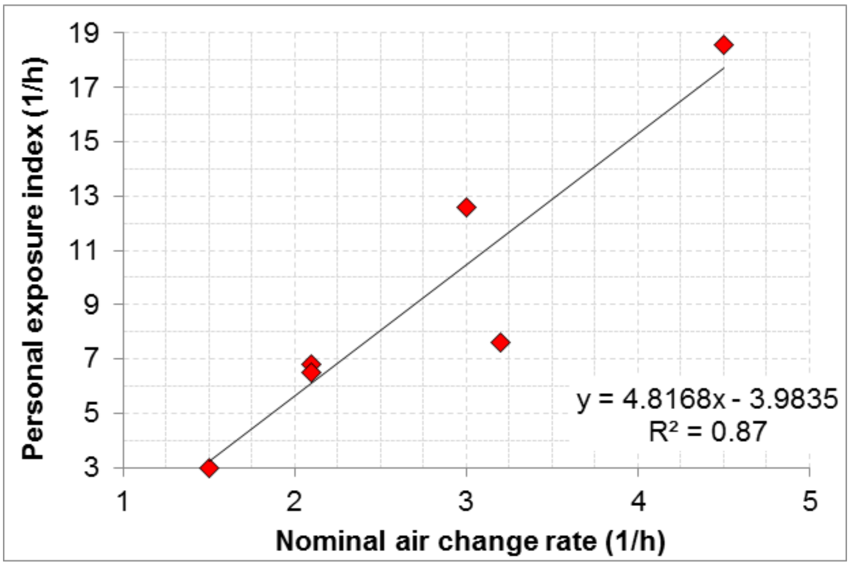

Figure 6. Personal exposure index as a function of the nominal air change rate.

Air change efficiency

Results of the air change efficiency measurements are presented in Table 3 and in Figure 7. In case 1 at the manikin breathing point and in position C2 at 1.1 m above floor level the results of ACE are not available. The values of ACE in Figure 7 are based on mean age of air calculated adding the mean transit time and the mean presence time (Krajčík, 2013). The results of ACE in the Figure 7 show the following:

- fresh air distribution was not superior to complete mixing in the point farther from the internal heat sources and at higher level (1.7 m);
- fresh air distribution was not superior to complete mixing in cases 5 and 6 at small air change rate (case 5 and 6) and high supply air temperature (case 5) in positions C1 and C2, farther from the internal heat sources;
- fresh air distribution to the inhalation zone of the occupant was always significantly higher than for complete mixing; it was also significantly higher close to the internal heat sources for cases with higher air change rates (cases 1 to 3), and for a case with low air change rate (2.1 ACH) and low supply air temperature (20.1 °C) (case 4).

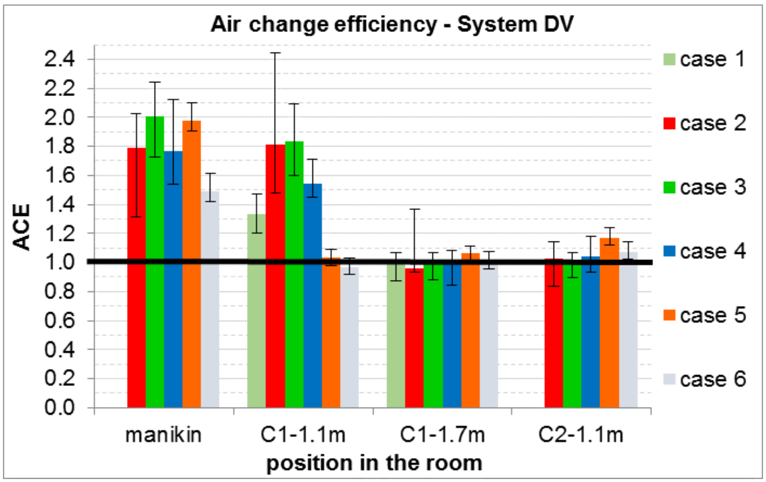

Figure 7. ACE measured in the occupied zone

<u>Uncertainty based on tests at complete mixing:</u>
With complete mixing, when the room air was mixed by two fans operated in the room, the 95 % confidence limit was 0.97±0.08, allowing to say that the 95 % confidence interval based on measurements performed at complete mixing compared with the true value of 1 is ±0.1. The analysis is based on measurements at four points.

<u>Uncertainty based on repeated measurements:</u>
For each repeated measurement, relative discrepancy between the two measurements was calculated and the confidence limit is given as a relative value. The 95 % confidence limit presents ±4% of the measured value. The analysis is based on measurements at six points.

<u>Uncertainty due to estimation of mean transit time:</u>
The uncertainty presented by the error bars in Figure 7 represents the impossibility to exactly determine the time elapsed from the moment the tracer gas entered the room until it reached the measurement point. The error bars present absolute values, and in the reality the error is likely to be lower. The large confidence limits in some of the cases indicate the high relative weight of the mean transit time in the overall mean age of air value.

CONCLUSIONS

The study has shown that displacement ventilation combined with floor cooling in a room with heat gains from a shaded glazed facade may result in a rather high vertical air temperature difference. Even at small air flow rates good ventilation effectiveness can be achieved, although mainly at low air change rates the measured ventilation effectiveness better than 1 refers only to the locations very close the heat sources. The ventilation effectiveness was always better in the occupant´s breathing zone than in other points in the room, when in the breathing zone it was significantly better than 1 even at the lowest nominal air change rate of 1.5 h^{-1}.

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