CLIMA 2016 - proceedings of the 12th REHVA World Congress

volume 5

Heiselberg, Per Kvols

Publication date:
2016

Document Version
Publisher's PDF, also known as Version of record

Unsteady Displacement Ventilation in Office Environments with Varying Thermal Loads

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Abstract
Subject of this study is the experimental assessment of unsteady displacement ventilation in an office with varying thermal loads. Heat gains are due to solar radiation onto the floor and internal heat sources due to humans and office appliances respectively. The effects of unsteady volume flow rates of supply air on spacial distributions of temperature and velocity of the room air are examined. The test cases are based on results of previous CFD-simulations. Simulative results indicate advantageous conditions for unsteady air supply regarding the removal of thermal load and ventilation effectiveness in comparison to constant volume flow rates of supply air. Unsteady displacement ventilation is constituted in periodic interruptions of the supply air flow. Hence, supply air flow pulsates, in between two pulses of supply air there is a pause without supply. In general, at the same inlet velocity, a pulsed supply air jet induces more room air when compared to jets generated by constant volume flow rates. Parameters for control of the HVAC-system are the periodic time consisting of pulse and pause, the inlet velocity of the pulsed supply air flow, average volume flow rate respectively, of the supply air. The final parameter to be assessed is the temperature of the supply air entering the room.

Keywords: Unsteady Displacement Ventilation – Thermal Load Distribution

1. Introduction

Unsteady displacement ventilation means that instead of constant flow rates, supply air flow rates change periodically. Different durations of supply air pulses, as well as pauses in supply are possible. Preceding CFD simulation shows for all tested cases, that at the same inlet velocity, the pulsed supply air jet induces more room air when compared to jets generated by same constant volume flow rates (seeFig. 1). When applying displacement ventilation, air is usually supplied with low velocity and turbulence, as well as not too large undertemperature to avoid draft in the foot region of users. Hence, one goal of unsteady displacement ventilation is to increase the undertemperature between supply and room air. This enables same removal of thermal load at smaller volume flow rates resulting in decreasing energy demand for fan power. Secondly, another possible advantage is examined. Problematic conditions for steady supply rates are identified in CFD simulations for rooms with increased thermal loads causing inadvertent convective drafts. Hence, the supply air does not attain the desired penetration depth into the room. This causes short-circuit flow between inlet and extract

air. Unsteady supply air pulses with very limited duration could be inserted into the room at a higher momentum. Due to the increased inlet velocity of the supply air the penetration depth into the room is increased depending on the distribution of heat sources in the room.

2. Background

Unsteady displacement ventilation is constituted in periodic interruptions of the supply air flow. Instead of constant flow rates, supply air flow pulsates, in between two pulses of supply air there is a pause without supply. Depending on the time t_{Puls} in s of the pulse, the time t_{Paus} of the pause and flow rate $\dot{q}_{Puls}$ in m³/h of the pulse, the average volume flow rate $\bar{\dot{q}}_{mean}$ in m³/h results. One period with time t_P in s thus consists of $t_{Paus} + t_{Puls}$. Interruption during t_{Paus} results in waves along the floor, because room air from higher and warmer layers flows down after every pulse and mixes with the colder supply air.

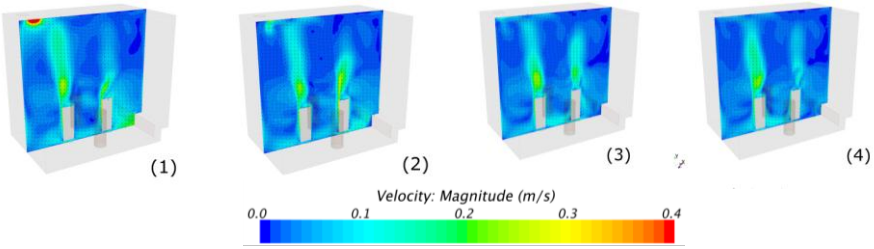

Fig. 1: CFD simulation of basic test case, displacement ventilated room with unsteady supply air flow showing one period consisting of pulse and pause

Underlying principle of displacement ventilation in general is to supply cool air to the lower part of a room, from floor integrated inlets or wall mounted diffusers. As the supply air is also denser than the room air, a layer of fresh air forms on the floor. Heat sources in the room warm up the air around them and effect convective plumes. The warmed air is transported to higher parts of the room, resulting in a gradient over the room height with stable temperature stratification. In the upper part of the room, outlets are positioned to extract the used air [1]. If the heat sources are also the sources of contamination, there is a concentration gradient over the room height, with larger concentrations at higher levels [4]. A displacement ventilated room with stratified layers of temperature shows a lower air temperature at floor T_{af} in °C than the temperature T_f in °C of the surface of the floor itself. This is caused by the radiative heat exchange between the warm ceiling and walls of the room. Further, the convective heat transfer from the surface of the floor to the supply air with temperature T_s in °C determines T_{af} [2]. If assumed that there is negligible amount of induction of room air

into the supply air flow $\dot{q}_s$ in m³/h, the simplified model for the dimensionless temperature κ at floor is [5]:

$$\alpha_{\mathrm{r}} \cdot A \cdot (T_{\mathrm{e}} - T_{\mathrm{f}}) = \alpha_{\mathrm{cf}} \cdot A \cdot (T_{\mathrm{f}} - T_{\mathrm{af}}) \tag{1}$$

$$\dot{q}_{\mathrm{s}} \cdot \rho \cdot c_{\mathrm{p}} \cdot (T_{\mathrm{af}} - T_{\mathrm{s}}) = \alpha_{\mathrm{cf}} \cdot A \cdot (T_{\mathrm{f}} - T_{\mathrm{af}}) \tag{2}$$

$$\kappa = \frac{T_{\mathrm{af}} - T_{\mathrm{s}}}{T_{\mathrm{e}} - T_{\mathrm{s}}} = \frac{1}{\frac{\dot{q}_{\mathrm{s}} \cdot \rho \cdot c_{\mathrm{p}}}{A} \cdot \left(\frac{1}{\alpha_{\mathrm{r}}} + \frac{1}{\alpha_{\mathrm{cf}}}\right) + 1} \tag{3}$$

with A, the floor area of the room in m². Heat transfer coefficients are α_r for radiative, α_{cf} for convective transfer, both in W/m²/K, and temperature T_e of the extract air in °C. Furthermore, the density ρ in kg/m² and c_P the specific heat of air in kJ/kg/K at constant pressure. A linear approximation for the temperature gradient s in K/m of the stratified air layers over the room height as follows [3]:

$$s = \frac{(1 - \kappa) \cdot (T_{\mathrm{e}} - T_{\mathrm{s}})}{h} \tag{4}$$

with the ceiling height h in m. For usual displacement ventilation systems, a minimum volume flow rate can be calculated in order to remove the cooling load $\dot{Q}_c$ in W. The equation applies only for extract air outlets at ceiling height, as for the temperature stratification [2]:

$$\dot{Q}_{\mathrm{c}} = \dot{Q}_{\mathrm{c}} \cdot \kappa + \dot{q}_{\mathrm{s}} \cdot \rho \cdot c_{\mathrm{p}} \cdot s \cdot h \tag{5}$$

$$\frac{\dot{q}_{\mathrm{s}} \cdot \rho \cdot c_{\mathrm{p}}}{A} = \frac{\dot{Q}_c}{A \cdot s \cdot h} - \frac{1}{\left(\frac{1}{\alpha_{\mathrm{r}}} + \frac{1}{\alpha_{\mathrm{cf}}}\right)} \tag{6}$$

For total temperature difference between supply and extract air (T_e - T_s) applies the following:

$$(T_{\mathrm{e}} - T_{\mathrm{s}}) = \frac{\dot{Q}_{\mathrm{c}}}{\dot{q}_{\mathrm{s}} \cdot \rho \cdot c_{\mathrm{p}}} \tag{7}$$

The temperature $T_{1.1}$ in °C of room air at 1.1m height is often referenced as room air temperature for thermal comfort .Hence, the temperature difference between supply air and of room air at 1.1m height $\Delta T_{1.1}$ in K is of interest [2]:

$$T_{1.1} - T_s = \Delta T_{1,1} = \kappa \cdot (T_e - T_s) + 1{,}1 \cdot s \qquad (8)$$

3. Test room

The test room built inside a large experimental hall, with a displacement ventilation system in place at a constant air temperature of about 22-22.5°C, which is approximately consistent with the average temperatures inside the test room. The geometrical data of the test room is A = 5.2m x 4.3m = 22.4m² floor area and h = 2.45m ceiling height, hence a room air volume of 55.88m³. The testroom is thermally insulated from its surroundings as follows. Below room level (airtight raised floor 0.4m) the supply air duct system is positioned. On one long side of the room, ceiling-high glass windows confine it. This enables good view of the inside of the room from the experimental hall. All other walls and the ceiling are well thermally insulated. Inside the test room, furniture and heat sources are positioned according to the experimental setup (see Fig. 2). Heat sources comprise office equipment, cylindric heat sources equipped with light bulbs and heating foils placed on the floor, representing solar heat gains (see Table 1). The position of the heating foils is varied (see Fig. 2). Surface temperature of the heating foil is 41°C to 44°C at all times.

Table 1: Overview heat sources inside test room

Description	**Number**	$\dot{Q}_{total}$ **in W**
Cylinders (diameter 0,3m, height 1m)	4	320
Desktop PC	2	160
PC Monitor	2	60
Heating foil, floor (total 2,625m²)	4	595
Desk light	1	42
Measuring equipment air velocity	16	56
Total (specific) room heat load $\dot{Q}_{total}/A$	1233W (54.9W/m²)	

Air is supplied using two low-velocity diffusers (*Caverion Q-R*, W x H x D: 2m x 0,3m x 0,25m) which are positioned along one side of the test room. The HVAC device to supply air for the test room is positioned in the basement below the experimental hall. Air temperatures, pressure and volume flow rate of the HVAC device are measured. To control the unsteady supply air flow, the fan of the HVAC device is regulated with an electronic frequency inverter. This is not an ideal way to create the unsteady supply air flow, as the temperature of the supply air inside the duct system

increases when the air flow is paused. Hence, there are limits for the lowest possible supply air temperature and duration t_{Puls} of the unsteady supply scenario.

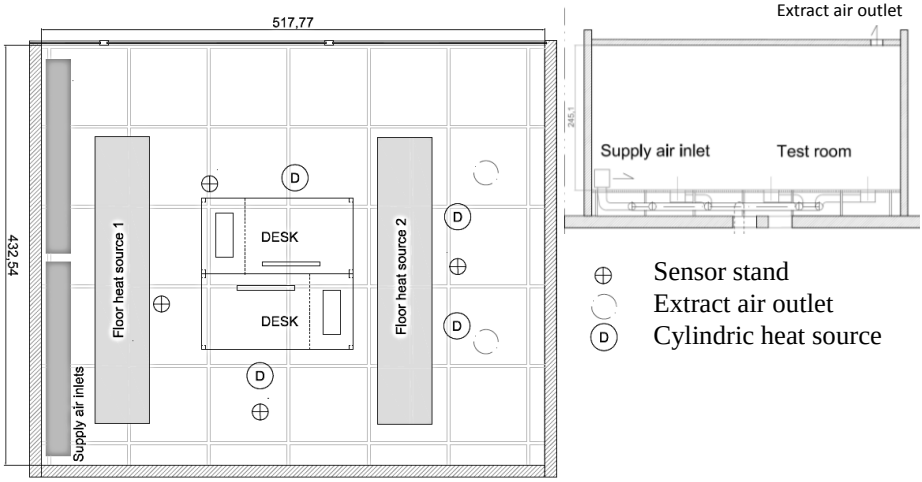

Fig. 2: Layout and section of the test room – positions of furniture, heat sources, sensor stands B1 to B4, supply air inlets and extract air outlets and raised floor. Varying positions of the floor heat source 1 and 2.

4. Test procedure

Air temperatures are measured with calibrated type K thermocouples (TEK) showing a uncertainty of max. +/-0,14K.. Air velocity sensors are Dantec omnidirectional anemometers (DS) with a measuring range 0-1m/s showing an uncertainty of max. +/- 0,02m/s, +/- 2% respectively. Sensors are positioned according to Table 2. At positions B1 and B2 stands with a height of 2m, and at B3 and B4 stands with a height of 0,6m are placed to measure temperature and velocity profiles of the room air.

Table 2: Overview measuring positions: B1 to B4 and extra sensors

Position	Sensor types	Sensor heights in m
B1	Velocity, air temperature	0,1/ 0,25/ 0,4 /0,6 /1,1 /1,7
B2	Velocity, air temperature	0,1/ 0,25/ 0,4 /0,6 /1,1 /1,7
B3	Velocity, air temperature	0,04 / 0,4
B4	Velocity, air temperature	0,04 / 0,4
At ceiling	Air temperature	2,35
Supply air	Air temperature	-
Extract air	Air temperature	2,45
HVAC	Diverse	-

Settling time to ensure “stationary” conditions of room air flow and temperature gradient in the test room is 5 to 10 periodic times before an experiment. All

experiments are carried out for at least 5 to 10 periodic times, maximum duration at 60 periodic times.

5. Test cases

The experiments assess varying thermal loads regarding their distribution in the test room. Unsteady volume flow rates of supply air and the corresponding average, steady flow rates are compared. Furthermore, fog experiments are conducted to visualize the room air flow in the test room.

Table 3: Overview test cases experimental assessment, room heat load $\dot{Q}_{total}$ is 1573 W (of which 595W of heating foils on floor)

Case No	Description cases a, b, c	t_P in s	t_{Puls} in s	t_{Paus} in s	T_s in °C	$\bar{\dot{q}}_{mean}$ in m³/h
#1a	Floor heat load position 1 (1m from inlet), supply air temperature 18°C	50	22.5	27.5	18.5	193.8
#1b		100	45	55	17.6	203.8
#1c		150	67.5	82.5	17.6	205.7
#2a	Floor heat load position 1 (1m from inlet), supply air temperature 16°C	50	25	25	15.9	225.2
#2b		100	50	50	15.8	226.0
#2c		150	75	75	15.7	225.2
#3a	Floor heat load position 2 (3.5m from inlet), supply air temperature 18°C	50	22.5	27.5	17.8	197.5
#3b		100	45	55	17.6	207.6
#3c		150	67.5	82.5	17.3	209.0

6. Experimental results

Fog experiments visualize the distinct room air flow pattern resulting from unsteady volume flow rates of supply air (see Fig. 1). During the time of an air pulse, the supply air enters into the room. Due to the undertemperature, it falls down to the floor in front of the air inlet which is clearly visible at the beginning of the experiment. It then accelerates along the floor as it warms up due to induction and mixing with room air. As the supply air flow interrupts, the preceding air pulse spreads out compatible to a wave in water, gaining thickness. The bottom layer of cold room air behind the air pulse is lower in height and moves at a distinctly slower speed than the pulsed air. After a while, air layers up to approximately 0.8m in the lower part of the room air are moving stimulated by the pulsating supply flow rate. The top layers of fog that are visible periodically change direction of movement away from and back towards the supply air inlet. The motion of moving, lower layers of air depend on the volume flow rate and the periodic time t_P of supply air flow consisting of $t_P = t_{Puls} + t_{Paus}$. The upper parts of the room air on the other hand, are more stationary without visible movement at any of the plumes e.g. above cylindric heat sources. The convective drafts above the heating foils placed on the floor are affected, also showing instability and periodic motion up to approximately 0.3m. During fog experiments at steady supply air flow rates, plumes from all heat sources are more stable.

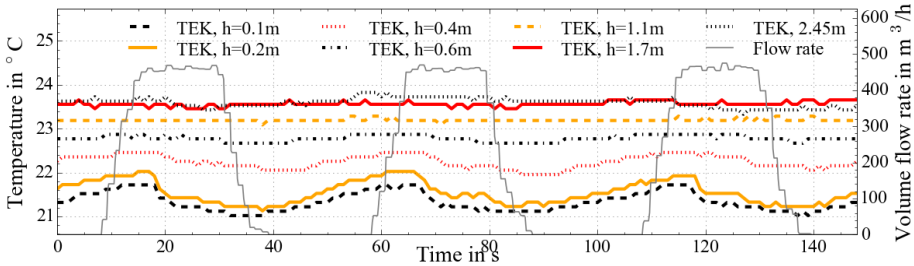

Fig. 3: Case #1a - temperature measurements at sensor stand B2 at 1m from supply air diffusers, during three periods of supply air flow

Fig. 3 shows temperature measurements of case #1a on all sensors at position B2 and the corresponding volume flow rate to indicate the times of supply air pulses into the room. Temperatures measurements up to a room height of 0.6m fluctuate periodically, with the peak values at the beginning of each supply air pulse at the bottom sensor and lagging farther as height increases.The temperature measurements of the sensors in 0.1m (T_{af}) and 0.2m range between 21°C to 22°C, the lower sensor showing always around 0.2K lower temperature. Range of fluctuation at the sensor in 0.4m height is 0.6K with a mean value of 22.0°C respectively. At a height of 0.6m, values of temperatures only fluctuate between 22.6°C and 23°C. Finally, temperature measured at the sensor at 1.1m does not show fluctuations at a nearly constant mean value of 23.4 °C. Respectively, temperature values at 1.7m and at the ceiling are also constant, both approximately 23.6°C. The total duration of the experiment was 60 periodic times, all temperature values showed an overall slight increase of approximately 0.2K during the course of the experiment.

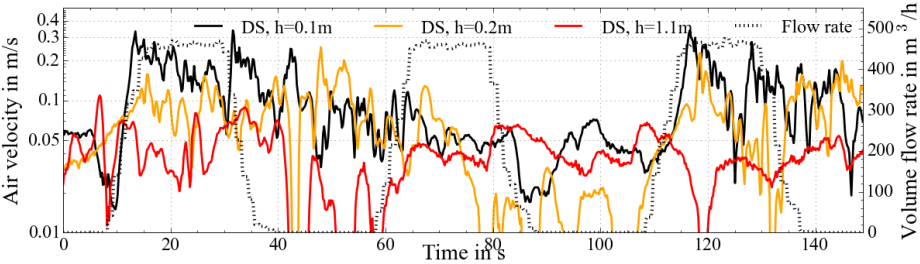

Fig. 4: Case #1a – air velocity measurements at sensor stand B2 at 1m from supply air diffusers, during three periods of supply air flow

In Fig. 4, the corresponding air velocity measurements of sensors in selected heights at position B2 are shown, as well as the volume flow rate. Fluctuations due to unsteady supply air coincide at both two lowest sensors at 0.1m and 0.2m with approximately every second supply air pulse. The peak velocities at the two lowest sensors occur during one pulse, and also during the duration of one subsequent pause. At the time of the next pulse, measured velocities are minimal for both lowest sensors.

Values of velocities fluctuate strongly at a height of 0.1m, with a maximum of 0.374m/s at a mean value of 0.10m/s. Velocity fluctuations at a height of 0.2m show a mean value of only 0.056m/s and corresponding lower peak values. At 1.1m height, the values for air velocity exceed 0.1m/s only twice during the course of the experiment for a couple of seconds each time. The rest of the time, measurements at 1.1m height show fluctuation of air velocity around the mean value of 0.048m/s with a range of approximately +/-0.04m/s. Minimum values at 1.1m height are measured at times, when the lowest sensors at 0.1m and 0.2m show peak velocities.

In Table 4, all test cases are compared regarding temperature conditions in the test room during time of supply air pulses to pauses. All temperature values are averaged using measurements of sensors at same height on stands B1 to B4 with exception of those directly influenced by the heating foils.

Table 4: Comparison of temperature conditions during pulse and pause of supply air for all cases #1 to #3

Pulse								Difference Pulse-Pause	
Case No	$\dot{q}_{Puls}$ in m³/h	T_{af} in °C	T_e - T_s in K	T_e-T_{af} in K	$\Delta T_{1.1}$ in K	κ -	s in K/m	ΔT_{af} in K	$\Delta T_{1.1}$ in K
#1a	339.8	21.66	5.04	1.91	4.83	0.62	0.78	-0.05	0.50
#1b	408.1	21.80	6.84	2.62	6.59	0.62	1.07	0.00	1.05
#1c	424.8	22.05	7.25	2.83	7.04	0.61	1.16	0.10	1.41
#2a	361.0	21.60	9.42	3.76	9.18	0.60	1.53	-0.15	0.82
#2b	413.5	21.55	9.52	3.78	9.29	0.60	1.54	0.01	1.41
#2c	417.8	21.42	9.57	3.87	9.37	0.60	1.58	0.14	1.76
#3a	334.9	21.75	8.08	4.11	7.66	0.49	1.68	-0.15	0.41
#3b	409.3	21.41	8.41	4.56	7.91	0.46	1.86	0.35	0.79
#3c	423.8	21.20	8.79	4.85	8.40	0.45	1.98	0.40	0.94

Temperatures T_{af} range from 21.2°C (#3c) to 22.1°C (#1b) for all cases. Temperature differences (T_e - T_s) between extract and supply air during supply air pulses show a broader range, for mean supply air temperatures during time of pulses see Table 3. Cases #1 with supply air temperatures around 18°C and the floor heat load near the supply inlet show the most narrow range from 5.0K for #1a increasing to 7.25K for case #1c. Highest differences are measured for cases #2 and average supply air temperatures of 16°C. In all cases #3, (T_e - T_s) is also higher than for #1. The reason for this is the temperature at the extract air outlet for #3. Value of T_e is 2.3K higher in case #3a, 1.4K higher in case #3b and 1.2K higher in case #3f. This is due to the position of the heating foils close to the extract air outlets. Analogeous, measured temperatures T_e in cases #2 are also 1.8K to 0.4K higher than for #1, in spite of lower values of T_s and higher mean volume flow rates. Reason for this is not clear. For all cases #1 to #3, (T_e -

T_s) increases with the duration of periodic time. Corresponding to the higher values of T_e in cases #2 and #3 are the higher differences in temperature (T_e-T_{af}). $\Delta T_{1.1}$ shows differences of average measured values of temperature sensors in 1.1m height to to extract air temperature, which are highest at over 9K in cases #2 due to the lowest values of supply air temperatures. When comparing cases #1 to #3 with same periodic times, one can see that the value of $\Delta T_{1.1}$ is from 2.7K for case #3a to 1.4K for case #3c higher than in respective cases #1. For all cases #1 to #3, values of $\Delta T_{1.1}$ increase with the periodic time. Evaluation of κ shows values of 0.6 in cases #1 and #2. For cases #3, values of κ range from 0.45 to 0.49, decreasing for longer periodic time. The reason is again the higher temperature of the extract air in cases #3. Analogeous, the values for the average temperature gradient s in the room are lowest in cases #1, highest in cases #3 and increasing with the periodic time for all cases. When comparing to temperature conditions during time of pause in supply air flow, Table 4 shows that for cases #1 and #2, values of temperatures T_{af} at floor are not more than 0.1K higher or lower than when there is supply air flow. For case #3b and #3c, T_{af} gets 0.4K warmer when supply air flow is interrupted. $\Delta T_{1.1}$, values are between 0.4k and 1.8K higher during pauses when there is no supply air flow. The values increase with longer periodic times, as there is more time during pauses for air temperatures at the supply air diffuser to regenerate.

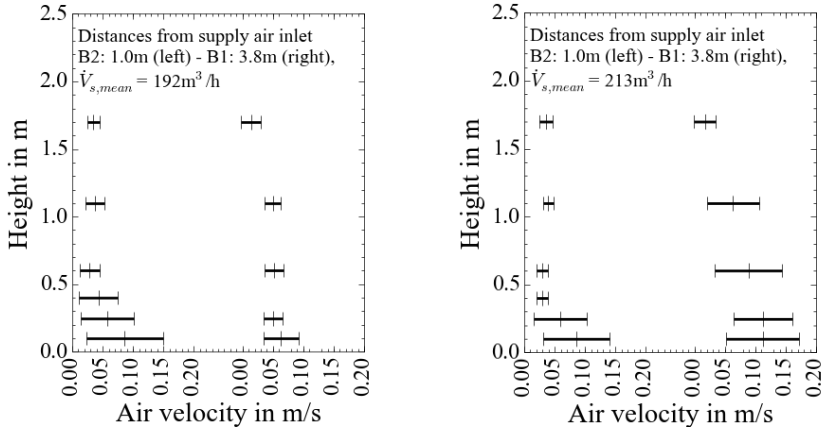

Fig. 5: Cases #1c and #3c – Profiles over room height of mean air velocities with standard deviation during three periods of supply air flow, variation of position of floor heating mats

Fig. 5 shows profiles over room height of mean air velocities with standard deviations of cases #1c and #3c. Because intensity of turbulence of room air is defined as the ratio of standard deviation to mean value of air velocity [6], the diagrams indicate intensity of turbulence as defined for steady flow rates. Both cases result in similar profiles at lowest sensor positions for position B2 (each left) close to the supply air

inlet. Mean values of air velocities are 0.09m/s at 0.1m and 0.07m/s at 0.2m respectively. In both cases, standard deviations are approximately 0.06m/s at 0.1m and 0.05m/s at 0.2m sensor height For all higher sensors at position B2, mean values remain around 0.04m/s to 0.05m/s in both cases, yet values of standard deviation are twice as high in case #1c. At sensor stand on position B1 (3.5m from inlet) on the other hand, in case #3c mean values of air velocities are approximately 0.11m/s at heights 0.1m and 0.2m, whereas only half of the values are measured in case #1c. Velocities degenerate in both cases over room height, yet in case #3c standard deviations remain high up to 1.1m sensor height indicating more than 50% intensity of turbulence.

7. Conclusions

Unsteady displacement ventilation which is constituted in fluctuating supply flow rates is studied for similar ratios of pulse to pause in volume flow. Experimental setup comprises a test room with typical office layout. Based on CFD test cases, different supply air temperatures are applied, and varying distributions of specific thermal load of approximately 55W/m² are examined. Measurements and first interpretation of temporal and spacial distributions of air temperatures and velocities are presented.

8. Outlook on thermal comfort

Extensive further experiments are carried out, specifically including acquisition of test persons' responses to thermal questionairs in order to gain insight into thermal comfort requirements under unsteady room climate conditions. Specifically the tolerance to short air pulses with velocities up to 0.5m/s in relation to duration of pulse and pause need examination, as well as thermal sensation of fluctuating temperatures in limited ranges up to 4 K.

Ackknowledgements

This study was conducted during the research project "IQ Lüftung" which is funded by the German Federal Ministry of Economic Affairs and Energy, funding code IGF 18498 N.

References

[1] Mathisen, H.M.; Nielsen, P. V.; Moser, A. "Displacement Ventilation in Non-Industrial Premises (REHVA Guidebook No 1)". 2007. Mundt, Elisabeth (Editor)

[2] Mundt, E. "The performance of displacement ventilation systems: experimental and theoretical studies". PhD Thesis. 1996. Kungliga Tekniska Högskolan

[3] Mundt, E. "Displacement ventilation systems—Convection flows and temperature gradients". Building and Environment. Volume 30, Issue 1, January 1995, pp.129–133

[4] Mundt, E.. "Contamination distribution in displacement ventilation—influence of disturbances". Building and Environment. Volume 29, Issue 3, July 1994, pp. 311-317, Spec. I. Papers from Indoor Air '93

[5] Mundt, E.. "Convection Flows above Common Heat Sources in Rooms with Displacement ventilation". Proceedings Roomvent '90, Oslo, 1990

[6] Willems, W.M., Schild, K., Dinter, S. "Handbuch der Bauphysik Teil 1". Friedr. Vieweg & Sohn Verlag. 1. Auflage 2006. Wiesbaden.