Thermal environment in simulated office rooms generated by active ceiling diffuser with radiant panels

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Abstract. Ventilation airflow rates are typically controlled based on occupancy, air quality and heat load levels in office rooms with variable air volume (VAV) system. Additional water-based cooling is often the most energy efficient to use when air cooling does not cover heat gains. This can be done by air-water system, by combining ceiling diffusers with radiant panels. The operation can be challenging especially with non-uniform heat loads. In the earlier study, it was concluded that active ceiling diffusers were able to generate a more uniform thermal environment than static ceiling diffusers. The thrown pattern is not constant with the static diffuser, but with the active diffuser, it is more uniform. This study was continued by analysing differences between the design of all-air and air-water systems with active diffusers and investigating higher heat load situations enabled by air-water system. The novelty of this research is to confirm the usability of the air-water system with active ceiling diffusers in an office environment. Office room situations were measured earlier in a full-scale test room with partial occupancy for studying differences between air distribution. Now that was done with CFD simulations. The same 3-person office room case without and with radiant panels was modelled first to validate CFD-simulation in all-air system case, and then to simulate the performance of the design with air-water system. Then 10-person meeting room case with air-water system was simulated with a higher, design heat load level. The thermal environment in the same office room cases measured in the full-scale test and modelled in the CFD simulation was near to each other. Supply air diffuser was modelled in CFD simulation with detailed geometry and other boundary conditions were similar to in full-scale test situation. RANS simulation method was used with SST turbulence model and with a fine computational grid. CFD simulations with higher heat load levels brought new findings for the air distribution with radiant panels. The increase of heat loads also increased room air velocities, but still local thermal environment remained at a good level. This confirms the usability of air-water system with active ceiling diffusers in variable conditions.

Keywords. Office, Thermal environment, Air distribution, Radiant panel
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1. Introduction

Ventilation airflow rates are typically controlled based on occupancy, air quality and heat load levels in office rooms with variable air volume (VAV) system. Additional water-based cooling is often the most energy efficient to use when cooling by cold ventilation air for occupancy and indoor air quality does not cover heat gains. This can be done by air-water system, by combining ceiling diffusers with radiant panels. The target of operation of these room units is to maintain a good thermal environment for occupants and stable supply air distribution in varying occupancy/heat load levels. This can be challenging especially with non-uniform heat loads. In the earlier study, it was concluded that active ceiling diffusers were able to generate a more uniform thermal environment than static ceiling diffusers [1]. The thrown pattern is not constant with the static diffuser, but with the active diffuser, it is more uniform due to constant supply air velocity. This study was continued by analysing differences between the design of all-air and air-water system with active diffusers and investigating higher heat load situations enabled by air-water system. The novelty of this research is to confirm the usability of the air-water system with active ceiling diffusers in the office environment.

2. Method

Office room cases were measured earlier in a full-scale test room with partial occupancy for studying differences in air distribution with different ceiling diffusers. Now that was done with CFD simulations in the cases with active ceiling diffusers without and with radiant panels. With CFD simulations more detailed view of the performance was possible to obtain. The geometry of the same measured and simulated test room setups is shown in Fig. 1. The dimensions of the test room were 6.1 m × 4.4 m x 2.7 m (L x W x H), with a floor area of 26.6 m^2. The operating principle of the active diffuser (595x595 mm attached to the suspended ceiling) is based on two air adjustment dampers: Fig. 2-1 A) adjustment for the constant static pressure level of the duct and B) active adjustment of supply airflow rate with linear movement of adjustment blade. The plenum of the active diffuser is sound attenuated so additional sound attenuator or VAV damper are not needed. Diffusers were selected with the manufacturer's design tool [2] for sound pressure level maximum 30 dB(A) with suitable supply air jet throw length for having a realistic design case. Additional 4 radiant panels (2995x595 mm attached to the suspended ceiling) are shown in Fig. 2-2 and Fig. 8.

Studied heat load levels and supply airflow rates are listed in Tab. 1 and 2. The same measured 3-person office room case (46 W/m^2floor) without and with radiant panels was modelled first to validate CFD-simulation in all-air system case O3AA (4.3 $l/s,m^2$floor), and then to simulate the performance of the design in the air-water system case O3AW (1.8 $l/s,m^2$floor). Then 10-person meeting room case with air-water system M10AW (4.3 $l/s,m^2$floor) was simulated with CFD in the higher, design heat load level (81 W/m^2floor) situation.

The thermal environment was measured with calibrated air velocity, turbulence and temperature sensors in steady-state conditions. The measurement grid is shown in Fig. 3. Sensors were at heights 0.1, 0.6, 1.1, 1.7, 2.1 and 2.5 m from the floor (total 270 measurement points). Sensors included in the occupied zone started from 0.5 m distance from walls and up to 1.7 m height from floor (total 100 measurement points). Velocity and turbulent intensity were measured with omni-directional hot sphere anemometers HT412 (accuracy +/- 0.02 m/s and +/- 1 % of readings with velocities 0.05 - 1.0m/s), and temperature with PT100 sensors (accuracy +/- 0.2 °C). Readings were three minutes average values.

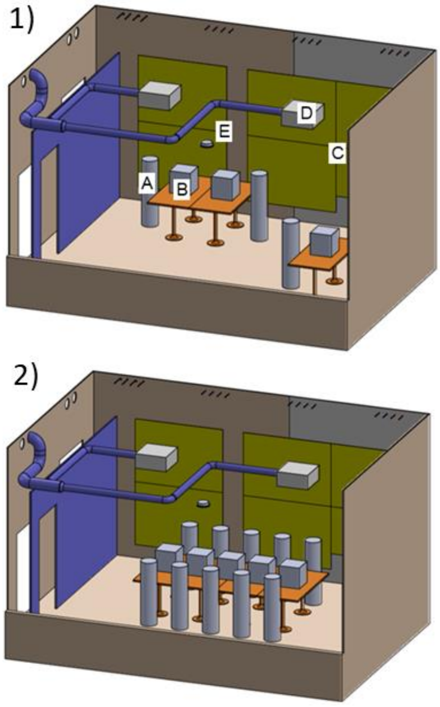


Fig. 1 - Geometry of the measured and simulated test room setups. Heat loads presented in office room setup: A) occupants, B) computers and C) simulated warm windows. Also supply air ductwork, D) diffusers and E) exhaust air valve.

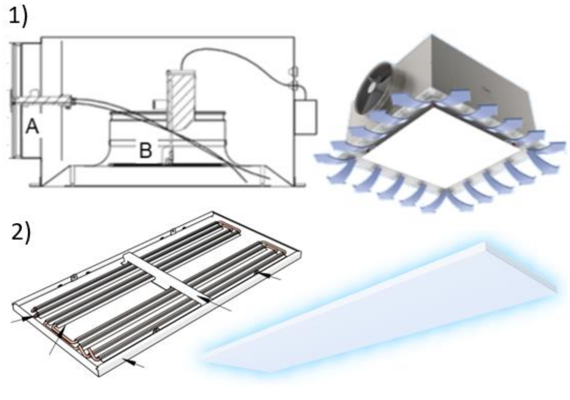


Fig. 2 – 1) Active radial ceiling diffusers and 2) radiant panels used for supply air distribution and cooling.

CFD simulations were done with Ansys CFX 2021 R2. An unstructured computational grid was constructed of 7.5 – 8.2 million elements (1.4 – 1.5 million nodes). Finer grid resolution was used near the supply air diffusers. Grid element edge length varied from 0.0015 m to 0.2 m with a growth rate of 1.2. Visualization of the grid is shown in Fig. 4. Supply air diffusers were modelled in CFD simulation cases with detailed geometry. Boundary conditions were similar to the full-scale test situation. Heat gains were modelled as wall heat fluxes based on the electrical power or manufacturer's cooling capacity data for radiant panels. The steady-state RANS simulation method was used. Turbulence was modelled by using the SST model with automatic wall treatment. Buoyancy was modelled by using air ideal gas with gravity. Radiation was modelled with the discrete transfer model. Cases were solved with high numerical resolution (2nd order with blend factors). Validation results will be published also separately with grid independence study [3].

Tab. 1 - Heat gains in measured and simulated cases.

Case	O3AA	O3AW	M10AW
Room type	Office	Office	meeting
Occupants	225 W	225 W	750 W
Computers	300 W	300 W	750 W
Lighting	112 W	112 W	112 W
Window solar load	600 W	600 W	549 W
Tot. gains	1237 W	1237 W	2161 W
Tot. gains / floor area	46 W/m^2	46 W/m^2	81 W/m^2

Tab. 2 – HVAC system operating data in measured and simulated cases.

Case	O3AA	O3AW	M10AW
HVAC system	All-air	Air-water	Air-water
Supply airflow	115 l/s	48 l/s	115 l/s
Supply airfl. / floor area	4.3 l/s,m^2	1.8 l/s,m^2	4.3 l/s,m^2
Fully-mixed CO_2 predict.	545 ppm	747 ppm	883 ppm
Design room temperature	25 °C	25 °C	25 °C
Supply air temperature	16 °C	16 °C	16 °C
Supply air cooling power	1242 W	518 W	1242 W
Radiant panel cooling power	0 W	719 W	919 W
Radiant panel cooling power / panel area	0 W/m^2	67 W/m^2	85 W/m^2
Total cooling power	1242 W	1237 W	2161 W

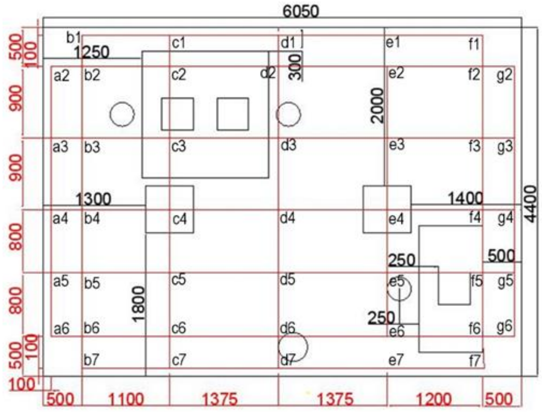


Fig. 3 - Measurement pole locations in office and meeting room cases: marked with letter a..g and number 1..7.

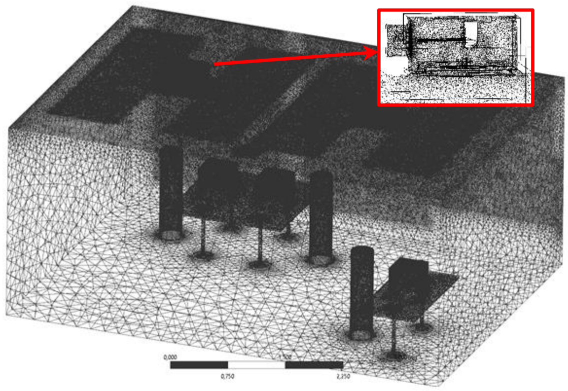


Fig. 4 – Visualization of the computational grid used in simulated cases with diffuser model grid magnified

3. Results

Measured air velocities and temperatures in all 270 measured locations are presented and compared to CFD simulation results in Fig. 5. This comparison was done by collecting data from the same locations and sorting it in ascending sequence. The thermal environment in the same office room cases measured in the full-scale test and modelled in the CFD simulation was near to each other. That analysis does not take into account if that specific velocity or temperature level occurred exactly at the same location. It can be seen that in the CFD results the velocity and temperature ranges were a bit wider. Still, overall trends were very near to each other. The average difference of absolute values between measured and CFD simulated velocity in the curve is 0.02 m/s and in the same location 0.05 m/s. The average difference of absolute values of temperatures in the curve is 0.13 °C and in the same location 0.34 °C.

Air temperature, velocity and draught rate distribution in the occupied zone are shown in Fig. 6. The occupied zone was defined by including values from 0.5 m from walls. Pole locations e5 and d5 were excluded in the meeting room case (overlapping with occupants), and only points at 0.1 m were included from pole locations c4, d4 and e4 (meeting room table was located there). These graphs show both measured and simulated cases. The highest draught rate levels were a bit underestimated in CFD simulation when compared to measurement results. The thermal environment in the case O3AW was slightly better than in the case O3AA (same heat load situations). Air velocity level was a bit lower, temperature more uniform and draught rate level lower. A clear increase in the air velocity level was seen in the M10AW case when compared to office room cases. The air temperature distribution was now similar to the O3AA case. The draught rate level was also clearly increased. Still, the draught rate in the occupied zone stayed below 20%, which indicated that local thermal comfort was at a good level (even if maximum measured draught rate levels would have been slightly higher like seen in O3AA curves).

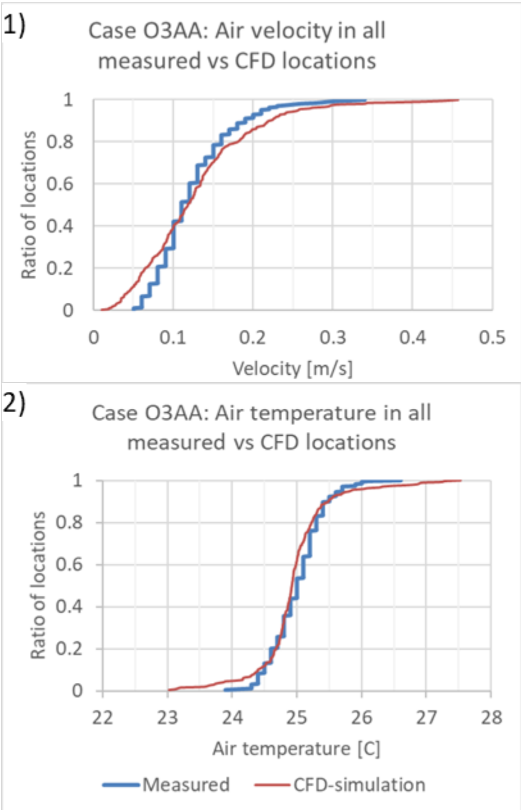


Fig. 5 - Measured and CFD simulated 1) air velocities and 2) temperatures in all measured locations.

Measured and simulated draught rate distributions on a cross-section plane in the middle of the room parallel to the simulated window wall are shown in Fig. 7. Scales in the measured and simulated distributions differ slightly, but colours are at similar levels for comparison. According to this, the CFD simulations predicted a similar flow field than in the measured case O3AA. Slightly smaller draught rate levels were visible in the case O3AW as seen earlier from the velocity distribution curves (Fig. 6). Clearly higher draught rate levels in the case M10AW were seen and that can be compared with case O3AA where the same supply airflow rate was used, but with a lower heat load level and without radiant panels.

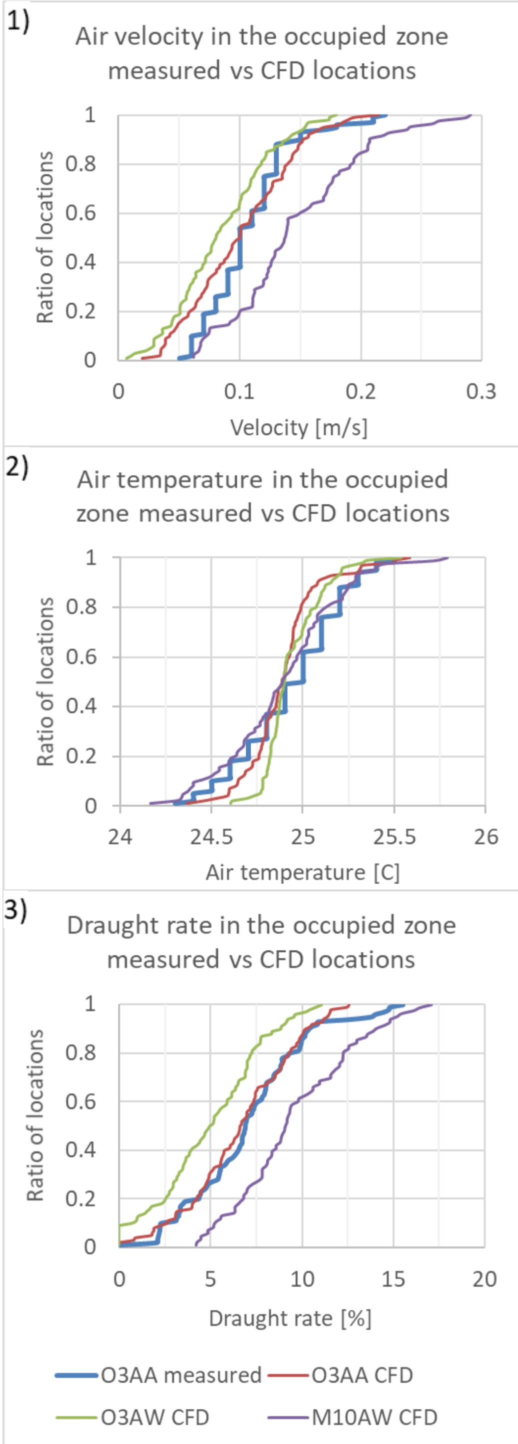


Fig. 6 - Measured and CFD simulated 1) air velocities, 2) temperatures and 3) draught rate levels in measured locations of the occupied zone.

Room air velocities over 0.20 m/s in CFD simulation cases are visualized in Fig. 8. There can be seen higher velocity level in the room in cases with higher supply airflow rate. This was especially clear in case M10AW with higher heat load level (with same supply airflow rate than in case O3AA) as discussed earlier. Even if an active ceiling diffuser provided a constant supply air velocity level at different VAV situations. It should be noted that the momentum of the supply air jet was higher with a higher supply airflow rate. This can be beneficial for uniform air distribution also in the cases with higher heat load levels and thus stronger convection flows in the room.

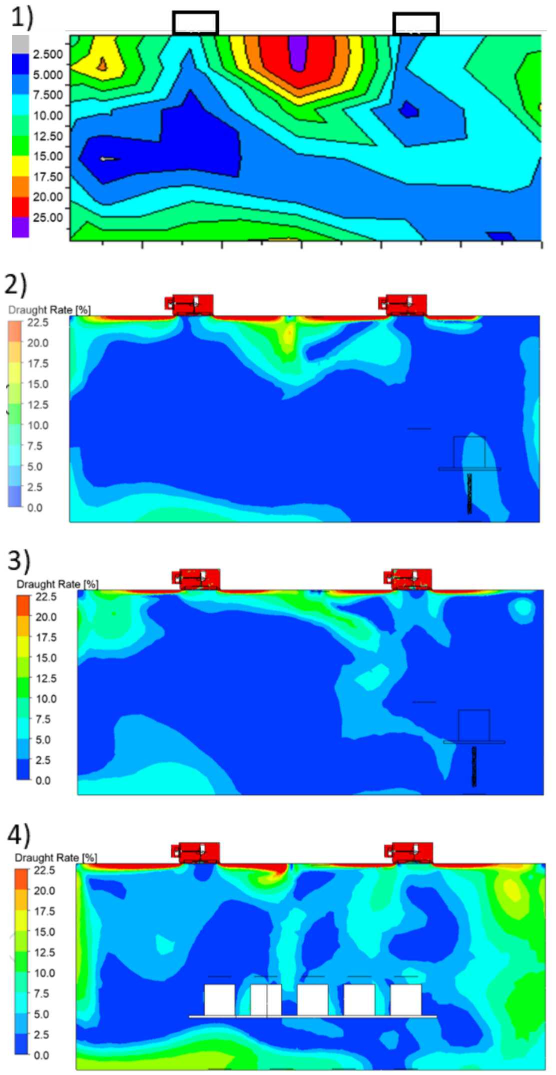
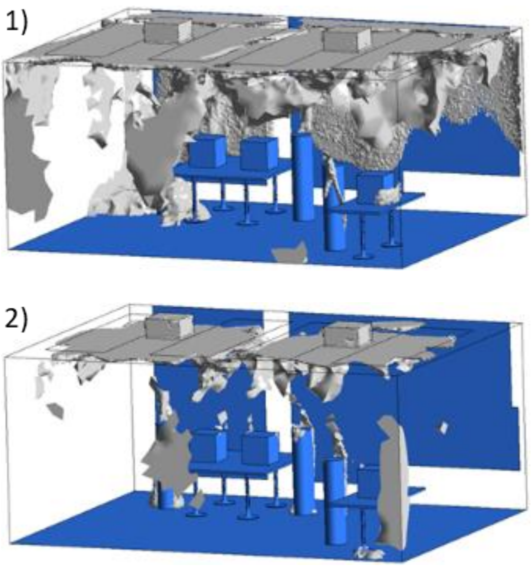


Fig. 7 –Draught rate distribution 1) in measured case O3AA and in CFD simulation cases 2) O3AA, 3) O3AW and 4) M10AW



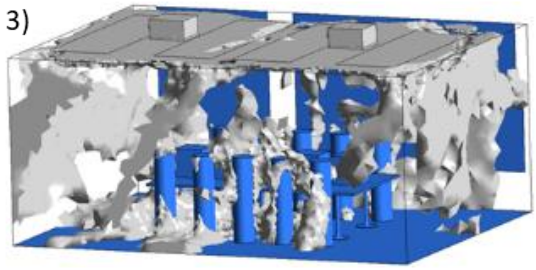


Fig. 8 – Room air velocity over 0.20 m/s in simulated cases 1) O3AA, 2) O3AW and 3) M10AW

4. Conclusions

The thermal environment in the same office room cases measured in the full-scale test and modelled in the CFD simulation was near to each other. This was based on the comparison of measured air velocities and temperatures in 270 measured locations with CFD simulation results.

The thermal environment in the same office room case with ceiling diffusers and radiant panels for cooling was slightly better than cooling with only supply air distribution. Room air velocity level was a bit lower, temperature more uniform and draught rate level lower.

A clear increase in the room air velocity level was seen in the meeting room case with high heat loads when compared to office room cases. The draught rate level was also clearly increased. Still, the draught rate in the occupied zone stayed below 20%, which indicated that local thermal comfort was at a good level also in the high heat load case. This confirms the usability of air-water system with active ceiling diffusers and radiant panels for providing a good thermal environment in variable conditions.

5. Acknowledgement

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6. References

[1] Wu Y., Mustakallio P., Kosonen R., Kaukola T., Chen J., Liu H., Li B. Experimental study of five different VAV air terminal devices under variable heat gain conditions in simulated office and meeting rooms. Building and Environment 2022;209:108641.

[2] Halton HIT Design product selection tool, 2019, Halton Oy, Finland.

[3] Kanerva P., Mustakallio P. Comparison of CFD analyses and experiments of active ceiling diffusers. In: Indoor Air 2022 (submitted). 2022 Jun 12-16; Kuopio, Finland.

DATA ACCESS STATEMENT

The datasets generated during and/or analysed during the current study are not available because they contain detailed product drawings of the diffusers, but the authors will make every reasonable effort to publish them in near future.