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Performance of Chilled Beam with Radial Swirl Jet and Diffuse Ceiling Air Supply in Heating Mode

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Abstract
The performance of diffuse ceiling air supply and chilled beam with swirl jet (CSW) in heating mode (winter situation) was studied and compared with regard to the generated indoor environment. An office mock-up with one occupant was simulated in a test room (4.5 x 3.95 x 3.5 m^3 (L x W x H)). A window (6.5 m^2) with cold surface (14 °C) was simulated by radiant panels. Four CSW chilled beam units were installed symmetrically on the suspended ceiling together with two exhaust vents. The diffuse ceiling inlet was made of standard perforated acoustic tiles (0.5% total degree of perforation). The room air temperature was kept at 21 °C. Tracer gas was used to simulate pollution from floor and desk. The experimental conditions comprised: 1) night time without heat sources in the room; the room air conditioning system was used to heat up the room; 2) heat load generated by an occupant (simulated by dressed thermal manikin) and a laptop; 3) heating by convectors positioned under the window (convectors used alone and convector used together with CSW supplying isothermal air for ventilation). The heat distribution provided by the systems was not effective compare to the distribution provided by convector. The tracer gas concentration in the occupied zone was considerably higher than the concentration at the exhaust. Airflow rate considerably higher (2.5–5.9 times higher) than the minimum ventilation rate required in the standards was needed to safeguard the indoor air quality.

Keywords - diffuse ceiling air supply; chilled beams; space heating

1. Introduction

Nowadays, with building legislation demanding sustainable buildings with low energy consumption the building practice has changed and highly well insulated and air tight buildings with heating demand reduced to a minimum are introduced. In such building the heat generated by people, light and equipment is sufficient to maintain a comfortable room temperature in the cold winter months. In fact, most new buildings today have cooling demand in the working hours throughout the whole year.

In the design of low energy buildings, it may be questioned the need for a separate heating system when the demand for space heating only applies for a few hours in the night and in the weekends. In some new buildings space heating has been entrusted to the air distribution system by supplying warm air to the space in the night to maintain the desired room temperature until people show up in the morning [1–2].

Few room air distribution systems are well suited for space heating because the ventilation effectiveness will be reduced which results deterioration of the indoor air quality. Displacement ventilation is not suitable for space heating because the outdoor air is supplied with a low momentum and the air will rise towards the ceiling instead of being distributed along the floor in the occupied zone. Mixing ventilation can with a proper design be used for space heating, but the ventilation effectiveness decreases when the supply temperature increases. If the temperature of the supply air is 0–5 °C higher than the temperature in the breathing zone the ventilation effectiveness, is typically 0.8–0.9, and if the supply temperature is higher than 5 °C the ventilation effectiveness is typically in the range 0.4–0.7 [3]. This implies that that the ventilation rate needs to be increased accordingly in order to obtain the required indoor air quality (ventilation effectiveness of 0.5 implies that the ventilation rate needs to be doubled).

This paper presents results of the performance of chilled beam with swirl radial jet and diffuse ceiling air supply in heating mode. Only part of the collected and analyzed results is presented in this paper.

2. Methods

Experimental Facilities

A full-scale test room (L x W x H = 4.5 m x 3.95 m x 3.5 m) was used to simulate an office with two desks (Figs. 2–3). A window with area of 6.5 m^2 was simulated by radiation panels circulating cold water to achieve the surface temperature of 14 °C. A suspended ceiling system made of acoustic mineral wool tiles was mounted at height 2.7 m. The ceiling of the room was insulated and the temperature of the air surrounding the room was kept equal to that in the room. Four chilled beam units with radial swirl diffuser of type Halton CSW (Fig. 1c) were installed symmetrically and recessed into the suspended ceiling together with two extract vents. The wall and ceiling of the void above the suspended ceiling was carefully sealed with a plastic membrane, and air supply ducts were installed with bends facing upwards to pressurize the void and ensure even distribution of air when the void works as plenum (Fig. 1a). The mineral wool tiles were perforated with tubes with an inner diameter of 13.8 mm (Fig. 1b), constituting a total degree of perforation 0.5%. Two electrical convectors were installed under the window (Fig. 2).

a) b) c)

Figure 1: a) Plenum (ceiling void) of diffuse ceiling air supply. b) Air distribution principle of diffuse ceiling air supply [4]. c) Air distribution principle of chilled beam with swirl jet [5].

An occupant was simulated by a dressed thermal manikin (0.75 clo and 75W) representing an average female body size. A laptop (88 W) was simulated by light bulb installed inside metal box with output control.

Experimental Conditions

Three experiments; one with the chilled beam, with swirl diffuser (CSW) and two with the diffuse ceiling air supply (DC) were performed in a heating mode with conditions as specified in Table 1. The performance of the systems was compared with convectors combined with isothermal air supply from radial diffuser (CSW chilled beam with bottom plates closed to disable induction of room air) and with convectors and no air supply.

Table 1: Experimental conditions.

Case	Air flow rate $[L\cdot s^{-1}\cdot m^{-2}]$	Air supply temp. [°C]	Air extract temp. [°C]	Room temp. [°C]	Supplied heat [W]
DC empty office	2.7	34.6	25.2	21.1	782
CSW empty office	2.1	21.0[1]	24.6	21.1	641
Convector empty office	2.1	21.0	21.0	21.1	300
Convector empty office no air supply	-	-	-	21.2	250
DC office heat load	2.7	28.3	23.7	21.4	399

[1] Supply air temperature from beam nozzles. Air leaving the beam is mixture of supply air and induced air heated by coil with supplied heat.

The window temperature was maintained at 14.0 °C in the experiments corresponding to a fixed heat loss of −364 W (−20 $W\cdot m^{-2}$ floor area).

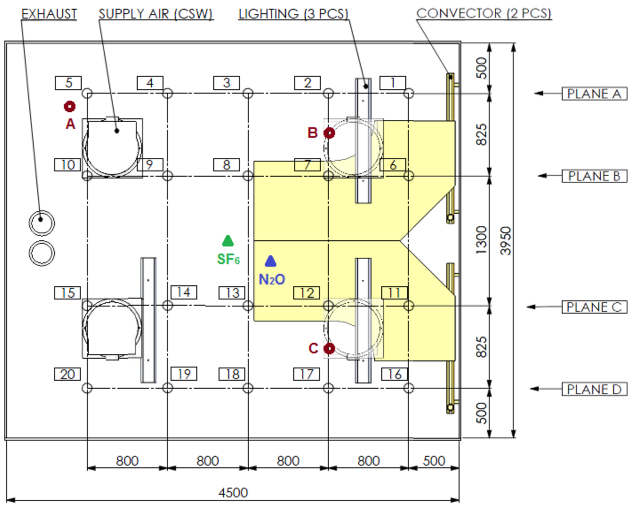

Figure 2: Experimental set up cases "DC/CSW/Convector empty office" and case "Convector empty office no air supply". Mean air speed and mean air temperature measured in locations numbered from 1–20 (locations 6, 7, 11 and 12 excluded). Concentration of tracers gases measured in locations A–C (heights: 0.1, 1.1, 1.7, 2.4 m)

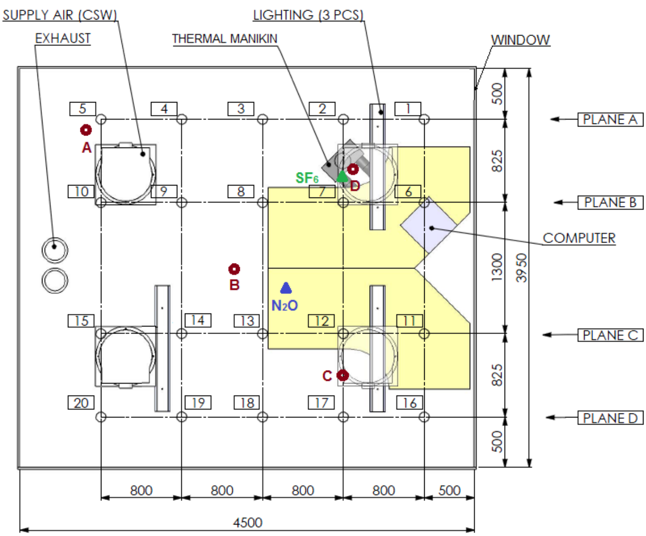

Figure 3: Experimental set up case "DC office heat load". Mean air speed and mean air temperature measured in locations numbered from 1–20. Concentration of tracers gases

measured in locations A–D (heights: A: 0.1, 1.1, 1.7, 2.4 m, B: 0.1, 1.1, 1.7, 2.4 m, C: 1.1, 1.7, 2.4 m, D: 1.1 m).

The target for the room temperature (defined as the average temperature at 1.1 m in 16 locations (Figs. 2–3) in the occupied zone) was 21 °C in all experiments. In the case "CSW empty office" hot water was supplied to the active chilled beams and the room temperature was maintained by adjusting the water temperature. In the cases "DC empty office" and "DC office heat load" the room temperature was maintained by adjusting the air supply temperature. In the case "DC office heat load" the thermal manikin and the laptop simulator were switched on, adding heat of 163 W in the room. In the cases ”Convector empty office” and ”Convector empty office no air supply” the room temperature was maintained by adjusting the power of the convectors.

The conditions of the cases "DC/CSW/Convector empty office" and the case ”Convector empty office no air supply" (Fig. 2) represent an office in winter night time heating situation without heat sources in the room. The condition of the case "DC office heat load" (Fig. 3) represent an office in winter situation with an occupant and a laptop present.

Measuring Procedure

Measurements of air speed and air temperature were conducted in several locations in the occupied zone to evaluate the velocity conditions and the temperature distribution (vertical temperature difference and non-uniformity in the temperature distribution). The air speed and air temperature was measured in six heights (0.1, 0.2, 0.5, 1.1, 1.5, 1.8 m) above the floor level at twenty locations positioned symmetrically in a grid as shown in Figs. 2–3. The measurements were initiated when the room conditions became stabile. Three minutes mean values for the air speed and air temperature, in addition to the turbulence intensity, were obtained for each measurement point. When the measurements in one of the locations (x, y) finished, the stand with the sensors was moved to the next location. A waiting time of three minutes was introduced after the stand was moved to another position before the next series of measurements were initiated.

Smoke was released into the supply to visualize the air flow pattern in the room. Smoke visualization was performed in each of the experiments and was recorded by a digital video camera.

Tracer gas measurements were conducted in the experiments for the purpose of evaluating the ventilation effectiveness of the systems investigated. In cases "DC/CSW/Convector empty office" Sulfur hexafluoride (SF_6) and Nitrous oxide (N_2O) was used to simulate emissions from the flooring and table respectively. In the case "DC office heat load" SF_6 was used to simulate an active source of contamination (combined heat source and contaminant source) – in this case the biofluents emitted from a human being. SF_6 was released from the torso of the simulated occupant. Nitrous oxide (N_2O) was used to simulate emissions from a passive source represented by emissions from a table. The tracer gas was released as a constant dose (0.15 $L \cdot s^{-1}$) through a sponge fastened to the source to prevent high momentum of the released tracer gas flow. The points of release are indicated on Figs. 2–3. The tracer gas concentrations in the room were allowed to reach steady state conditions before the concentration measurements were initiated. Air was sampled through tubes from several points in the room and from the supply and return ducts, and the concentrations of SF_6 and N_2O were monitored. The measurement locations and points are defined in Figs. 2–3. When the concentrations reached stable values, thus steady state conditions, repeated concentration measurements (8 – 20 measurements) were performed to allow for statistical analysis of the results and to calculate the mean in time concentration in each point. The mean values were used to calculate ventilation effectiveness and normalized concentration.

Measuring Equipment

The air speed was measured using a multichannel low Velocity Thermal Anemometer (Measurements System HT-400, manufactured by “Sensor”), consisting of six transducer units HT-426-0 with omnidirectional velocity probes type HT-412. The velocity measurement range of the instrument is 0.05–5 $m \cdot s^{-1}$. The instrument measures speed with accuracy ±0.02 m/s. The characteristics of the anemometer comply with the requirements in the standards. The air temperature was measured using six Pt100 Class A sensors positioned close to each velocity probe. The systems were interfaced a computer for the purpose of data collection.

The air flow rates in the supply and return ducts were measured with two Furness Controls FCO33 (Orifice plate according ISO 5167 with differential pressure transmitter). The total water flow rate to the CSW chilled beams was measured with Krohne Electromagnetic flow meter IFC 010. These two systems were interfaced a computer for the purpose of data collection.

The concentration of tracer gases were monitored by a Brüel & Kjær Photoacoustic Multi Gas Analyzer—Model 1302; repeatability 1% of the measured value. In-home built sampler and dozer with software were used to dose the tracer gases (0.15 $L \cdot s^{-1}$) and to collect the air samples for the photoacoustic multi gas analyzer.

The ventilation system had a built in smoke machine (F-100 Performance fog generator, High End Systems, USA) which was used to release smoke into the supply to visualize the flow pattern from the air terminal device.

3. Results and discussion

The performance of the studied systems, namely diffuse ceiling air supply (DC) and chilled beam with radial swirl diffuser (CSW) in heating mode was compared with space heating by convectors in two cases – with and without isothermal air supply.

The smoke visualization conducted in cases "DC empty office" and "CSW empty office" did not show any specific air flow pattern. The smoke was mainly distributing under the ceiling and was not able to reach further down than 0.3–0.6 m from ceiling level. In case "Convector empty office" smoke visualization showed good mixing (helped by the buoyancy flows). In case "DC heat load" the visualization showed that the smoke was able to reach further down in the occupied zone compared with cases "DC empty office" and "CSW empty office", but not as far down as where the occupant was sitting. Smoke visualization was not conducted in case "Convector empty office" no air supply.

The temperature distribution in the occupied zone in cases "DC empty office" and "CSW empty office" showed weak temperature stratification, with 20–21 °C in the lower parts and 21–22 °C in the upper parts. The air speed distributions showed low air speeds in the range 0–0.1 $m{\cdot}s^{-1}$ in the occupied zone. The temperature distribution in case "Convector empty office" was more uniform and in the range 20.5–21.5 °C in the whole occupied zone. The air speed distribution was in the range 0–0.14 $m{\cdot}s^{-1}$.

The room temperature was maintained at 21 °C in all experiments (Table 1). However, the supplied heat in cases "DC empty office" and "CSW empty office" was respectively 2.5 times and 2 times higher compared with case "Convector empty office" (Table 1). The warm air supply was ineffective in distributing heat to the occupied zone and as seen from the extract temperature in Table 1 a considerably part of the supplied heat was shortcutting directly to the extract.

The supplied heat in case "Convector empty office no air supply" was 50 W lower compared with case "Convector empty office" (with isothermally supplied ventilation air). This can partly be explained by generally lower air speeds in the occupied zone, thus less forced convection at the window surface.

Case "DC office heat load" introduced a heat load of 163 W (heat load from an occupant and a laptop). Still the supplied heat was 33% higher compared with case "Convector empty office" (Table 1). As in case "DC empty office" and "CSW empty office" much of the supplied heat in case "DC office heat load" was shortcutting directly to the extract, as seen from the extract temperature, and the system proved to be ineffective in distributing heat to the occupied zone

The impact of the airflow pattern on the distribution of pollution in the room was assessed by normalized concentration of tracer gas in several heights and positions in the room (normalized concentration was defined as the tracer gas concentration at the measured point divided by tracer gas concentration measured at the exhaust) and by the ventilation effectiveness in the breathing zone of the occupant in case “DC office heat load” (the ventilation effectiveness was defined as tracer gas concentration measured at the exhaust divided by the tracer gas concentration at the measured point). The results for normalized concentrations in location C (Fig. 2–3) in the room is presented in Fig. 4. The results for normalized concentrations in location A and B showed the same tendency as the presented results for location C. The mean normalized concentration in the occupied zone and the ventilation effectiveness in the breathing zone are presented in Table 2.

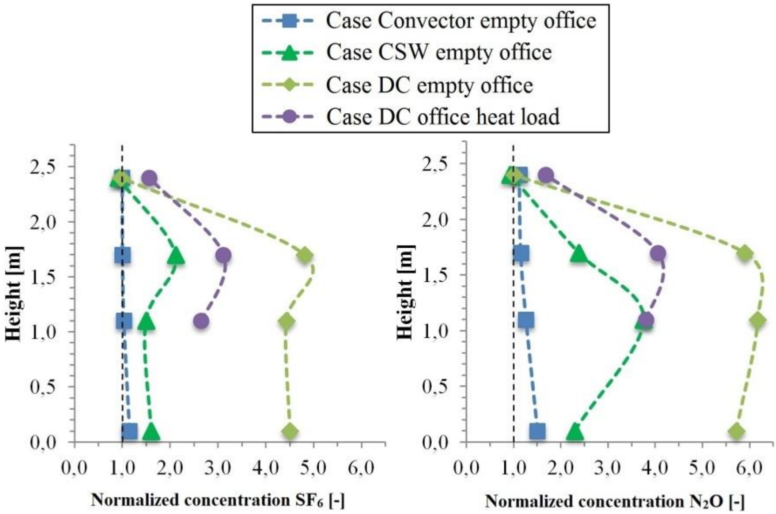

Figure 4: Normalized concentrations in location C. The loaction is indicated in Figs. 2–3. y-axsis: height, x-axis: normalized concentration.

Table 2: Mean normalized concentration of tracer gas in occupied zone and ventilation effectiveness in breathing zone.

Case	Mean normalized concentration [-] SF_6	N_2O	Ventilation effectiveness breathing zone [-] N_2O
DC empty office	5.0	5.9	-
CSW empty office	2.0	2.5	-
Convector empty office	1.2	1.2	-
DC office heat load	3.1	4	0.24

The results in Fig. 4 and Table 2 show that diffuse ceiling air supply and CSW chilled beam did not create complete mixing in the occupied zone. Minimum ventilation rates in the standards assume ventilation effectiveness of 1, i.e. normalized concentrations of 1 (normalized concentration is inverted ventilation effectiveness). In case “DC empty office” the mean normalized concentration of SF_6 and N_2O in the occupied zone was 5 and 5.9 respectively. These results imply that increase of the outdoor air flow rate 5.9 times above the minimum ventilation rate was needed to safeguard the indoor air quality. In case “CSW empty office” the average normalized concentration of N_2O imply that outdoor air flow rate 2.5 times above the minimum ventilation rate was needed to safeguard the indoor air quality. Convectors and isothermal air in case ”Convector empty office” did create good mixing in the occupied zone and the system safeguards the air quality at ventilation rates 20% above minimum rates.

Case “DC office heat load” introduced a few asymmetrical positioned heat sources in the room, namely an occupant and a lap-top. This slightly reduced the mean normalized concentration in the occupied zone compared with case “DC empty office”. However, the ventilation effectiveness in the breathing zone of the occupant was 0.24 and the result imply that increase of the outdoor air flow rate 4.2 times above the minimum ventilation rate was needed to safeguard the indoor air quality.

4. Conclusions

Chilled beam system with radial swirl diffuser (CSW) and diffuse ceiling supply (DC) are ineffective in heat and clean air distribution compared to convectors combined with isothermally supplied ventilation air.

The two systems can create optimum thermal environment in space heating situations however it will require energy consumption much higher than the convector system.

The systems create inefficient mixing in the occupied zone. The outdoor air flow rate needs to be increased up to 2.5 times with the CSW system and

5.9 times with the DC system above the minimum required flow rate in order to safeguard the desired indoor air quality.

Chilled beam system with radial swirl diffuser can create optimum thermal environment when used for heating in situations without heat sources present in the room.

Diffuse ceiling air supply can create optimum thermal environment in space heating situations however it will require energy consumption much higher that convector system.

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