Topic B3: Control of indoor environment

CAN DEMAND CONTROLLED VENTILATION REPLACE SPACE HEATING IN OFFICE BUILDINGS WITH LOW HEATING DEMAND?

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SUMMARY

The indoor climate was evaluated in a cubicle office of the first office building with passive house standard in Norway. The office building (http://miljohuset-gk.no/) is located in Oslo and operational since August 2012. In this building, demand controlled ventilation is used to cover the heating demand by supplying warm ventilation air into the rooms when needed. The purpose of this study is to determine whether this strategy provides a good thermal comfort and ventilation effectiveness, and therefore constitutes a relevant solution for buildings with low heating demand. Measurements were carried out in the cubicle office during the coldest days of winter 2013-2014, and revealed a good indoor climate for a broad range of supplying and heat load conditions.

INTRODUCTION

The stricter requirements in terms of reduction of CO_2 emissions and energy use in the building regulations lead to buildings with highly insulated and airtight envelopes. In this context, the passive house standard, initially developed in Germany, was adapted to countries with colder climate, such as Norway (NS 3701, 2012).
The main idea behind the passive house concept is to reduce the heating demand drastically. At the same time, the increased airtightness makes it even more crucial to provide fresh air to the occupants, in order to avoid adverse effects on well-being, productivity, and health (Wargocki et al., 2000). Replacing space heating with air heating, *i.e.* to supply warm air into the rooms at the hygienic ventilation rate, then appears like an attractive solution.

Figure 1. a) GK environmental building, Oslo. b) View of an active supply air diffuser.

The office building (Fig.1.a) is the first office building with passive house standard in Norway. It features great envelope performances, both in terms of thermal insulation (U-value walls: $0.14\ W/m^2/K$, U-value windows: $0.78\ W/m^2/K$) and airtightness: $0.23\ vol.h^{-1}$ at *50 pa* pressure difference. As a consequence, the measured final energy consumption is reduced to $49\ kWh/m^2$ yearly, and the peak heating load is inferior to $10\ W/m^2$. In addition, active supply air diffusers are used in each room in order to control the ventilation rate according to room demand, and maintain an acceptable indoor temperature and indoor air quality (Fig.1.b).

In this context, the research and development project Forklima (http://www.sintef.no/Projectweb/For-Klima/) is carried out in the building. The aim of this project is to assess whether it is possible to cover the heating demand with warm ventilation air exclusively during the coldest days, without causing thermal discomfort or poor ventilation efficiency. And if applicable, how high supply temperature is acceptable before indoor climate problems occurs.

In order to evaluate the indoor climate, measurements were performed during the coldest days of winter 2013-2014 in an unoccupied cubicle office located on the north façade of the building, and adjacent to occupied open plan offices.

The results from the field measurements are presented in this paper. Questionnaires related to indoor climate in the occupied open plan offices, as well as Computation Fluid Dynamics simulations of the airflow patterns are ongoing and will complete the present study.

METHODOLOGIES

The cubicle office where the measurements were performed is presented on Fig.2. Its dimensions are $4.25\ m * 2.25\ m * 2.70\ m$, corresponding to $9.6\ m^2$ of floor area. A mixing ventilation strategy is employed: the ventilation air is supplied radially into the room through an active air supply diffuser located at the ceiling and exhausted at ceiling level close to the door. The diffuser employed is composed of 6 moving plates which are adjusted in order to control the supplying section and maintain a constant supply velocity for all ventilation rates.

Figure 2. View of the equipped cubicle office.

Indoor climate measurements

The thermal comfort in the room was assessed according to the EN 15251 (2007) standard in terms of operative temperature, thermal stratification, and draught rate. The measurements were performed in the "occupied zone", *i.e.* a virtual volume of the room whose boundaries are located *0.6 m* from the walls and at a height of *1.8 m*. Three different heights were considered for the measurements (ISO 7726, 1998): *0.1 m* (ankle level); *1.1 m* (head of a sitting person); *1.7 m* (head of a standing person). The list of equipment used for the measurements is summarized in Table 1. The equipment was calibrated before use.

Table 1. Characteristics of the equipment used for the indoor climate measurements.

Measured parameter	Equipment	Number	Location	Uncertainty
Air temperature	Type T thermocouple	7	Supply. Room (0.1m - 1.1m - 1.7m – 2.2m – 2.5m). Exhaust	±0.3°C
Air velocity magnitude	Omnidirectional thermal anemometer	3	0.1m - 1.1m - 1.7m	±0.06 m/s
Operative temperature	Black globe thermometer	1	1.1m	±0.3°C
Wall temperature	Type T thermocouple	15	0.1m - 1.7m - 2.2m	±0.3°C
SF_6 concentration	Multigas doser/sampler	3	Supply. Room. Exhaust	±2.5 % of measure
Ventilation rate	Hot wire anemometer	1	Supply	±1 l/s

In addition, tracer gas tests were carried out in order to evaluate the ventilation effectiveness in the room. Tracer gas (SF_6) was injected in the supply duct at a constant rate of *0.5 ml/s*, in order to track the fresh air. SF_6 concentration was then measured in the exhaust and in the occupied zone, as well as in the supply duct, upstream of the dosing location. When steady state conditions were reached, the ventilation effectiveness was calculated according to Eq. (2).

$$\varepsilon_c = \frac{C_{room} - C_s}{C_e - C_s} \qquad (2)$$

A ε_c value of *100 %* corresponds to perfect mixing (identical concentration in the room and at the exhaust), a value lower than *100 %* to short-circuiting, and a value higher than *100 %* corresponds to a piston flow situation.

Test conditions

In order to assess the indoor climate in the cubicle office for given boundary conditions, the measurements were performed for constant supply temperature and airflow rate set points. The considered results were then obtained in steady-state conditions. The supply and heat load conditions for the tested cases are presented in Table 2.

Table 2. Boundary conditions for the tested cases.

			Low airflow rate High supply temperature		High airflow rate Low supply temperature	
Measured parameter	**Designation**	**Unit**	**Case 1**	**Case 2**	**Case 3**	**Case 4**
Ventilation rate	$\mathbf{Q_s}$	l/s	17.4	16.8	48.1	49.4
Supply temperature	$\mathbf{T_s}$	°C	31.1	32.0	24.2	24.1
Temperature difference	$\mathbf{\Delta T}$	°C	7.5	5.0	1.2	0.8
Outside temperature	$\mathbf{T_{out}}$	°C	-4.8	-7.0	-2.9	-1.8
Heating power	$\mathbf{P_s}$	W/m^2	16.4	10.7	7.6	4.8
Internal heat gains	**P**	W/m^2	3.1	29.9	3.1	29.9

where P_s is the heat power calculated according to Eq.(1), ρ is the density of air, Q_s is the ventilation rate (in m^3/s), and ΔT is the difference between the supply temperature from the diffuser and the average room temperature. The latter is obtained by an average of the five air temperatures measured in the room.

$$P_s = \rho C_p Q_s \Delta T \qquad (1)$$

- Case 1 and Case 2 correspond to the situation where air is supplied at a low airflow rate and with a high temperature difference. The aim of these cases was to assess the risk for short-circuiting. Short-circuiting of the fresh, warm air, is responsible for poor indoor air quality in the occupied zone, as well as energy loss.

- Case 3 and Case 4 were carried out with a high airflow rate and a low temperature difference. The purpose was to evaluate the risk for discomfort by draught for these conditions, resulting from a too high air velocity in the room.

- In addition, several heat load conditions were studied in order to estimate the influence of the heat sources on the airflow and on the resulting comfort. Case 1 and Case 3 were performed without internal gains (apart from the laptop used to log the tracer gas measurements). For Case 2 and Case 4, a heated cylinder representing an occupant was placed in front of the desk, and lighting was set one during the test. This accounted for *120 W* and *136 W* of sensible heat, respectively.

RESULTS AND DISCUSSION

Operative temperature

The operative temperature measured in the occupied zone for the tested cases is consigned in Table 3. For all cases, the operative temperature is higher than *22.9°C*, which corresponds to the best category (I) of the EN 15251 (2007) standard.

When no internal gains were considered, and for an outside temperature of *-2.9°C*, an operative temperature of *22.9°C* could be maintained with a heating power of only $7.9\ W/m^2$ (Case 3). However, the operative temperature gets too high in the office when internal gains are considered in addition to the warm supplied air: *26.7°C* and *25.0°C* for Case 2 and Case 4, respectively.

Table 3. Measured operative temperature for the tested cases.

		Unit	Case 1	Case 2	Case 3	Case 4
Operative temperature	T_{op}	°C	23.4	26.7	22.9	25.0

Hence, the internal gains cover a great part of the heating demand, underlining the good performances of the building's envelope. As a consequence, preheating of the supplied air above room temperature seems necessary only during the colder days, or when the offices are empty. Nevertheless, the cases with a higher heating power (Case 1 and Case 2) are valuable in order to test air heating for buildings with higher heating demand, *i.e.* conventional or refurbished buildings.

Vertical air temperature difference

The air temperature measured at five different heights in the room is presented on Fig.3 for the tested cases.

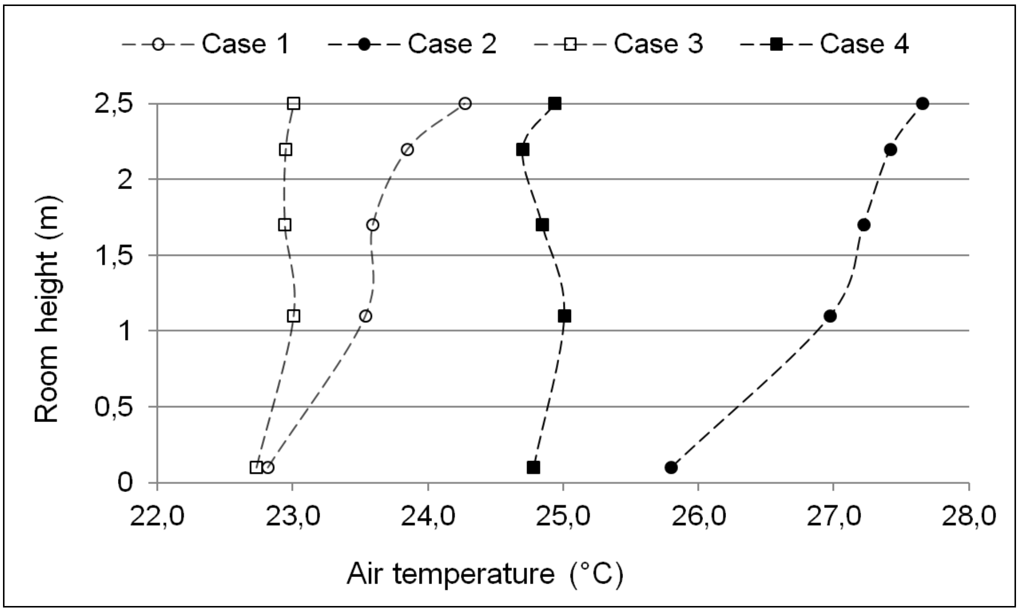

Figure 3. Thermal stratification in the room for the tested cases (empty markers: without internal gains; filled markers: with internal gains).

A too high temperature difference between ankle level (height of *0.1 m*) and head level (*1.7 m*) can lead to discomfort. A gradient of *4.2°C* corresponds with a predicted percentage of dissatisfied of *10 %* (ISO 7730, 2005). For all cases, the thermal stratification is reduced, ranging from *0.1°C* to *1.4°C* between ankles and head level. The risk of discomfort by thermal stratification is therefore very low for the tested cases.
Despite of the low ventilation rate and the high supply temperature, the vertical temperature difference is reduced for Case 1 and Case 2, with a gradient of *0.8°C* and *1.4°C* between ankles and head level, respectively. When a higher airflow rate is considered, the temperature is even more homogeneous in the room, with very low temperature gradients: *0.2°C* for Case 3 and *0.1°C* for Case 4. This emphasizes the efficiency of the considered air diffuser in terms of mixing and induction of ambient air. Moreover, the stratification seems to be influenced only marginally by the presence of the internal heat gains.

Air velocity in the occupied zone

Draught is one of the major risks associated with air heating (Fraefel, 2000). A too high air velocity leads to local cooling of the body, which is perceived as uncomfortable. Draught risk depends on local air velocity magnitude, temperature, and turbulence intensity. Assuming a local turbulence intensity between *30%* and *60%*, it is possible to state that the risk of draught is limited with an air velocity magnitude inferior to *0.15 m/s*. This value corresponds with a predicted percentage of dissatisfied inferior to *10%* (ISO7730, 2005). The air velocity profiles measured in the occupied zone for the tested cases are presented on Fig.4. It has to be noted that the measurement uncertainty after calibration of the probes is ±*0.06 m/s*.

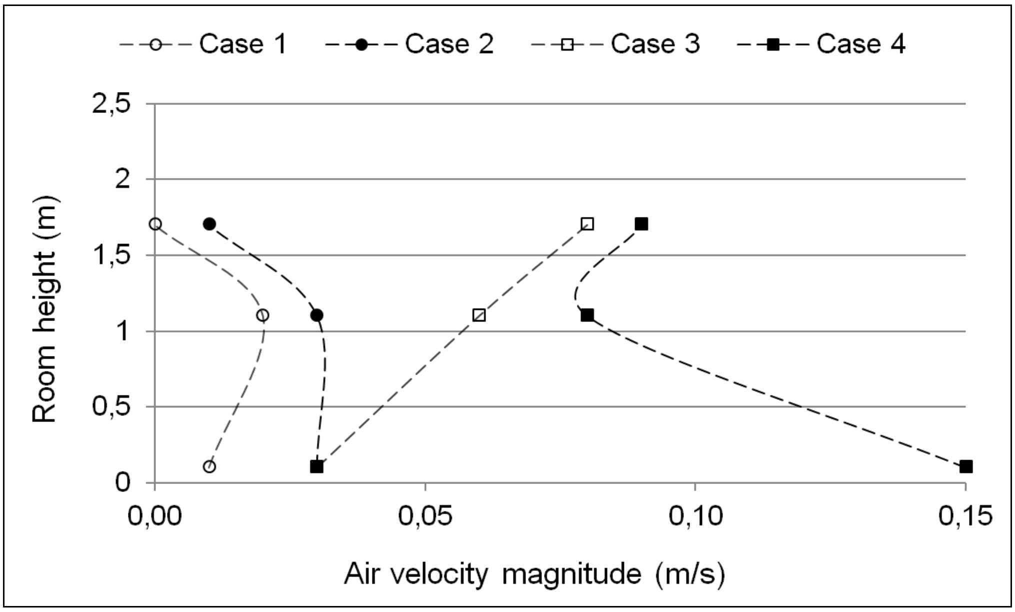

Figure 4. Air velocity magnitude in the occupied zone for the tested cases (empty markers: without internal gains; filled markers: with internal gains).

For Case 1, Case 2 and Case 3, the air velocity values are much lower than 0,15 m/s, thus satisfying the recommendations of the standard. Case 4 may present a limited risk of draught at ankle level where a velocity magnitude of *0,15 m/s* was measured.

As expected, the air velocity is higher in the cases with higher airflow rate (Case 3 and Case 4). This results from an increased entrainment of ambient air in the jet. It is also possible that the air jet from the diffuser flows back into the occupied zone after impinging on the walls, causing an increase of velocity.

Similarly, the internal gains are responsible for an overall increase of the air velocity magnitude in the room, which stems from the entrainment of the ambient air into the buoyant plumes developing on the heat sources. The velocity profile obtained for Case 1 (without internal gains) and Case 2 (with internal gains) is almost identical. This is a strong indication that the airflow pattern in the room is similar in both cases, and driven by buoyancy forces. On the other hand, the air velocity profile is similar between *1.1 m* and *1.7 m* for Case 3 and Case 4, but completely different at ankle level (*0.15 m/s* for Case 4 *vs. 0.03 m/s* for Case 3). This indicates a change in the airflow pattern in the room, and particularly at ankle level when internal gains are considered. It could result from the fact that buoyancy forces play a bigger role in Case 4, while the airflow in Case 3 is governed mostly by inertia forces.

Ventilation effectiveness

The tracer gas concentrations at steady-state and values of ventilation effectiveness obtained for the tested cases are presented in Table 4. The ventilation effectiveness ranges from *89 %* (Case 1), to *102 %* (Case 2).

Table 4. Measured tracer gas concentrations and ventilation effectiveness for the tested cases.

		Low airflow rate High supply temperature		High airflow rate Low supply temperature	
		Case 1	**Case 2**	**Case 3**	**Case 4**
Supply (upstream of dosing)	**C_s (ppm)**	0.2	0.3	0.1	0.1
Room	**C_{room} (ppm)**	37.5	48.3	11.5	13.6
Exhaust	**C_e (ppm)**	42.1	47.4	11.5	13.6
Ventilation effectiveness	**ε_c**	89 %	102 %	100 %	100 %

For Case 1, the tracer gas concentration at the exhaust is higher than the concentration in the occupied zone. This means that a fraction of the fresh, warm air leaves the room without reaching the occupied zone, *i.e.* that short-circuiting occurs. This leads to a ventilation effectiveness of *89 %*, which is acceptable given the unfavorable conditions considered.

However, when internal gains are taken into account (Case 2), the mixing increased and the ventilation effectiveness rises to *102 %, i.e.* perfect mixing. This is consistent with the raise of air velocity in the room observed beforehand. Heat sources, as small as the occupants themselves, have therefore a beneficial impact on the indoor environment quality when heating with air. This is particularly true for a low ventilation rate and a high supply temperature gradient.

A perfect mixing ($\varepsilon_c = 100\ \%$) is obtained as well for a high airflow rate and a low supply temperature (Case 3 and Case 4). There, no difference was observed when internal heat gains were considered (Case 4), since the maximal value for mixing ventilation was already obtained without internal gains (Case 3). This points out the good performances of the diffuser in terms of induction for a broad range of supplying conditions.

As long as the room is occupied, a good ventilation effectiveness is therefore guaranteed. However, short-circuiting can occur when heating rooms which are not occupied, leading to energy loss. A possible solution would be to use presence detectors to avoid supplying air at low airflow rate, high supply temperature in empty rooms, but rather increase the airflow rate. Using a high efficiency heat recovery exchanger on the extract air also reduces the impact of the short-circuiting to a great extent.

CONCLUSIONS

The thermal comfort and the ventilation efficiency were evaluated in a cubicle office of the first office building with passive house standard in Norway. Air heating associated to demand controlled ventilation was used to cover the heating demand. Overall, the measurements report a very good thermal comfort for a broad range of supplying conditions, corresponding to category I of the EN 15251 (2007) standard. For all cases, the risk of draught is limited, with a maximum air velocity of *0.15 m/s* in the occupied zone. Ventilation effectiveness was measured as well, ranging from *89 %* for unfavourable conditions, to *102 %*. Perfect mixing was obtained when internal heat gains were considered. This highlights the beneficial impact of the internal gains on the ventilation effectiveness, especially for low airflow rates.

Using air heating associated with demand controlled ventilation therefore appears to be a relevant solution in office buildings with low energy demand. The good results obtained are accountable largely to the air supply diffuser employed. Using active air supply diffusers whose supplying section varies according to the airflow rate is therefore recommended in order to avoid draught and ensure a satisfactory indoor climate.

Questionnaires related to indoor climate in the occupied open plan offices, as well as CFD simulations of the airflow patterns in the cubicle office are ongoing, and will complete the present study. If interested, see http://www.sintef.no/Projectweb/For-Klima/ for more information.

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