

## TEMPERATURE EFFICIENCY, AIR CHANGE EFFICIENCY AND VENTILATION EFFICIENCY IN AN OFFICE ROOM WITH DISPLACEMENT VENTILATION

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### ABSTRACT

A field study of temperature efficiency, air change efficiency and ventilation efficiency in an office room with a displacement ventilation system has been made.

The air change efficiency and the ventilation efficiency have been determined by using a tracer gas (N<sub>2</sub>O) technique. Concentrations have been measured at four heights in the room and in the supply and exhaust air. Air temperatures at different positions and heights, air supply temperature, air exhaust temperature, outdoor air temperature, insolation, surface temperatures and air velocities at different positions in the room have been measured.

Temperature efficiency was dependent on the supply air temperature. The air change efficiency was very much influenced by the balance between supply air flow rate and the exhaust air flow rate. Values for the ventilation efficiency in the breathing zone were almost equal to those for a dilution ventilation system.

### INTRODUCTION

Displacement ventilation systems can be characterised by an air supply at floor level, an exhaust at or near the ceiling, a low velocity of the supply air and an air supply temperature a few degrees below room air temperature. With this system two zones, one with relatively low air temperatures and low concentrations of pollutants in the lower occupied zone of the room and one with relatively high air temperatures and high concentrations of pollutants in the upper part of the room are created. As a result a higher temperature efficiency (improved indoor comfort) and a higher air change and ventilation efficiency (improved indoor air quality) can be realised in principle in the occupied zone of a room compared to a room with a dilution ventilation system.

The aim of this study will be to determine the performance of a displacement ventilation system in a practical situation.

### MATERIALS AND METHODS

Temperature efficiency ( $E_t$ ) is calculated from the temperature of the exhaust air, the temperature of the supply air and the air temperature in the room as:

$$E_t = (\theta_e - \theta_s) / (\theta_w - \theta_s)$$

with  $\theta_e$  : temperature exhaust air (°C)  
 $\theta_s$  : temperature supply air (°C)  
 $\theta_w$  : air temperature occupied zone (°C)

It is assumed that for a dilution ventilation system (with complete mixing)  $\theta_e = \theta_w$ , and therefore  $E_t$  equals 1.

The contaminant removal efficiency can be characterised by the ventilation efficiency [1]. Ventilation efficiency ( $E_v$ ) is calculated from the concentrations of a tracer gas in the exhaust air, in the supply air and in the room as:

$$E_v = (C_e - C_s) / (C_w - C_s)$$

with  $C_e$  : concentration in exhaust air (ppm)  
 $C_s$  : concentration in supply air (ppm)  
 $C_w$  : concentration in occupied zone (ppm)

When the tracer gas is introduced directly into the room the concentration in the supply air is equal to 0. The ventilation efficiency is then calculated as:

$$E_v = C_e / C_w$$

The air renewal process can be described with the so-called transient air change efficiency. To determine the transient air change efficiency a tracer gas decay method can be used. Tracer gas is introduced to the air supply or to the room. When a steady-state situation is reached the supply is stopped. Subsequently the decay of the tracer gas concentration at different heights in the room is registered.

For a dilution ventilation system with complete mixing of the room air the decay of the tracer gas concentration can be described as:

$$C_t = C_0 * e^{-n.t.}$$

with  $C_t$  = concentration after t sec (ppm)  
 $C_0$  = concentration in steady-state situation (ppm)  
 $n$  = air change rate, equal to  $q_v/V$  ( $s^{-1}$ )  
 $t$  = time (s)  
 $q_v$  = supply air flow rate ( $m^3/s$ )  
 $V$  = volume of the room ( $m^3$ )

This can also be written as:

$$\ln C_t = \ln C_0 - n.t.$$

In a concentration-time diagram with log-scaling of the concentration axis,  $n$  is the slope of a straight line.

By measuring the decay of the tracer gas concentration in a room the air change of a displacement ventilation system can be compared to that of a dilution ventilation system. From the slope of the decay-curves (symbol  $h$ ) at the different levels the so-called apparent air change rate can be determined [1]. With this apparent air change rate the efficiency of the air change in a room is expressed in the equivalent air change rate of a dilution ventilation system.

The transient air change-efficiency  $E_d$  is now defined as:

$$E_d = h/n$$

with  $h$  = measured apparent air change rate ( $s^{-1}$ )

## DESCRIPTION OF THE OFFICE ROOM

The measurements were carried out in the office of BRN Catering in the Dutch city of Capelle aan den IJssel. The measurement room is located on the second floor, beneath the roof. The room (see figure 1) has a floor area of  $16.2 \text{ m}^2$  and a height of 2.7 m. The window (double glazing) in the outside wall has an area of  $3 \text{ m}^2$  (width 2.3 m, height 1.3 m). The complete building is covered with plates of tinted glass at the outside. Between the double glazing and the exterior glass plate horizontal blinds are mounted. The transmission of solar radiation through the window is 0.4 (blinds up) and 0.2 (blinds down).

The air supply device for the displacement ventilation (width 0.48 m, height 0.68 m) is placed in a corner at the outside wall. The bottom of the device is 0.27 m. above floor level. The air exhaust is mounted in the ceiling at 1 m from the rear wall. The supply air temperature is centrally controlled for the entire building and varies from  $20^\circ\text{C}$  with an outside temperature of  $0^\circ\text{C}$  and below, to  $16^\circ\text{C}$  with an outside temperature of  $20^\circ\text{C}$  and above. In summer, cooling is only activated when the outside temperature exceeds  $20^\circ\text{C}$ .

Supply air and exhaust air flow rates are constant during working hours. Initially, the supply air flow rate was  $200 \text{ m}^3$  per hour. The exhaust air flow rate was only  $92 \text{ m}^3$  per hour. The difference between the supply air flow rate and the exhaust air flow rate was balanced by exfiltration through the crack under the door in the rear wall, through leaks between the air supply duct and the floor and through cracks between the walls and the ceiling. From measurements, it was estimated that  $40 \text{ m}^3$  per hour exfiltrated underneath the door,  $25 \text{ m}^3$  per hour via the leaks between the air supply duct and the floor and  $40 \text{ m}^3$  per hour through the cracks near the ceiling. To determine the influence of the disbalance between supply air flow rate and exhaust air flow rate measurements were also carried out with an exhaust air flow rate equal to the supply air flow rate ( $210 \text{ m}^3$  per hour).

The internal heat load during working hours varied from  $332 \text{ W}$  ( $20 \text{ W/m}^2$ ) to  $532 \text{ W}$  ( $33 \text{ W/m}^2$ ) with contributions from the lighting ( $70 \text{ W}$ ), the power supply for the anemometers ( $12 \text{ W}$ ), a personal computer ( $150 \text{ W}$ ) and one to three persons ( $100$  to  $300 \text{ W}$ ). The heat loss from one person was simulated by a  $100 \text{ W}$  lamp in a 10 liter tin.

## EXPERIMENTAL PROCEDURE

Measurement of air temperatures, globe temperature, surface temperatures, air velocities and solar radiation.

Temperatures were measured using thermocouples. Measurements of the air temperatures in the room were carried out at four heights (0.1 m; 0.6 m; 1.1 m; 1.7 m) at five different positions (labelled southwest, northwest, centre, southeast and northeast, respectively) as shown in figure 1. Also measured were the supply air temperature, the exhaust air temperature, the air temperatures in the adjacent rooms, the temperature in the cavity between the double glazing and the exterior glass plate and the outdoor temperature.

The globe temperature was measured at a height of 1.1 m in the centre of the room. Surface temperatures of the walls, floor, ceiling and the glass were measured in the centre of each.

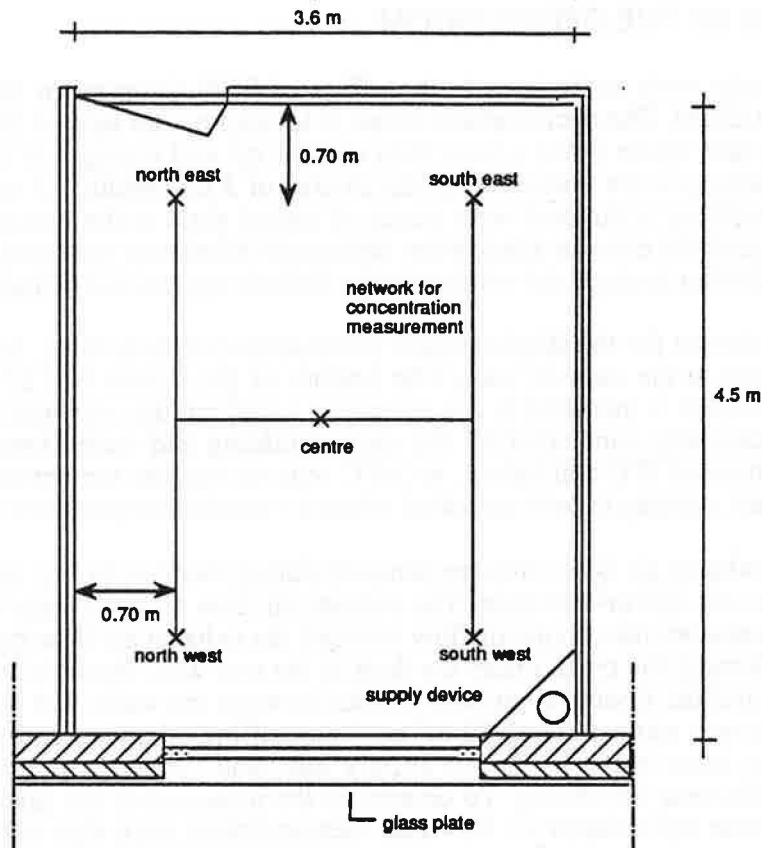


FIGURE 1. Room with measurement positions.

Thermo-anemometers, omnidirectional and with a time constant of two minutes, were used for the measurement of air velocities at 0.1 m and 0.6 m at position southwest, at 0.1 m, 0.6 m, 1.1 m and 1.7 m at position centre, and at 0.1 m at position northeast. Additional measurements of the turbulence intensity of the mean air velocity were carried out with an omnidirectional thermo-anemometer with a low time-constant (Bruel&Kjaer).

The solar radiation was measured with a pyranometer in the cavity between the double glazing and the exterior glass plate.

### Measurement of the concentrations

For the measurements of the air change efficiency the tracer gas decay method (step-down) was used. The length of the duct between the last junction and the air supply device was only 1.5 m. To achieve an acceptable mixing of the tracer gas part of the supply air was extracted from the duct as close to the junction as possible and led via a tube to a mixing box where the tracer gas ( $N_2O$ ) was supplied. By means of a fan in the box the air and tracer gas were mixed. Via a second tube the air is delivered back to the supply duct. The concentrations were sequentially measured at four heights in the room (0.1; 0.6; 1.1 and 1.7 m), in the supply air duct (just before the supply device) and the air exhaust.

To determine the mean value of the concentration at each level a network of tubes in the shape of an 'H' was constructed. Three ends of the 'H' were closed. At the fourth of each 'H' end sample tubes were connected that led to the measuring unit. Air was sampled through small holes, equally spaced along the length of the tubes of each 'H'.

From each "H", from the supply duct and from the air exhaust samples were sucked through tubes into a channel selector and delivered sequentially to a gas analyser (Miran 80, Foxboro Analytical). Every 40 seconds a different sample tube is connected to the gas analyser.

For the measurement of the ventilation efficiency tracer gas was introduced to the room at a constant rate. Concentrations were measured in the same way as described before.

Two personal computers (Hewlett Packard 9816) and two data-acquisition control units (Hewlett Packard 3497A) were used to control the measurements and to store the measurement data on floppy disc.

The measurements were carried out from July 26 to August 10, 1989.

## RESULTS AND DISCUSSION

During the measurement period the daily maximum of the outdoor air temperature varied from 17.8 °C to 26.2 °C. Daily global radiation on a horizontal plane varied from 561 J/m<sup>2</sup> to 2231 J/m<sup>2</sup>. On sunny days the air temperature in the cavity reached values up to 50 °C.

The vertical profile of the room air temperature (mean of five values at each height) is given in figure 2.

Due to the low supply air temperature the lowest mean air temperature was found near floor level. Above 1.10 m there was practically no increase of air temperature. The difference between the mean air temperatures at 0.10 m and 1.70 m varied from 3.0 to 3.5°C with a supply air temperature of 18°C. Close to the supply air device this difference increased to 5°C. When cooling was off and the air supply temperature varied from 19 to 22°C this difference was approximately 1°C less. ISO 7730 (2) recommends a maximum temperature difference of 3°C between ankles and head.

The vertical stratification of the air temperature can be expressed in the mean temperature efficiency for each level (see Table 1).

For the levels of 1.10 m and 1.70 m the temperature efficiency was close to unity so there was no improvement for these levels compared to the use of a dilution ventilation system. For a sitting person the mean temperature efficiency was 1.6, so some improvement was reached.

Temperature efficiency was dependent on the supply air temperature. A higher supply air temperature gave a higher efficiency although the air temperature in the room was higher. This presumably somewhat unexpected result can be explained by the smaller differences between the air temperature in the room and the air supply temperature; in other words, the divisor in the calculation of the temperature efficiency. This indicates that some caution should be taken when interpreting temperature efficiency values.

TABLE 1. Temperature efficiency at different heights.

Supply air temperature	Height above floor (m)			
	0.10 m	0.60 m	1.10 m	1.70 m
18°C (cooling on)	2.0 - 2.2	1.5 - 1.7	1.1 - 1.2	1.0 - 1.1
19 to 22°C (cooling off)	2.0 - 2.6	1.5 - 2.0	1.2	1.1



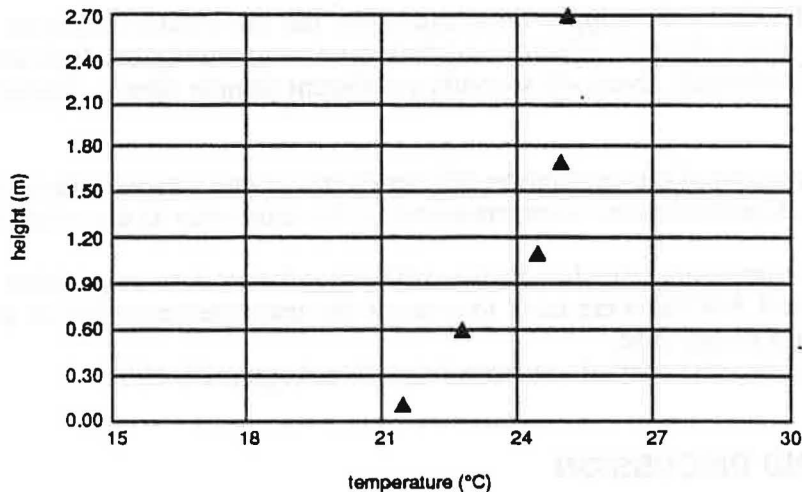


FIGURE 2. Vertical profile of mean air temperature at different heights. Air supply temperature 18°C.

The different internal and external heat loads (outdoor temperature and solar radiation) had no effect on the temperature efficiency. There was no disturbance of the stratification by people walking around the room. No difference in temperature efficiency was found between the situation with an exhaust air flow rate of 92 m<sup>3</sup> per hour and that with an exhaust air flow rate of 210 m<sup>3</sup> per hour.

At a height of 0.60 m at the position southwest and at a height of 0.10 m at the position centre relatively high air velocities of 0.35 m/s and 0.16 m/s, respectively were found. These two positions are within the air supply jet. At the other measuring points the air velocities were lower than 0.10 m/s. Turbulence intensity was measured on July 28 1989 from 16.00 to 17.00 hours. At a height of 0.60 m at position southwest the turbulence intensity was 63%. At a height of 0.10 m at position centre the turbulence intensity was 28%. According to (3), with air temperatures of 21.5 and 21.2, respectively, this would lead to percentages dissatisfied due to draught of approximately 67% and 16%, respectively.

The air change efficiency expressed in the mean apparent air change rate and the transient air change efficiency are given in Table 2.

TABLE 2. Mean values of the apparent air change rate and the transient air change efficiency.

	Exhaust air flow rate	
	92 m <sup>3</sup> /h	210 m <sup>3</sup> /h
apparent air change rate (s <sup>-1</sup> )	0.00132	0.00228
nominal air change rate (s <sup>-1</sup> )	0.00127	0.00133
transient air change efficiency (-)	1.04	1.71

The values are based on the measurements during working hours (8.00-18.00 hours). A cycle was established of one hour of tracer gas supply and one hour without it. Within one hour the concentrations reached steady-state values.

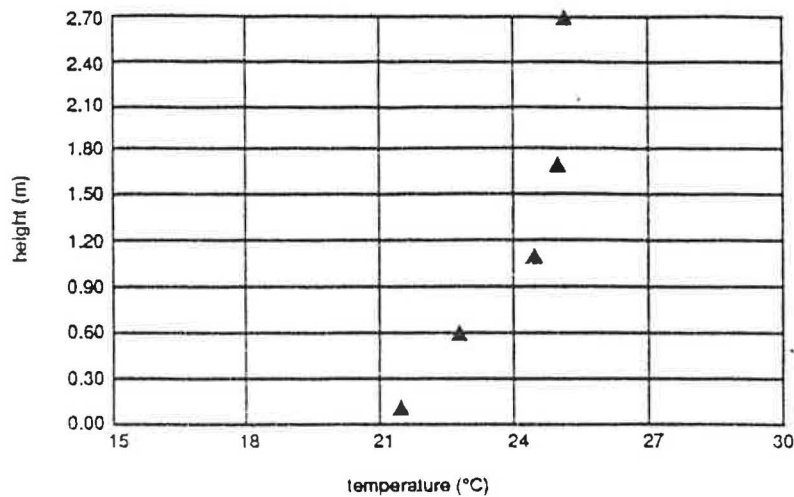


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The highest air change efficiency was found in the situation with the high exhaust air flow rate (210 m<sup>3</sup>/hour). The mean transient air change efficiency was 1.71. In the situation with the exhaust air flow rate of 92 m<sup>3</sup>/hour the transient air change efficiency was close to 1. This lower transient air change efficiency was caused by the short-circuiting of the air due to the exfiltration near floor level.

The transient air change efficiency was higher in the afternoon than in the morning. This was probably caused by the bigger stratification of the air in the room, due to the lower supply air temperature in the afternoon.

No major influences on the air change efficiency caused by the presence of people walking around in the room were found.

For the measurement of the ventilation efficiency tracer gas is introduced directly to the room at different positions.

From the results (Table 3) it can be seen that the values for the ventilation efficiency are very much dependent on the position of the source in the room. When the source was placed close to the supply device relatively low values were found. For the breathing zone (1.10 m and 1.70 m) the measured ventilation efficiency was close to unity so no improvement compared to a dilution ventilation system was reached. Walking around in the room had a substantial effect on the concentration distribution for the levels of 0.10 m and 0.60 m.

TABLE 3. Mean ventilation efficiency at different heights for different positions of tracer gas supply.

Position supply	Height above floor (m)			
	0.10 m	0.60 m	1.10 m	1.70 m
<i>exhaust air flow rate 92 m<sup>3</sup>/h</i>				
height 0.10 m near centre	-	3.0	0.6 - 0.7	0.6 - 0.7
height 1.10 m near centre	5.6	5.0	-	0.8
height 1.10 m, near position northwest	2.9	2.0	-	0.9
height 1.10 m, near position northeast	6.0	5.3	-	0.9
height 0.10 m, near position northeast	-	1.5	0.9	0.9
height 0.10 m, near position southwest	-	1.3	1.2	1.2
height 0.60 m, near position southwest	1.4	-	1.0	1.0
<i>exhaust air flow rate 210 m<sup>3</sup>/h</i>				
height 1.10 m, near centre no people present	10 - 12	8 - 10	-	1.1
two people present, walking	4.5 - 5	3.5 - 4	-	0.9

## CONCLUSIONS

A mean air temperature efficiency of 2.1, 1.6, 1.1 and 1.05 for the levels 0.10 m, 0.60 m, 1.10 m and 1.70 m was found in an office room with a displacement ventilation system. This implies some improvement compared to a dilution ventilation system. Temperature distribution in the room was found to be dependent on the supply air temperature. No effect of the internal heat load, outdoor temperature, insolation, exhaust air flow rate or people walking around was found.

The air change efficiency was very much dependent of the balance between supply air flow rate and exhaust air flow rate. A relatively high exfiltration rate near floor level, due to a disbalance between supply air flow rate and exhaust air flow rate, led to a low transient air change efficiency. A high transient air change efficiency was reached when the exfiltration near floor level was limited by balancing supply air flow rate and exhaust air flow rate. When displacement ventilation systems are used care should be taken to avoid exfiltration near floor level.

The values for the ventilation efficiency in the breathing zone were close to unity so no improvement compared to a dilution ventilation system was reached here. The measured values of the ventilation-efficiency were very much dependent on the position of the source in the room.

## Acknowledgement

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