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Displacement Ventilation Systems in Office Rooms

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ABSTRACT

Ventilation by displacement is a type of mechanical ventilation that is becoming more popular in Scandinavia. Its physical principle is based on the utilization of density or thermal stratification in order to achieve unidirectional (piston-like flow pattern) flow in the occupied zone. This paper summarizes results from experimental work carried out with regard to this type of ventilation.

Measurements carried out in an office room showed that there are limits to this type of ventilation. There are specific requirements in the design of supply air terminals in order not to exceed comfort criteria with regard to acceptable air velocities. Furthermore, the temperature gradient associated with this type of ventilation puts a restriction on the maximum heat load.

Ventilation by displacement gives rise to larger room-average ventilation effectiveness than traditional mixing systems. However, with the flow rates normally supplied to office rooms, the concentration in the breathing zone is the same as with traditional mixing systems.

INTRODUCTION

By "displacement ventilation" we mean supply of air with a low-velocity diffuser located at floor level. The air is supplied with less temperature than the ambient and extracted air at ceiling level. The aim of this ventilation principle is to create supply air conditions in the occupied zone, while the aim with traditional mixing systems is to create extract-air conditions in the whole room. Ventilation by displacement based on the supply of air at low velocity at floor level has been installed in industrial halls with large room heights in Scandinavia for at least two decades. For about five years this type of ventilation has been installed in office rooms. It is also becoming popular in Germany (Fitzner 1987). A number of laboratory tests of the performance of this type of ventilation system have been reported (Mathisen 1988; Mathisen and Skaret 1983; Nielsen 1988; Sandberg and Sjoberg 1983; Sandberg 1985; Skaret 1987; Palonen et al. 1988). The local thermal discomfort due to this type of ventilation has been explored and is reported by Wyon and Sandberg (1989). Special dynamic models that describe the evolution of the characteristic front (see Figure 1) have also been developed (Sandberg and Lindstrom 1987).

This paper describes laboratory tests carried out in order to see if this type of ventilation gives rise to better air quality in office rooms than traditional "mixing" ventilation.

Physical Principle

When mechanical ventilation is used, the air is usually supplied with a very high velocity in the form of a turbulent jet. Air is entrained into the jet and large secondary air motions are set up in the room. This gives rise to an intensive mixing of both ventilation air and contaminants and, in the perfect system, the concentrations will be the same everywhere in the room as in the extract duct.

$$C_e(\infty) = \frac{\dot{m}}{q_s} \tag{1}$$

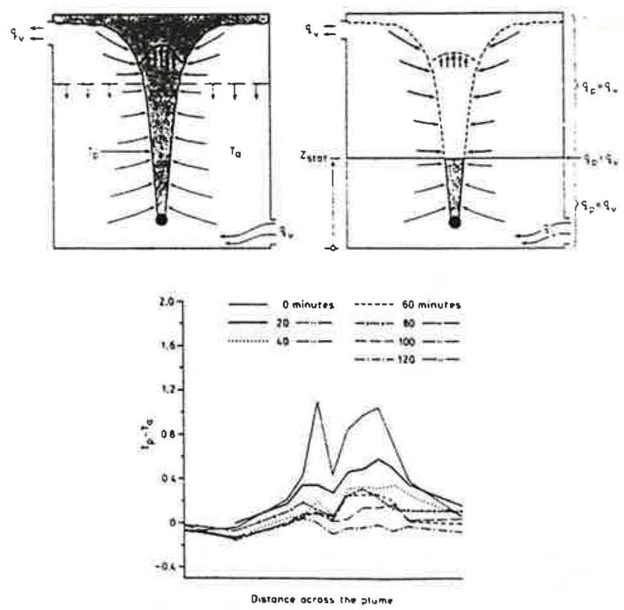

Figure 1 Evolution of the "front" and the temperature difference between the plume and the environment

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where

$\dot{m}$ = release rate of contaminant

$C_e(\infty)$ = equilibrium concentration in the extract duct

q_s = volumetric flow rate of ventilation air

The characteristic feature of a free jet is that its momentum is conserved and in this type of ventilation the supply of air is a source of momentum. This type of ventilation may therefore be called momentum-based ventilation.

In systems where one tries to achieve ventilation by displacement, the air is supplied at floor level and with a low momentum. The air is supplied with a lower temperature than the ambient and the ventilation air is now both a source of momentum and buoyancy. We assume that both the velocity and the volumetric flow rate have the same distribution, $f(z)$, over the surface of the diffuser and are a function only of the height, z.

$$u(z) = \bar{u}_s \cdot f(z) \cdot dz$$

$$q(z) = q_s \cdot f(z) \cdot dz$$

where

$$\int_0^H f(z)dz = 1$$

H = height of the terminal (m)

$\bar{u}_s$ = characteristic initial mean velocity ($\bar{u}_s = q_s/A_s$) (m/s)

The flow force, M_s, of the air entering the room is the sum of the momentum flux and the excess hydrostatic pressure due to the temperature difference.

$$M_s = \rho_s u_s q_s + \rho_s g' A_s \bar{Z} \quad \text{(N)}$$

where

A_s = characteristic initial cross section (m^2)

g' = reduced acceleration of gravity ($g' = g\frac{\Delta\rho}{\rho} = g\frac{\Delta T}{T_a}$) (m/s^2)

T_a = temperature (K)

ρ_s = density of air (kg/m^3)

ΔT = temperature difference between the flow and the ambient (°C)

$\bar{Z}_s$ = height of the centroid $\bar{Z} = \int_0^H z \cdot f(z)dz$, (m)

The specific buoyancy flux, B_s, from the supply air diffuser is equal to:

$$B_s = g'q_s \quad (m^4/s^3)$$

When the supply air is colder than the ambient air, it drops down to the floor. The order of magnitude of the velocity of the air, $u(0)$, at the point where the air hits the floor can easily be obtained from an energy balance. The total influx, E_s, of kinetic and potential energy to the room from the diffuser is

$$E = \tfrac{1}{2}\rho_s q_s \bar{u}_s^2 + g\Delta_s \bar{Z}$$

When we assume that there is no entrainment of air or exchange of heat, the density and the volumetric flow rate are conserved. Therefore, the flow of energy, $E(0)$, where the air hits the floor is equal to

$$E(0) = \tfrac{1}{2}\rho_s q_s u(0)^2 + g\Delta_s \bar{Z}(0)$$

where

$u(0)$ = mean velocity where the air hits the floor (m²/s)

After equating the above two expressions, we can solve for $u(0)$:

$$u(0) = \sqrt{\bar{u}_s^2 + 2g'(\bar{Z}_s - \bar{Z}(0))} \quad \text{(m/s)}$$

We assume that the velocity distribution of the the air flowing out on the floor is uniform ("top hat" profile). Therefore, we may set

$$Z(0) = \frac{h(0)}{2}$$

where

$h(0)$ = the thickness of the layer of air when it drops down on the floor

A theoretical analysis shows that one should expect that the thickness, $h(0)$, of the air layer is a function of the geometry of the terminal, downstream conditions, and the distance from the diffuser where the flow is dissolved. However, experience shows that the layer of air flowing out on the floor is very thin ($\approx$ 10 cm) and therefore $\bar{Z}(0) << \bar{Z}_s$ and we may put $u(0)$ equal to

$$u(0) = \sqrt{\bar{u}_s^2 + 2g'\bar{Z}_s} \quad \text{(m/s)}$$

If the airflow coming from the supply air diffuser is uniformly distributed over the height of the terminal, then

$$f(z) = \frac{1}{H}$$

$$\bar{Z}_s = \frac{H}{2}$$

$$u(0) = \sqrt{\bar{u}_s^2 + g'H}$$

A linear distribution of the airflow, with the airflow equal to zero at $z = H$, gives rise to

$$f(z) = \frac{2}{H}(1 - \frac{z}{H})$$

$$\bar{Z}_s = \frac{H}{3}$$

$$u(0) = \sqrt{\bar{u}_s^2 + \tfrac{2}{3}g'H}$$

In order to characterize the supply of buoyancy to the room, it is convenient to define a velocity, U_B, as:

$$U_B = \sqrt{g'H} \quad \text{(m/s)}$$

The nondimensional Archimedes number, Ar, is the ratio between the buoyancy and the momentum (inertia) and is defined as

$$Ar = \frac{U_B^2}{\bar{u}_s^2} = \frac{g'H}{\bar{u}_s^2} \qquad (1)$$

Please observe that the square root of the Archimedes number is equal to the ratio between the two velocity scales.

Figure 1 shows the evolution of the flow pattern in a system designed to give rise to displacement ventilation. The room is equipped with a low-velocity terminal located

TABLE 1

Diffusers

	Width x Height [cm]	Geometry	Perforation	Perforation degree [cm^2]	Symbols used in figures
1	18 x 75 (1350 cm^2)	Semicircular	Rectangular holes (59 mm^2)	50 %	◡
2	38 x 50 (1900 cm^2)	Semicircular	Quadratic holes (24 mm^2)	10 %	◒
3	45 x 50 (2250 cm^2)	Flat	Quadratic holes (92 mm^2)	50 %	□
4	45 X 50 (2250 cm^2)	Flat	Quadratic holes (24 mm^2)	9 %	■

at floor level and with the extract air terminal located at ceiling height. In the room there is a combined contaminant and heat source that generates a turbulent plume. The airflow in the plume is always directed upward and the plume entrains ambient air during its upward course and thus mixes ambient air with its own airflow. First is shown a sketch of the flow pattern shortly after the source has been turned on. When the first light contaminant reaches the ceiling, it spreads out laterally and produces a layer, a "front," that starts to descend. The front stops where the airflow in the plume is equal to the ventilation airflow supplied. This recirculation of heat and contaminant causes the contaminant and temperature difference between the plume and its environment to diminish. At the bottom of Figure 1 is an example of recorded temperature difference, $T_p - T_a$, between the plume and its environment. When steady state has been reached, there is no temperature difference between the contaminant and its environment. That is, complete mixing is achieved in the upper part of the room and the concentration there becomes equal to the concentration in the extract, which is given by Equation 1. Below the stationary front level, the airflow in the plume is smaller than the supplied ventilation airflow rate. This implies that the airflow in the ambient is directed upward. As a result, we obtain a unidirectional flow in the whole lower zone and the conditions outside the plume are equal to the conditions in the supply air. Therefore, if we ignore the concentrations in the narrow plume in the lower zone, the room average ventilation effectiveness, $<\epsilon>^{-1}$, becomes approximately equal to:

$$<\epsilon>^{-1} = \frac{C_e(\infty)}{<C>} \approx \frac{C_e(\infty)}{\frac{(H_r - z_{sta})}{H_r} C_e(\infty)} = \frac{H_r}{(H_r - z_{sta})}$$

where

$<C>$ = room-average concentration

H = height of room

That is, the ventilation effectiveness is very strongly related to the height, z_{sta}, of the stationary level. Note that the above definition of the ventilation effectiveness is based on the room average concentration, which is not necessarily equal to the concentration to which people are exposed. We will consider an example in order to gain insight into what order of magnitude of the ventilation effectiveness one should require in order to achieve better air quality with deplacement ventilation. Consider a room where a person is sitting. The room height is 2.6 m and the person is breathing air from the 1.2 m level. Therefore the level of the stationary front, z_{sta}, must at least be 1.3 m. This implies that, according to the above approximation, the ventilation effectiveness must amount to at least 200%.

Test Room

The measurements were made in a test room in a laboratory that has the form of a common type of office module measuring 4.2 × 3.6 × 2.5 m (L × W × H). The "external" wall had three narrow windows and was built against a wall of water-filled radiators that could be heated or cooled to simulate external weather conditions.

In all tests the office was ventilated by means of a low-velocity diffuser attached to the rear wall. The diffuser was either standing on the floor or raised 8 cm above floor level. The purpose of lifting the terminals above the floor surface was to see if this would give rise to an increased entrainment of air. The air left the room through a slot 2.2 m above floor level in the rear wall.

In the tests four different types of low-velocity diffuser were used. Two of them were semiround, while the remaining two were flat. The two semiround diffusers and the two flat ones had different degrees of perforation. The manufacturers had hoped that the degree of perforation would affect the entrainment. The properties of the diffusers are presented in Table 1. Diffuser 1 was used in the comfort tests reported by Wyon and Sandberg (1989).

TABLE 2

Summary of Test Conditions

Diffuser "Season"	Flow rate [m^3/h]	$T_s - T_e$ [^oC]	$T_{1.1}-T_{0.1}$ [^oC]	E_v [Watt]	U_s [m/s]	U_B [m/s]	Ar
1 W	100	-5.7	1.9	182	0.41	0.38	0.86
1 W	200	-2.1	0.8	142	0.82	0.23	0.079
1 S	100	-9.1	3.4	303	0.41	0.48	1.37
1 S	200	-4.7	1.7	313	0.82	0.34	0.17
1 S(8 cm)	100	-8.5	3.1	284	0.41	0.46	1.26
1 S(8 cm)	200	-4.6	1.8	303	0.82	0.34	0.17
2 S	100	-8.6	3.2	287	1.49	0.38	0.07
2 S	200	-4.1	1.4	270	2.99	0.26	0.0076
2 W	200	-2.5	0.7	168	2.99	0.20	0.0045
2 W	100	-5.3	1.8	178	1.49	0.30	0.041
2 S(8 cm)	100	-8.4	3.3	280	1.49	0.37	0.062
2 S(8 cm)	200	-4.7	1.6	310	2.99	0.28	0.0088
3 W	100	-5.7	2.1	188	0.25	0.31	1.53
3 W	200	-3.0	1.4	200	0.50	0.22	0.19
3 S	200	-4.7	2.4	310	0.50	0.28	0.31
3 S	100	-9.8	4.0	325	0.25	0.41	2.69
3 S(8 cm)	100	-9.7	3.9	323	0.25	0.40	2.56
3 S(8 cm)	200	-5.1	2.6	342	0.50	0.29	0.33
3 I	200	0.2	0.0	11	0.50	-	-
3 I	100	-0.1	0.2	3	0.25	-	-
4 I	100	0.0	0.1	0	1.44	-	-
4 I	200	0.1	0.0	0	2.88	-	-
4 S	200	-5.0	2.0	330	2.88	0.29	0.01
4 S	100	-9.5	3.9	317	1.44	0.40	0.077
4 S(8 cm)	100	-9.5	3.9	316	1.44	0.40	0.28
4 S(8 cm)	200	-4.8	1.8	320	2.88	0.28	0.097
4 W	100	-4.4	1.7	147	1.44	0.27	0.188
4 W	200	-2.2	0.8	148	2.88	0.19	0.066

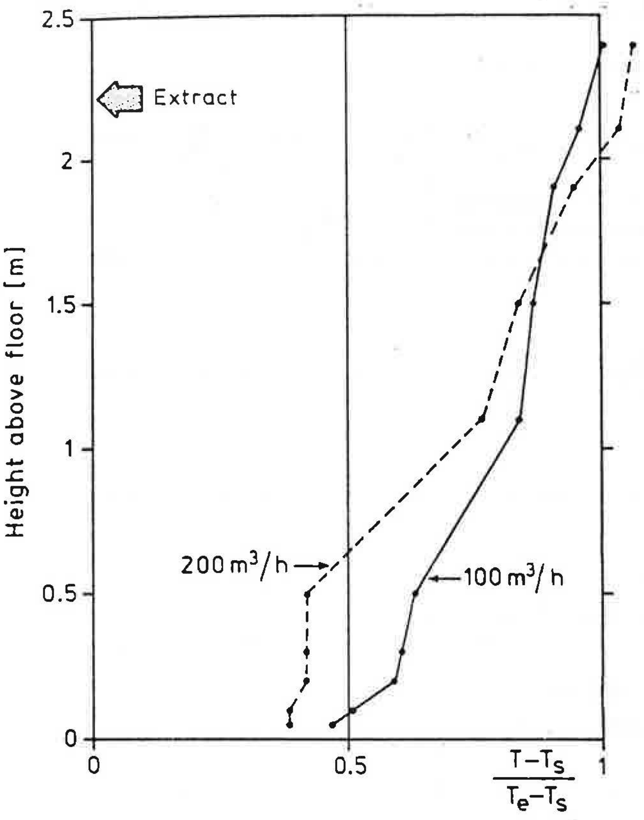

Figure 2 *Non-dimensional temperature profile in the room*

The tests were carried out at two flow rates (100 m^3/h and 200 m^3/h) and at two different cooling loads corresponding to a summer (S) and a winter (W) condition, respectively. Some tests were also made under isothermal conditions (I). In all tests reported in this subsection, a person was simulated by a 100 W electric bulb. In the winter conditions the total internal load was 415 W while in the summer conditions total internal load amounted to 295 W.

Table 2 shows a summary of the actual test conditions. Numbers in parentheses under the heading diffuser denote the cases when the supply air terminal was raised 8 cm above the floor area.

Temperature Field

The room air temperatures were recorded in the middle of the room at 10 points located along a vertical line. In Table 2 is presented the recorded temperature difference, $T_{1.1} - T_{0.1}$, between 1.1 and 0.1 m above floor level.

Figure 2 shows two typical vertical temperature profiles recorded in the room 3.20 m from the supply air terminal. The temperature is given as the recorded temperature minus the supply air temperature divided by the difference between the supply and extract. There is a temperature increment on the way from the supply air terminal to the point of measurement, which was located about 3.20 m from the supply air terminal. We see that the temperature difference between room air and supply is always less than half the difference between the extract and the supply. In each test the supply air temperature was chosen so that the room air temperature would be as close as possible to the temperature in the adjacent rooms. This was done in order to reduce the transfer of heat to or from the test room. Furthermore, before the measurements started, each test condition was run for a sufficiently long time to achieve steady state. Despite these precautions, the heat loss, E_v, due to the ventilation air

$$E_v = \rho c_p (T_s - T_e)$$

where

T_e = temperature of the extract air

T_s = temperature of the supply air

exhibited a strong dependence on the magnitude of the ventilation airflow. At low airflow rates (0.5–1.0 room volumes/h) only 30% to 40% of the heat given off within the room was removed by the ventilation air. However, when the ventilation airflow rate was greater than 2 room volumes per hour, all internal heat was transported from the room by the ventilation air. This behavior is not fully understood. It requires quite a detailed model of both the convective and radiative heat transfer within the room for a correct analysis. A possible explanation of this phenomenon may be as follows: the plumes originating from the heat sources transport the heat upward to the ceiling, raising its temperature. As a result, the ceiling radiates heat toward the other room surfaces and the floor is also warmed. This heat is transferred both to the floor structure and the air. When the ventilation airflow increases, the velocities along the floor increase, which means, in turn, that a greater fraction of the heat is transferred to the room air.

Figure 3 shows the temperature difference between 0.1 and 1.1 m above floor level as a function of the temperature difference between supply and extract. There is an almost linear relationship. We see that this relationship is more or less independent of the shape of the supply air terminal and the degree of perforation. Nor does raising the supply air device to 8 cm above floor level alter the situation. According to the ISO/DIS 7730 recommendation, the temperature difference between these levels must not exceed 3 K. Based on this criterion, the temperature difference between supply and extract in an office room of normal room height must not exceed 7-8 K. This is in accordance with results from tests carried out in a room of the the same size but equipped with another type of low-velocity terminal. However, it is 1 K smaller than found in the investigation reported by Palonen et al. (1988).

Velocity Field

The velocities were recorded along the centerline of the diffuser beginning at 50 cm from the diffuser and to a point 325 cm from the diffuser. The velocities were recorded both at 3 and 7 cm above the floor surface. The spread of the air on the floor and the velocity field to that terminal gives rise to complex phenomena. Figure 4 shows examples of velocities recorded when the flow rate is 100 m^3/h, while Figure 5 shows the velocity recorded when the flow rate is 200 m^3/h. Figure 6 shows the spread of the air close to the diffuser.

The spreading of the air on the floor and the velocity field is dependent on many factors. Close to the diffuser there is a "near-field" region where the air is accelerating in the vertical direction and the air undergoes a free fall.

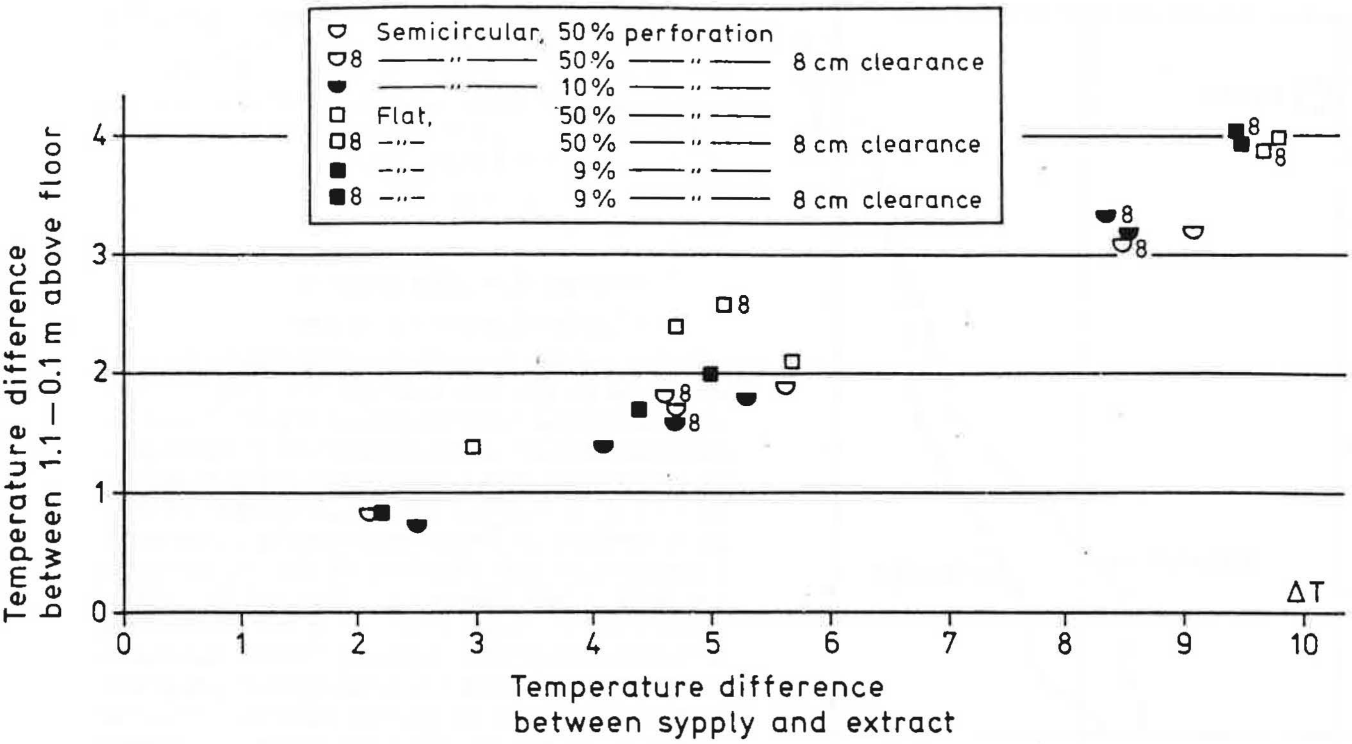

Figure 3 *Temperature difference between 1.1 m and 0.1m above the floor level as a function of the temperature difference between the extract and supply.*

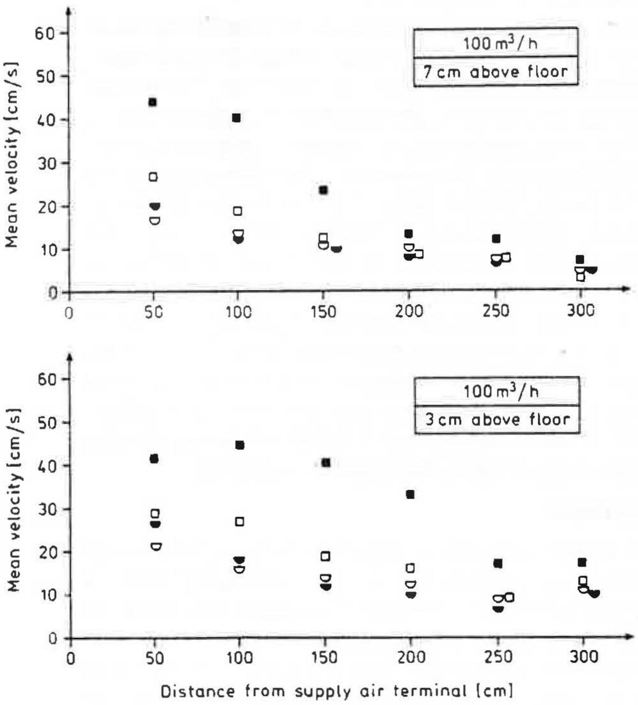

Figure 4 *Example of recorded mean velocities at summer conditions when the flow rate is 100 m^3/h (2.6 room volumes/h). Legend, see Figure 3*

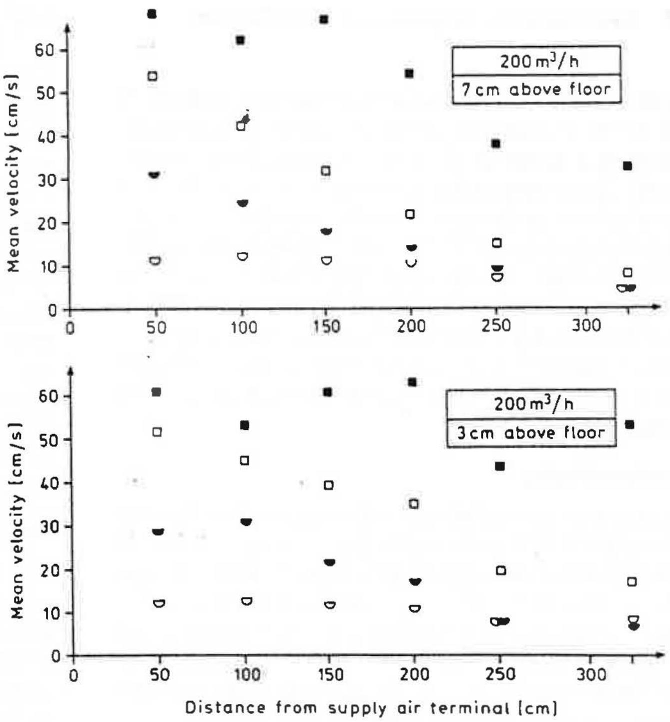

Figure 5 *Example of recorded mean velocities at summer conditions when the flow rate is 200 m^3/h (5.2 room volumes/h). Legend, see Figure 3.*

The point where the air drops down starts the "far-field" region. In the "far-field" region we can identify at least the following factors:

- shape (geometry) of the diffuser
- the ratio between the momentum flux and the buoyancy, that is, the Archimedes number

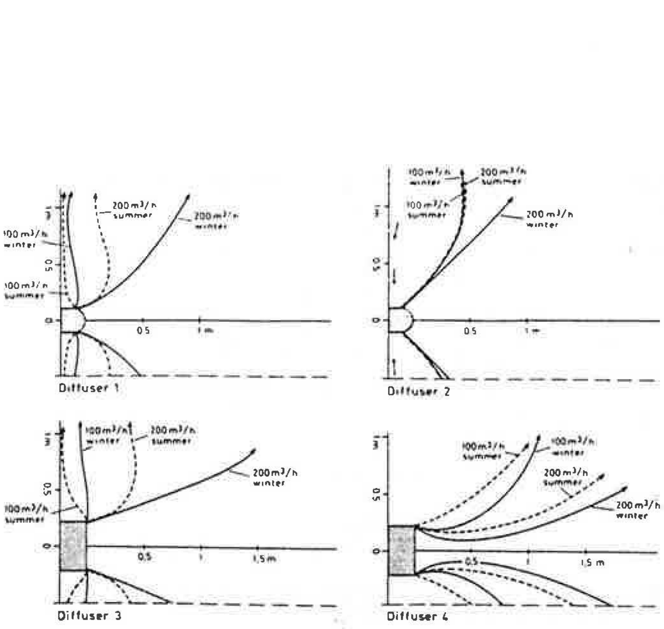

Figure 6 *Spread of the air close to the diffuser*

— the distance between the diffuser and the point where the flow is dissolved
— the geometry of the room.

The flow of air with less density than the ambient air from this type of diffuser cannot be analyzed as a classical wall jet. When the temperature difference is sufficiently large, the flow is probably best analyzed as a gravity current (Sandberg 1989). This means that the spread of the air consists of motion in the direction of the supply, driven both by the initial momentum flux and the buoyancy flux, and in the lateral direction there is a gravitational spread with velocity v_g equal to:

$$v_g = K\sqrt{(g'h)}$$

where

K = constant
h = thickness of the air layer

If we make the following assumptions regarding the layer of air that is spread out on the floor:

— the density (temperature) difference between the air layer and its ambient is constant
— the thickness of the air layer is constant
— there are no vertical velocities

then the theory predicts that the velocity decay vs. distance for a three-dimensional flow becomes:

$$u(x) = u(0)e^{-\frac{2\sqrt{g'}}{q_s}x}$$

That is, we have an exponential decay of the velocity. The growth of the width, b, of the flow becomes

$$b(x) = b(0)e^{+\frac{2\sqrt{g'}}{q_s}x}$$

where

$b(0)$ = intial width

If the above two expressions are correct then the importance of the thickness of the air layer is demonstrated. The theoretical analysis shows the height of the air layer is strongly related to magnitude of the distance between the diffuser and the point where the air layer is dissolved.

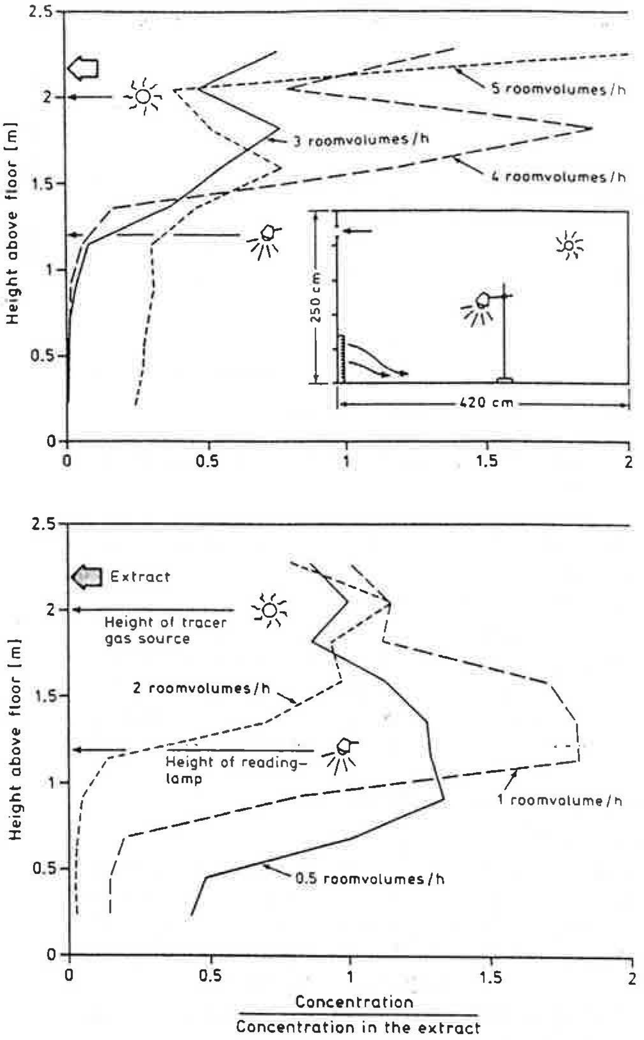

Figure 7 *Concentration profiles in the room at different ventilation air flow rates*

From Figures 4 through 6 we see the strong effect of the governing parameters. At the lowest flow rate (100 m³/h), the air is spread over a wide sector (180°) when diffusers 1-3 are used. This holds true both for the summer and winter conditions. This means that in this case the flow is strongly influenced by the buoyancy. Diffuser 4 is an exception that gives rise to a narrower spread of the air. When the flow rate is increased, the angle of spread diminishes for all terminals. The flat diffusers give rise to the smallest angle of spread. Again terminal four is the extreme case. Within a distance of 1.5 m from the terminal, a contraction of the flow occurs. At 3 cm above floor level, the velocity stays constant up to a distance of 2 m from the terminal. This gives rise, as we see, to unacceptably high velocities. The conclusion to be drawn is immediate: in order to avoid complaints about the thermal sensation, a terminal that gives rise to a spread of the ventilation air over a wide angle must be used. A smaller degree of perforation in order to achieve a larger entrainment does not improve the situation. The result is opposite to the one desired. The velocities in the room become higher when the perforation degree is reduced. The velocities at the terminal increase, but the

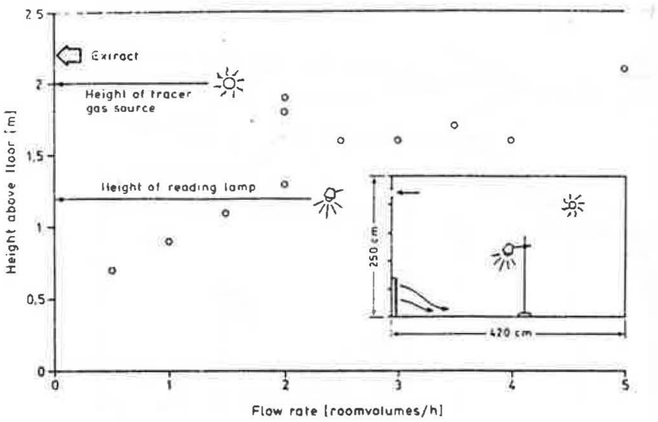

Figure 8 *Height of the stationary front at different ventilation air flow rates*

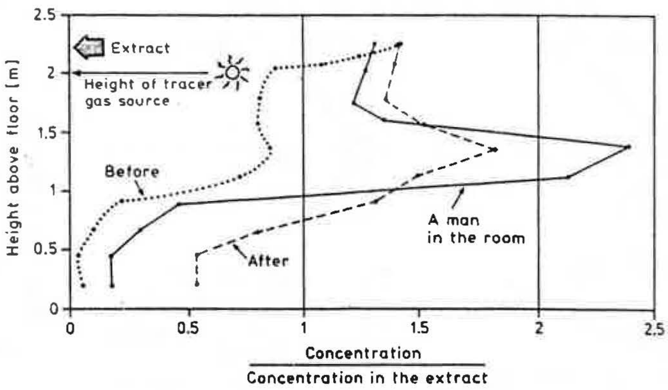

Figure 9 *Concentration profiles in the room when (a) the room is empty, (b) there is a person in the room, and (c) the person has left the room*

entrainment rate seems to be affected very little by changing the degree of perforation.

For all terminals, the velocity field up to 2.5 m from the diffuser is governed by the actual conditions at the supply air terminal. Beyond that point there is an influence from the wall opposite the diffuser. This may be ascribed to flow reversal due to reflection of the flow at the wall and, during winter conditions, downflow of cold air from the windows.

Level of Stratification in the Room

The level of the stationary front in the room is a very important air quality parameter. In order for ventilation by displacement to be regarded as being a better system than systems based on complete mixing (high-momentum systems), the stationary front must be pushed sufficiently high up in the room. Another important aspect is the stability of the stratification. For example, will the stratification be destroyed by the agitation caused by people moving around in the room? In order to gain some insight into these questions, several tests were carried out with an artificial contaminant source in the room. As a contaminant, a mixture of N_2O and helium gas was used, mixed in such proportions that the density of the gas mixture would be the same as for air.

During these experiments, the "windows" were insulated in order to keep the temperatures of the room surfaces close to the room air temperature. This was done in order to prevent boundary layer flow at the room surfaces. The ventilation air was supplied to the room via terminal 1, described in Table 1. In the first experiment, the heat source in the room consisted of a standard reading lamp with a 100 W electric bulb. The electric bulb was located 1.2 m above floor level. In order to be able to detect the stratification, the injection of the tracer gas was done at 2 m above floor level, which we expected to be well above the location of the stratification. Figure 7 shows the recorded vertical concentration profiles in the room. By the concentration in the extract in Figure 7 we do not refer to a measured concentration but a concentration that is calculated according to Equation 1. The strong stratification appears clearly at each flow rate. We see that at flow rates less than 2 room volumes per hour, the location of the stratification is below the location of the reading lamp. This surprising result must be an effect of the occurrence of boundary layer flows along the walls, despite the precautions taken in order to avoid them. However, this is what occurs in practice. Below the stationary front, the wall temperature is higher than the room air temperature, which, in turn, gives rise to upward-directed boundary layer flows along the walls. By radiation from the ceiling, the floor is heated, which may give rise to "thermals" starting from the floor. Continuity dictates that the "front" stop at a height, z_{stat}, where

$$q_s = q_p(z_{stat}) + q_+(z_{stat}) - q_-(z_{stat})$$

where

- q_p = volumetric flow rate in the main plume
- q_+ = volumetric flow rate in other (boundary layer of plume) flows directed upward
- q_- = volumetric flow rate in other flows directed downward

From the above relationship, we see that another flow directed upward will lower the stationary level, while falling boundary-layer flows have the opposite effect. In Figure 8, the stationary front is given as a function of the ventilation airflow rate. If we define the height of the occupied zone to be 1.8 m, we see that in order to push the contaminants above the occupied zone, a minimum flow rate of 120 m^3/h is needed (3 room volumes/h). This result is in accordance with results obtained earlier in a room of the same size reported by Sandberg (1985). A light gas was released 1.2 m above the floor. When the flow rate of air amounted to 75 m^3/h (2 room volumes/h) then the maximum ventilation effectiveness in an empty room amounted to 180% while it dropped to 140% with a thermal mannequin in the room.

It has been hypothesized that, although the location of the stationary front is located below the breathing zone, the air quality should still be better with this type of system. The argument for this has been that a part of the air inhaled is taken from the boundary layer flow surrounding the person. This boundary layer flow is fed from below by clean air. The surmise has been that the contaminant concentration in the inhaled air therefore should be lower than in the ambient. However, the measurements of the temperature in the plume presented in Figure 1 contradict this argument. A plume that starts in the clean layer below attains at steady state the same conditions as its ambient.

The stratification in the room is stable because it consists of a layer of light air floating upon a layer of heavier air. However, if the front between the warmer and colder air is disturbed, then waves will propagate along the front. These waves may overturn and break up the stratification. The disturbance may, for example, be set up by people walking in the room. In order to explore this, a special experiment was arranged. At first the room was empty. On the chair in the room a 100 W electric bulb was placed in order to simulate the heat given off by a person. The concentration profile, originating from the usual tracer gas source located 2 m above the floor, was recorded. Then a person opened the door and walked into the room and simultaneously the lamp was turned off. Again the concentration profile was recorded. Finally the person walked out of the room, and the concentration profile was monitored. The result is shown in Figure 9. In the empty room a clear stratification appears. With a person in the room, we obtain a profile similar to that of the empty room. However, the magnitude of the concentrations is larger than before. This may be ascribed to the circumstance that when the door to the room is opened, polluted air from the upper part of the room is dragged down to the lower zone. The concentrations at the height corresponding to the location of the stationary front are extremely high. This must be because the contaminant located below the front is "locked in." It is the same phenomenon as in an atmospheric inversion. When the person walked out of the room, the lamp was turned on again. Now the concentration profile almost recaptures its original shape. However, when the person opens the door again, more contaminated air is dragged down from the upper part of the room. As a result, the concentration levels become even higher than before.

SUMMARY

Tests carried out in an office room showed that large flow rates are needed to attain better air quality in the occupied zone than if the room were ventilated by a standard "complete mixing" system. With a standard reading lamp (100 W) as the only heat source in the room, a minimum flow rate of 120 m^3/h (3 room volumes/h) was needed to attain better air quality in the occupied zone than would be attained if the room were ventilated by a standard "mixing system." Therefore, the use of ventilation by displacement does not mean that any savings due to a lower airflow rate can be achieved.

The stratification is not destroyed by a person walking into the room. In unfavorable conditions, contaminants may be "locked in" beneath the stratification, which means that it takes a long time to evacuate these contaminants. Temperature and concentration profiles are not alike. The temperature profile is smoother and does not reveal the concentration stratification. The comfort criterion of a maximum vertical temperature gradient equal to 3 K/m puts a constraint upon the maximum possible cooling load, equal to about 25 W/m^2 floor area. In order to avoid complaints of thermal discomfort due to high velocities at relatively low temperatures, supply air terminals that spread the air over a large sector must be used.

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