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VENTILATION EFFICIENCY IN AN OCCUPIED OFFICE WITH DISPLACEMENT VENTILATION — A LABORATORY STUDY

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A number of room-ventilation design principles are in use. Common practice in offices is to use mixing ventilation. Increasingly, however, displacement ventilation is being applied; ideally, this involves fresh air displacing contaminated air without mixing. There is very little quantitative documentation of actual improvements in air quality resulting from the installation of such ventilation systems. By means of a chamber study (an occupied office), the performance of displacement vs. mixing ventilation was evaluated in terms of exposure to a simulated (tracer gas) body odor. Flow patterns of air and body odor were evaluated using a signal-response tracer gas technique. The mean age of air was split into two parts: a transit and a presence time, respectively. In case of a "smelling" occupant walking around in the office displacement ventilation improved air quality by a factor of 2 as compared to mixing ventilation at an air change rate of 4.3 h^{-1}. At a higher air change rate (7.5 h^{-1}), the improvement came to a factor of 2.7. Displacement ventilation is always at least as good as mixing ventilation or better; however, it performs better with higher air exchange rates which may exceed those specified in accepted ventilation standards.

INTRODUCTION

A main objective of general mechanical ventilation is to provide for acceptable air quality in the zone of occupancy. In this respect, the performance depends on the ventilation design principle. In recent years, office ventilation systems using vertical displacement air flow have been installed at a growing rate (Holmberg et al. 1990). From the displacement design principle, cool air fills the room from below, and "old" air is displaced upwards. The air flow pattern in the room is assisted by convective currents from any heat source, for instance, the occupants. Contaminants released from heat sources may be entrained in the convective plumes (Breum et al. 1990a). The contaminants could, for instance, be body odor from occupants. If convective upcurrents leaving the occupied zone are not balanced by supply of incoming air and high level extract, a layer (a "front") of heated and contaminated air at the ceiling starts to descend. The front stops where the air flow rate of the convective upcurrents equals the supplied air flow rate (Sandberg and Blomqvist 1989). In the upper flow region, recirculation of air diminishes any spatial non-uniform distribution of contaminants. In the lower flow region a non-uniform distribution is sustained by the displacement flow (Breum and Skotte 1991).

The level of the front in a room is an important air quality parameter. As a design goal, the front should be located above the zone of occupancy. Another important aspect is stability of the stratification. A study of tobacco smoke flow fields in an undisturbed room with heat sources mentioned that occupants may cause a disorganized stratification (Nickel 1990). A study on flow fields of a contaminant (tracer gas) released from a source in an undisturbed room with a heat source reported that the mixing process of air and tracer gas in the upper flow region was influenced by people entering the room, but no disorganized stratification was observed (Sandberg and Blomqvist 1989). No data seem available on stability of the stratification in a room occupied by people having a normal activity level. The primary objectives of this study were to supply data on flow fields of air and contaminants in an experimental office with seated occupants doing paper work or using a personal computer. Comprehensive studies are available on thermal environment related to displacement ventilation (Wyon and Sandberg 1990; Melikov and Nielsen 1989). Therefore, thermal comfort aspects were not included in this study. Dissatisfaction caused by body odor is well established in recent research (Fanger 1987). For the present study, a simulated body odor (tracer gas) was selected as a contaminant. The theoretical models used for characterizing the flow fields of air and contaminants are summarized first.

FLOW FIELD CHARACTERIZATION

Flow fields of air and contaminants are usually very complex, involving turbulence, so that a detailed description is extremely difficult or even impossible and experimental methods characterizing average behavior have to be used. A contaminant source may have its own momentum flux creating its own flow pattern. Consequently, flow fields of air are generally not identical to the flow fields of contaminants. It is therefore necessary to characterize flow patterns of air and contaminants separately (Breum et al. 1990b). The theoretical models used for characterizing the flow fields are summarized below. A mechanically ventilated room (volume V m^3) with one supply duct (flow rate Qs $m^3\ min^{-1}$) and one exhaust duct (flow rate Qe $m^3\ min^{-1}$) is considered. A balanced system with a constant flow rate Q is assumed, i.e., $Q=Q_s=Q_e$. The nominal air change rate is denoted n, and n=Q/V. The nominal time constant of the ventilation process is τ_n, and $\tau_n=V/Q$.

Concepts of age analysis

Consider a fluid element of air or contaminant entering a room. A transit time is needed by the fluid element to be locally felt at a small volume centered at an arbitrary point p within the room (Gardin and Fontaine 1990). For clarification, the transit time is shown in Fig. 1. for the case that fluid elements (from t=0) enter the room at a constant rate. Let the mean transit time be denoted τ_p. The fluid element is locally present for some time and then leave. Let the mean presence time for all fluid elements felt at p be denoted δ_p. Note that the reciprocal of the presence time represents the local air renewal rate (Gardin and Fontaine 1990). The age of a fluid element at p is the time that has elapsed since it entered the room. Let the mean age be denoted μ_p. Mean transit time, mean presence time, and mean age are related by

$$\mu_p = \tau_p + \delta_p \tag{1}$$

It is noted that the mean transit time and the mean presence time depend on flow conditions and can not be predicted beforehand.

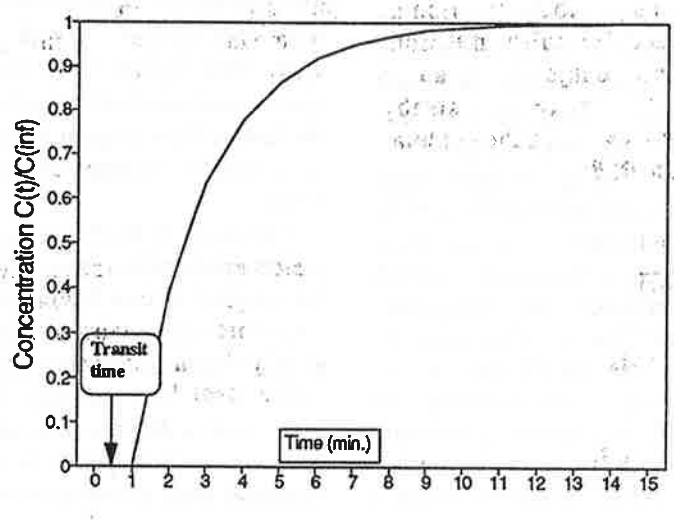

Fig. 1. Transit time needed by a fluid element to be locally felt.

Three different populations of fluid elements in the room may be defined (Sandberg and Sjöberg, 1983): (1) the total population of all fluid elements within the room, (2) a local population of all fluid elements within a small volume centered at an arbitrary point p, and (3) the population of fluid elements leaving the room. For each of the populations mentioned, there is a cumulative age distribution F(t), which is the fraction of the fluid elements of an age less than or equal to t. F(t) is defined over (0, ∞) so that F(0)=0 and F(∞)=1. The corresponding age frequency distribution f(t) is derived as

$$f(t) = \frac{dF(t)}{dt} \tag{2}$$

or

$$F(t) = \int_0^t f(t)\,dt$$

The mean of the distribution is μ, where

$$\mu = \int_0^\infty t\,f(t)\,dt = \int_0^\infty (1-F(t))\,dt \tag{3}$$

The age distribution can be determined experimentally using a signal-response technique: the signal being the injection of tracer gas or a contaminant and the response the measured concentration. The age distribution of fresh air delivered to the room from a duct may be obtained from labelling the air with a tracer gas injected at a point located in the duct. Three different injection strategies are widely used: (1) decay ("step-down"), (2) continuous injection at a constant rate ("step-up"), and (3) pulse injection. In the present study, the continuous injection strategy was used. Let the injection begin at t=0. At steady state, the concentration at p is $C_p(\infty)$, and the cumulative age distribution is (Breum 1988)

$$F_p(t) = \frac{C_p(t)}{C_p(\infty)} \tag{4}$$

From Eq. 3, the mean age μ_p is

$$\mu_p = \int_0^\infty (1-\frac{C_p(t)}{C_p(\infty)})\,dt \tag{5}$$

If p is at the exhaust, the population of fluid elements leaving the room is also described by Eq. 5. The mean age of fluid elements leaving the room is called the residence time. It is noted that the residence time of air leaving the room is equal to the nominal time constant of the ventilation process (Skåret 1986). Let concentration at the exhaust be denoted $C_e(t)$. The age frequency distribution of all fluid elements within the room is (Breum 1988)

$$f(t) = \frac{1-\frac{C_e(t)}{C_e(\infty)}}{\int_0^\infty (1-\frac{C_e(t)}{C_e(\infty)})\,dt} \tag{6}$$

The mean age of all fluid elements within the room is denoted $<\mu>$. From Equation 3 the mean age of all fluid elements within the room is

$$<\mu> = \frac{\int_0^\infty t\,(1-\frac{C_e(t)}{C_e(\infty)})\,dt}{\int_0^\infty (1-\frac{C_e(t)}{C_e(\infty)})\,dt} \tag{7}$$

Flow fields of fresh air

As a design goal for an efficient ventilation scheme of the zone of occupancy, the local mean age of air should be low as compared to the mean age of exhausted air. By definition (Skåret 1986), the displacement flow pattern is considered most efficient for exchange of air within a room. If the air flow pattern of the room investigated were like plug-flow, the mean age of air within the room would be "low". Let this mean age be denoted $<\mu_d>$. Then

$$<\mu_d> = \frac{V}{2Q} \tag{8}$$

However, the actual flow pattern of the room investigated may deviate from the defined ideal. Therefore, the mean age of air in the room would be "high". The air exchange efficiency, β, is by definition (Skåret 1986)

$$\beta = \frac{<\mu_d>}{<\mu>} = \frac{V}{2Q<\mu>} \tag{9}$$

It is noted that $0<\beta<1.0$. An air exchange efficiency of $\beta=1.0$ is achieved for ideal displacement flow, complete mixing is characterized by an efficiency of 0.5, while stagnant flow yields $\beta<0.5$.

Flow fields of contaminated air

The ventilation effectiveness, ε is a common indicator for the contaminant removal performance of a ventilation system. Let at steady-state the room average contaminant concentration be denoted $<C(\infty)>$. Assuming that no contaminants are delivered by the fresh air supply, the ventilation effectiveness as a room average is defined as (Breum et al. 1990b)

$$\varepsilon = \frac{C_e(\infty)}{<C(\infty)>} \tag{10}$$

At local level, the steady-state ventilation effectiveness, ε_p, is defined as

$$\varepsilon_p = \frac{C_e(\infty)}{C_p(\infty)} \tag{11}$$

It is noted that a measured local ventilation effectiveness depends heavily on the location of the measurement point. A high effectiveness is expected to be measured near an air supply inlet. For complete mixing, ventilation effectiveness is 1.0. By short-circuiting, the supply air is poorly utilized and ventilation effectiveness is less than 1.0. As a design goal for a desirable ventilation scheme, the local ventilation effectiveness for the zone of occupancy should be high.

METHODS

Description of the experimental office

The tests were performed in a laboratory using a mock-up of an office room (Fig. 2). In all tests air with a negative buoyancy was supplied through a low velocity air supply terminal device standing on the floor in a corner of the room. Air supply rate was measured by an orifice plate. Air from the room was exhausted through a grille mounted at center of the ceiling. Exhausted air flow rate was measured by an orifice plate. Two chairs (A and B) facing each other were located at a table in the room. A test rig (described below) was fixed on chair No. B. A reading lamp (60 W electric bulb) and a personal computer were standing on the table. The monitor was standing on top of the fan-cooled processing unit. Power required by the monitor and the processing unit was 80 W and 150 W, respectively.

Experimental procedure

Keeping a balanced ventilation process, tests were carried out at three different nominal air change rates at three different conditions of occupancy. In each test, the supply air temperature was set so that temperature at center of the room would be as close as possible to temperature in the laboratory. This was done in order to reduce convective boundary layer air flow at the walls caused by transfer of heat to or from the test room. Before each test, sufficient time was allowed to achieve steady-state conditions. Table 1 shows a summary of test conditions. Note that estimated convective heat loads of the office are listed in the table. During test periods, occupants were seated doing unrestricted work.

Tracer gas (SF_6) was injected from a gas storage bag. Its flow rate was maintained constant at 0.1 cm^3 min^{-1}

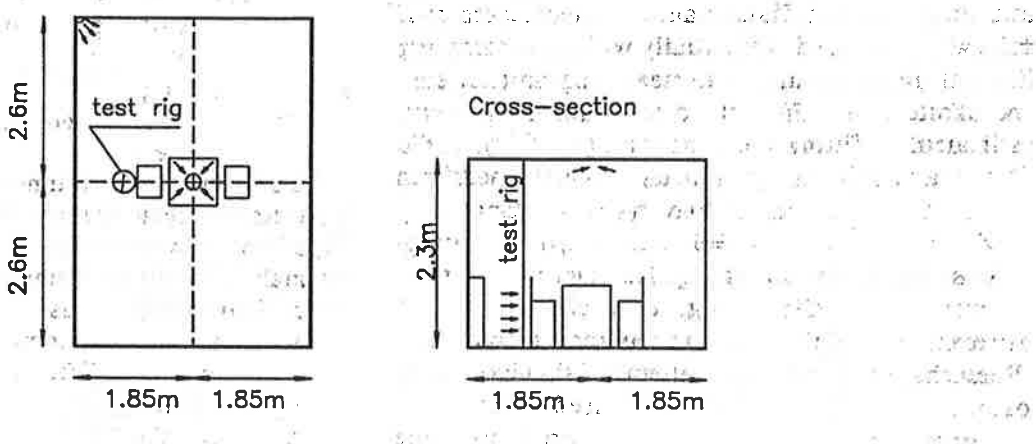

Fig. 2. Layout and cross-section of the test room (not drawn to scale).

Table 1. Estimated air exchange efficiency.

			Nominal air change rate (h^{-1})		
			1.4	4.3	7.5
Occupancy	Test condition	Heat load* (W)	Air exchange efficiency (%)		
1 seated person	A	160	59	65	58
1 seated person using a personal computer	B	370	55	66	62
2 seated persons. 1 person using a personal computer	C	470	57	64	69

* the convective heat load of a seated person was estimated to be 100 W (Nickel 1990).

by means of a gas-tight pump with an estimated control accuracy of ±3%. When measuring the mean age of fresh air, the tracer was injected into the supply duct at a distance from the inlet of more than 80 times the duct diameter. When it enters the room, the tracer may be considered homogeneously mixed with the supply air (Presser and Becker 1988). Tracer gas concentrations were recorded at a test rig fixed at chair No. B. Data were obtained at the following levels (given in m) above the floor: 0.05, 0.40, 0.75, 1.10, 1.45, 1.80, and 2.05. Data were also obtained at the exhaust duct. Data at an estimated accuracy of ±5% were collected sequentially with a 9-s sampling interval using a multipoint measuring unit (Breum and Skotte 1991). From the data obtained, $F_p(t)$ was estimated by fitting the function $a(1-e^{-bt+c})$ to the data obtained. By integration, μ_p was estimated from Eq. 5. The mean transit time, τ_p, was estimated by solving the equation $F_p(t)=0$, i.e., $\tau_p=c/b$. Finally, ς_p was estimated by solving Eq. 1. Room mean age of air was estimated by integration (Eq. 7), and the air exchange efficiency was obtained from Eq. 8. When characterizing flow patterns of simulated body odor, an identical equipment was used for injecting tracer gas from a point (the navel) on skin of the person seated in chair No. A. When steady state conditions were achieved, the person started to walk around in the office randomly. The person kept walking at normal speed until steady-state conditions were achieved. Local ventilation effectiveness was estimated (Eq. 11) from the data obtained.

RESULTS

Tests were performed at three different nominal air change rates. The estimated air exchange efficiencies are listed in Table 1. For clarification, estimated parameters characterizing the air renewal process at local level are only given for one (4.3 h^{-1}) of the air change rates investigated. However, these data may be considered representative. Figure 3 shows the estimated local mean age of air normalized with a reference to the residence time. Figure 4 shows the estimated mean transit time normalized with a reference to the mean transit time of air at the exhaust. Figure 5 shows the estimated mean presence time normalized with a reference to the mean presence time of air at the exhaust.

The flow field of a simulated body odor was characterized at two different activity levels of the occupant emitting the "odor". Figure 6 shows the reciprocal of the local ventilation effectiveness in case of a seated person being the contaminant source.

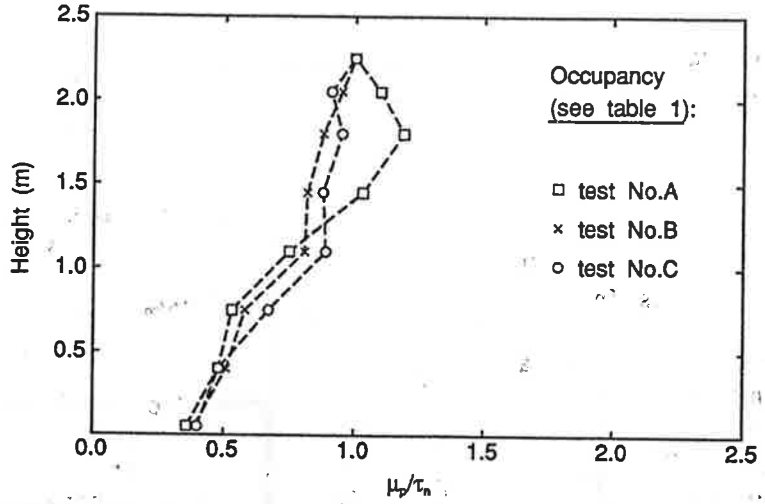

Fig. 3. Vertical distribution of the local mean age of air, μ_p, normalized with a reference to the nominal time constant, τ_n, of the ventilation process.

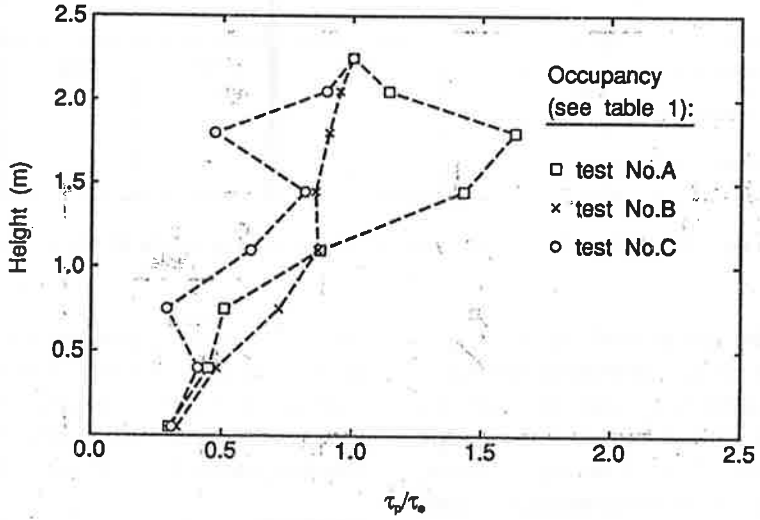

Fig. 4. Vertical distribution of the local mean transit time of air, τ_p, normalized with a reference to the mean transit time of exhausted air, τ_e.

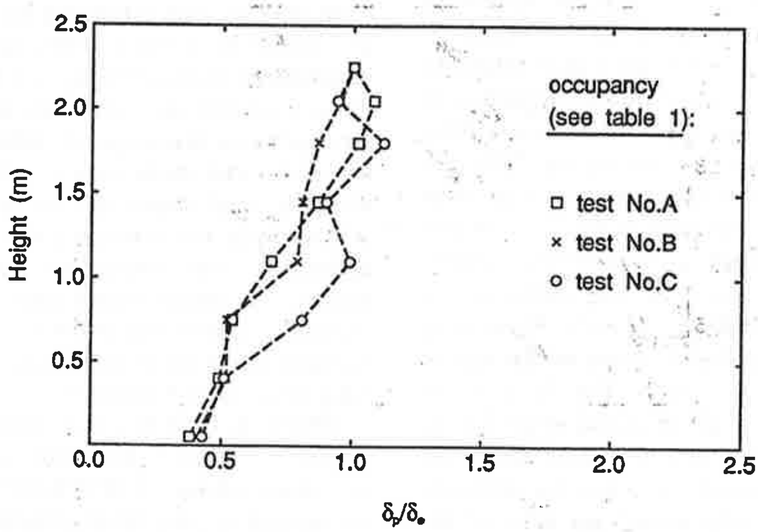

Fig. 5. Vertical distribution of the local mean presence time of air, δ_p, normalized with a reference to the mean presence time of exhausted air, δ_e.

Figure 7 shows the reciprocal of the ventilation effectiveness in case of a walking person being the contaminant source.

DISCUSSION

Several standards have been set for ventilation and fresh air supply rates to offices. In the USA, the American Society of Heating, Refrigeration and Air Conditioning Engineers has set a standard which will require 36 $m^3 h^{-1}$ per person (ASHRAE 1989). Following this standard, the present office occupied by two persons will require an air change rate of 1.6 h^{-1}. This study included air change rates of 1.4 h^{-1}, 4.3 h^{-1}, and 7.5 h^{-1}.

Qualitatively, the pattern of air flow in a room can vary from one extreme (short circuiting) to the other (displacement flow). Between these, there is perfect mixing. The estimated air exchange efficiencies (Table 1) indicated a tendency towards ventilation by displacement. For comparison a field study of office buildings ventilated from the mixing design principle reported air exchange efficiencies ranging from 36% to 57% (Majanen et al. 1987).

As convective heat emission from a source is increased, the generated convective air flow rate is increased by a factor of the cubic root of the increase in heat emission (Nielsen 1988). Consequently, at constant air supply rate the level of the front above floor is decreased as convective heat emission is increased. From the local mean age of the air (Fig. 3),

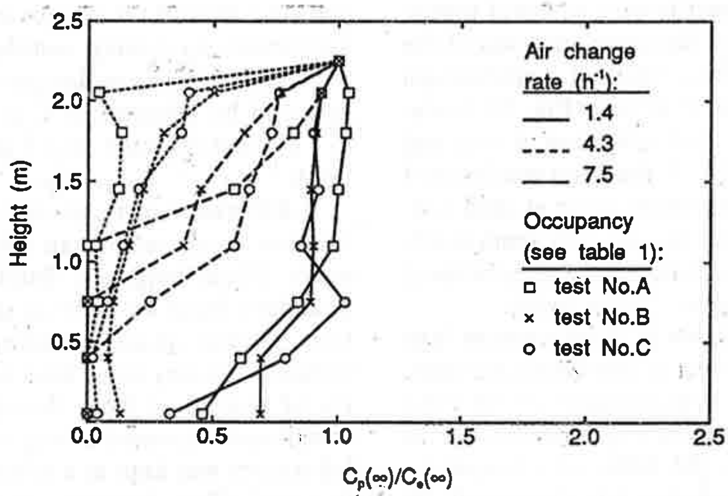

Fig. 6. Vertical distribution of the reciprocal of the local ventilation effectiveness, ε_p, in case of seated occupants.

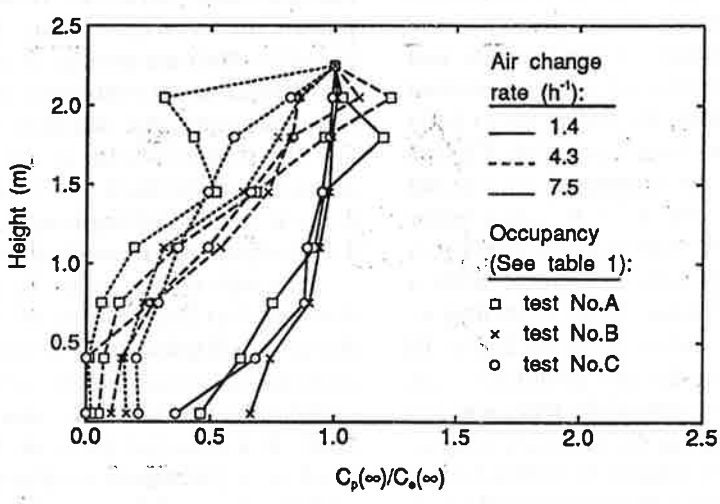

Fig. 7. Vertical distribution of the reciprocal of the local ventilation effectiveness, ε_p, in case of an occupant walking around in the room.

the location of the front depended on heat load conditions. From Fig. 3, the location at condition No. A was not obvious. At conditions B or C, the front was located between 0.75 m and 1.10 m above floor. However, the spatial resolution in data was insufficient to observe any difference between those two heat load conditions. It is noted that at a level of 1.0 m above floor, the convective air flow rate generated by an occupant may range from 20 m^3 h^{-1} to 70 m^3 h^{-1} depending on the vertical gradient in air temperature (Fitzner 1989).

In general, it is desirable that supply air flows rapidly towards the zone of occupancy and that its renewal rate is large. In other words, for the zone of occupancy, the transit time as well as the presence time should be low as compared to the nominal time constant of the ventilation process (Gardin and Fontaine 1990). A vertical stratification of local transit time was observed (Fig. 4). However, location of the front was not obvious. Also a vertical stratification of local presence time was observed (Fig. 5). Location of the front at heat load condition A was not obvious. Consistent with Fig. 3, the front was located between 0.75 m and 1.10 m above floor at load conditions B and C. From literature, no data seem available for comparison of the obtained estimates of local transit time and presence time, respectively.

The basic idea of ventilation by displacement is to achieve supply air conditions in the occupied zone. Therefore, the level of stratification is of vital importance. As observed from Fig. 6, exposure to simulated body odor in the zone of occupancy increased as nominal air change rate decreased. At an air change rate of 1.4 h^{-1}, air quality at levels exceeding 0.75 m above floor was as if the room were ventilated from the mixing design principle. Air quality was improved as nominal air change rate was increased. Considering the breathing zone of a seated person to be 1.1 m above floor, the exposure to body odor caused by a colleague (test condition C) was reduced by a factor of 7 as compared to mixing ventilation at an air change rate of 7.5 h^{-1}. At a lower air change rate (4.3 h^{-1}), the improvement came to a factor of 1.7. This finding was consistent with a previous chamber study (Fitzner 1988) reporting an improvement in air quality with a factor of 2.5 to 10 by changing the ventilation design principle from mixing to displacement. Location of the front was not obvious in all experiments (Fig. 6). At an air change rate of 1.4 h^{-1}, the front was located between 0.40 m and 0.75 m above floor. At an air change rate of 7.5 h^{-1}, the front was located more than 2.05 m above floor. The location of the front was not obvious at an air change rate of 4.3 h^{-1}. However, from the local mean age of the air (Fig. 3), the front was located between 0.75 m and 1.10 m above floor at an air change rate of 4.3 h^{-1} (test condition B and C).

From time to time, occupants walk around in a real office. As observed by comparing Fig. 7 with Fig. 6, a walking occupant may cause increased mixing in a room. This finding was consistent with a previous study reporting that occupants even at a low activity level may disorganize the thermal stratification (Nickel 1990). Another chamber study reported that people entering an undisturbed room caused an increased contaminant concentration in the zone of occupancy (Sandberg and Blomqvist 1989). A substantial improvement in air quality was achieved by increasing the air change rate from 4.3 h^{-1} to 7.5 h^{-1} (Fig. 6). However, in case of a "smelling" occupant walking around the improvement became less. With a reference to mixing ventilation, exposure to body odor caused by a colleague (test condition C) was reduced by a factor of 2 at an air change rate of 4.3 h^{-1} and a factor of 2.7 at an air change rate of 7.5 h^{-1}.

A difference in temperature between air and surfaces set up convective air flows. A previous chamber study (Heiselberg and Sandberg 1990) noted that boundary layer air flow at the walls may influence flow patterns of air and contaminants in a room. To reduce boundary layer flow at walls caused by transfer of heat to or from the office, the difference in temperature between center of the test office and the laboratory was kept at a minimum (less than 2-3°C). In a real office, boundary layer flows may, to an yet unknown extent, influence flow patterns of air and contaminants. However, from the present study displacement ventilation may have potential for improved indoor air quality in offices. It is emphasized that displacement ventilation may call for an air change rate exceeding that required by accepted standards. For the present study, an air change rate exceeding an accepted standard (ASHRAE 1989) by a factor of 2.7 was needed to improve air quality by a factor of 1.7 as compared to mixing ventilation.

Although not included in the presnet study, thermal comfort aspects are of vital importance in the design of a displacement ventilation system. Air supply terminals located near the floor create nonuniformity in the velocity and temperature fields of the occupied zone. In a chamber study on thermal environment related to displacement ventilation, Wyon and Sandberg (1990) observed for a wide range of heat loads that thermal conditions at floor level may cause discomfort due to cold feet, ankels, and legs. However, in a large field study, Melikov and Nielsen (1989) found that well designed displacement ventilation systems may create thermal comfort in rooms, and, in recent years, office ventilation systems using displacement air flow have been installed at a growing rate in Scandinavia (Holmberg et al. 1990).

CONCLUSIONS

The study showed that in terms of reducing exposure to body odor in an occupied office, the performance of displacement ventilation may exceed that of mixing ventilation. However, a high nominal air change rate was needed to "lift" the front to a level above the zone of occupancy. At an air change rate of 7.5 h^{-1}, exposure of a seated occupant to a simulated body odor from a seated colleague was reduced by a factor of 7 as compared to mixing ventilation. At a lower air change rate (4.3 h^{-1}), the improvement came to a factor of 1.7.

Occupants walking around in an office ventilated from the displacement design principle may cause increased mixing in the room. Even so, displacement ventilation provided an improved air quality as compared to mixing ventilation. In case of a "smelling" occupant walking around in the office, the improvement came to a factor of 2 at an air change rate of 4.3 h^{-1} and 2.7 at an air change rate of 7.5 h^{-1}.

From the present chamber study, displacement ventilation may have potential for improving air quality in offices. Displacement ventilation is always at least as good as mixing ventilation; however, it performs better with higher air change rates which may exceed those specified in accepted ventilation standards. Flow patterns of air and contaminants in real offices may be different from those in a chamber study. Therefore, intervention studies in the field are needed.

REFERENCES

ASHRAE (American Society of Heating, Refrigeration and Air Conditioning Engineers). Ventilation for acceptable indoor air quality. ASHRAE 62-1989. Atlanta, GA: American Society of Heating, Refrigeration and Air Conditioning Engineers; 1989.

Breum, N.O. The air exchange efficiency of a lecture hall. In: Vincent, J.H. ed. Ventilation '88. Oxford: Pergamon Press; 1988: 373-380.

Breum, N.O.; Soehrich, E.; Lund Madsen, T. Differences in organic vapor concentrations in the breathing zone resulting from convective transport from spillage on clothing. Appl. Occup. Environ. Hyg. 5: 298-302; 1990a.

Breum, N.O.; Takei, H.; Rom, H.B. Upward vs. downward ventilation air flow in a swine house. ASEA Trans. 35(5): 1693-1699; 1990b.

Breum, N.O.; Skotte, J. Displacement air flow in a printing plant measured with a rapid response tracer gas system. Building Serv. Res. Technol. 12: 39-43; 1991.

Fitzner, K. Impulsarme Luftzufuhr durch Quellüftung. Heizung, Lüftung Klima Haustech. 39: 173-181; 1988.

Fitzner, K. Förderprofil einer Wärmequelle bei verschiedenen Temperaturgradienten und der Einfluss auf die Raumströmung bei Quellüftung. Klima-Kälte-Heizung 10: 476-481; 1989.

Fanger, P.O. A solution to the sick building mystery. In: Seifert, B, Esdorn, H.; Fischer, M.; Rüden, H.; Wegner, J., eds. Proc. 4th international conference on indoor air quality and climate. Vol. 4. Berlin (West): Institute for Water, Soil and Air Hygiene; 1987: 49-55.

Gardin, P.; Fontaine, J.R. General ventilation characterization. In: Fanger, P.O.; Strindehag, O., eds. Proc. Roomvent '90. Oslo: Norsk VVS; 1990.

Heiselberg, P.; Sandberg, M. Convection from a slender cylinder in a ventilated room. In: Fanger, P.O.; Strindehag, O., eds. Proc. Roomvent '90. Oslo: Norsk VVS; 1990.

Holmberg, R.B., Eliasson, L., Folkesson, K.; Strindehag, O. Inhalation-zone air quality provided by displacement ventilation. In: Fanger, P.O.; Strindehag, O., eds. Proc. Roomvent '90. Oslo: Norsk VVS; 1990.

Majanen, A.; Helenius, T.; Seppanen, O.; Roos, R. Air exchange efficiency in residential and office buildings. In: Fanger, P.O.; Strindehag, O., eds. Proc. Roomvent '87. Stockholm: Svensk VVS; 1987.

Melikov, A.K.; Nielsen, J.B. Local discomfort due to draft and vertical temperature difference in rooms with displacement ventilation. ASHRAE Trans. 95(2): 1050-1057; 1989.

Nickel, J. Air quality in a conference room with tobacco smoking ventilated with mixed or displacement ventilation. Fanger, P.O.; Strindehag, O. Proc. Roomvent '90. Oslo: Norsk VVS; 1990.

Nielsen, P.V. Displacement ventilation in a room with low-level diffusors. In: Kälte-Klima-Tagung, 1988. Munich: Deutcher Kälte- und Klimatechnischer Verein, e.V.; 1988.

Presser, K.H.; Becker, R. Mit Lachgas dem Luftstrom auf der Spur. Heizung, Lüftung Klima Haustech. 39: 7-14; 1988.

Sandberg, M.; Sjöberg, M. The use of moments for assessing air quality in ventilated rooms. Build. Environ. 18: 181-197; 1983.

Sandberg, M.; Blomqvist, C. Displacement ventilation systems in office rooms. ASHRAE Trans. 95(2): 1041-1049; 1989.

Skåret, E. Contaminant removal performance in terms of ventilation effectiveness. Environ. Int. 12: 419-427; 1986.

Wyon, D.P.; Sandberg, M. Thermal manikin prediction of discomfort due to displacement ventilation. ASHRAE Trans. 96(1): 67-75; 1990.