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# Displacement Ventilation in Industry—a Design Principle for Improved Air Quality

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*There is very little quantitative documentation of actual improvements resulting from the installation of new general ventilation systems in industrial settings. Therefore the performance of the common mixing design principle was compared to the displacement design principle by means of an intervention study in a workshop (V = 12,000 $m^3$), where thermoplastics were moulded. An experimental signal-response tracer gas technique was used. In terms of supplying fresh air to the zone of occupancy the displacement system was better than the mixing system by a factor of 2. In terms of the exposure level to a simulated contaminant (tracer gas) the displacement system was better by a factor* 1.5–18.

## INTRODUCTION

GENERAL room ventilation systems have traditionally been designed to provide for acceptable air quality and thermal comfort in the zone of occupancy under the assumption that the supplied air is perfectly mixed in a room. The advent of displacement ventilation systems have brought the usefulness of the perfect mixing assumption under question [1]. The features of the two design principles are summarized below.

From the mixing design principle air jets often are supplied at ceiling level with a high momentum. Room air is entrained into the jets so generating secondary recirculating air flows in the room. This mixing process diminishes a spatial non-uniform air temperature and contaminant distribution in the room. The supply air is the main source of momentum, and therefore mixing ventilation may be called high momentum ventilation [2].

From the displacement design principle cool air is supplied with a low momentum through large inlet devices near the floor. The cool air is heated by heat sources in the room, and convective plumes are formed above the heat sources. Often contaminants released from the heat sources are entrained in the convective plumes. The ventilation air is both a source of momentum and buoyancy. Therefore displacement ventilation may be called low momentum ventilation [2]. If the convective upcurrents leaving the occupied zone are not balanced by the supply of 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 [2]. The level of the front in a room is an important air quality parameter [3]. As a design goal the front should be located above the zone of occupancy.

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Displacement flow systems are now increasingly replacing the traditional mixing flow systems in industrial buildings. In Scandinavia the present market share of displacement systems for industrial ventilation was estimated to be 50% [4]. Using mixing flow systems as a reference it is of interest to validate the potential of displacement systems for improved air quality. Data on air flow fields of mixing versus displacement systems are available from the laboratory [2] and from industry [3, 5]. However, flow fields of air are generally not identical to the flow fields of contaminants [6]. Therefore an intervention study of air and contaminant flow fields of mixing versus displacement ventilation has been made in a workshop where thermoplastics were converted into products by injection moulding. A complex mixture of air contaminants may evolve from thermally stressed thermoplastics [7]. For the present study a simulated mixture (tracer gas) was selected as a contaminant. Theory on which the present study is based (age analysis) is summarized first.

## MATERIALS AND METHODS

A contaminant source may have its own momentum flux creating its own flow pattern. Consequently, the flow pattern of contaminants usually differ from those of fresh air. Flow fields of air and contaminants in a confined space are usually very complex, involving turbulence, so that a detailed description is extremely difficult. However, concepts of age distribution theory provide useful tools for the quantification of flow fields [5]. Fluid elements of air or contaminants entering a room remain in it for some time and then leave: their age is equal to the time spent in the room. Three different populations of fluid elements can be distinguished [8]:

- total population of all fluid elements of air within the room,

- local population of fluid elements of air at an arbitrary point p within the room, and
- population of fluid elements of air leaving the room.

Each population may be characterized by their cumulative age distribution, which is, for any $t$, the fraction of fluid elements with an age less than or equal to $t$. $F(t)$ is defined over $(0, \infty)$ so that $F(0) = 0$ and $F(\infty) = 1$. The corresponding age frequency distribution $f(t)$ is derived as:

$$f(t) = \frac{\mathrm{d}F(t)}{\mathrm{d}t} \quad \text{or} \quad F(t) = \int_0^t f(t^*)\,\mathrm{d}t^*. \tag{1}$$

The mean of the distribution is $\mu$ where:

$$\mu = \int_0^\infty tf(t)\,\mathrm{d}t = \int_0^\infty [1-F(t)]\,\mathrm{d}t. \tag{2}$$

The age distribution may be experimentally determined using signal-response techniques. The signal is the injection of tracer gas at the air supply duct or at the source of contaminant flow. The responses are the measured tracer gas concentrations either at selected points within the room or at the flow exit. In principle, any type of tracer signal may be used. However, the most common signals in ventilation studies are [8]: decay ("step-down"), continuous injection at a constant rate ("step-up"), and pulse injection. The continuous injection strategy was applied in the present study. The concentration of tracer gas at a position p within the room at time $t$ is denoted $C_p(t)$, and the steady-state concentration is denoted $C_p(\infty)$. The local mean age, $\mu_p$, of the air or the contaminant at a position p is determined from [5]

$$\mu_p = \int_0^\infty tf(t)\,\mathrm{d}t$$

$$= \int_0^\infty [1-F_p(t)]\,\mathrm{d}t = \int_0^\infty \left[1-\frac{C_p(t)}{C_p(\infty)}\right]\mathrm{d}t. \tag{3}$$

The local mean age of air is a very useful parameter in detecting stagnant regions in a room. In stagnant regions the local mean age is elevated compared to well ventilated regions.

Let tracer-gas be injected at a constant rate $q$ (cm$^3$ min$^{-1}$), from $t = 0$, into an air supply duct and mixed homogeneously with the supply air, $Q_m$ (m$^3$ min$^{-1}$), before entering the room. The steady state supply air tracer gas concentration, $C_m$, is

$$C_m = \frac{q}{Q_m}. \tag{4}$$

At a point p within the room some of the supplied fresh air may come from other sources than the air supply duct considered. The proportion, $PR_p$, delivered by $Q_m$ can be estimated from [5]

$$PR_p = \frac{C_p(\infty)}{C_m}. \tag{5}$$

The exhausted air flow rate is denoted $Q_e$.

## DESCRIPTION OF THE WORKSHOP

In the workshop thermoplastics were converted into products by injection moulding. An injection-moulding machine consists of two principal parts, i.e., an injection unit and a clamp unit [9]. The injection unit melts the plastic and injects it into the mould. The clamp unit opens, closes, and holds the mould closed against the pressure of the melt. The size of an injection-moulding machine is described by the capacity of the clamp unit. Most injection units are extruders that are specially built for injection-moulding machines. The size of the injection unit is described by its shot capacity, which is the maximum volume of melt that can be injected in a cycle. A wide range of injection-moulding machines (Nos. 1–16; Fig. 1) were installed in the workshop. The machines were controlled by programmed electronic control systems which also controlled mechanical take-off devices. In terms of the capacity of the clamp units the machines ranged from 15–1200 tonnes. In terms of the short capacity the size of the injection units ranged 50–9000 cm$^3$. There were two shifts, each of 5–10 persons mainly for loading and unloading.

During the first period of the study the room was ventilated from the mixing design principle. The layout and cross-section of the workshop ($V$ = 12,000 m$^3$) are shown in Fig. 1. Fresh air (no recirculation) was supplied at a constant rate by two ducts located 5.0 m above the floor and delivered through grilles at the same level (Fig. 1). Throughout the test period the air supply temperature varied between 20.5 and 20.9°C. For practical reasons it was not possible to obtain the temperature at the exhaust. Based on workshop floor area the heat load generated in the room was approximately 10–15 W m$^{-2}$. The main heat sources were located at the machines. During the study some machines (Nos. 1–5, 7) were on stand-by. Data on the intensity of the individual heat sources were not available. No local exhaust systems were installed at the machines. The heated and contaminated air was exhausted by a duct located 7.5 m above the floor (Fig. 1). The ventilation was designed for an unbalanced system, $Q_e > Q_m$, to keep an inward air flow through two permanently open doors (2.2 × 2.4 m, each) to other heated, departments of the factory.

During the second period of the study the workshop was ventilated from the displacement design principle. The layout and cross-section of the workshop are shown in Fig. 2, the only difference from Fig. 1 being a re-arranged ducting of the ventilation system to serve air terminal devices at floor level. Fresh air (no recirculation) was introduced at a constant rate into the zone of occupancy through large area low velocity air terminal devices (semi-cylindrical: 2.1 m height, 0.4 m dia). Throughout the test period the air supply temperature varied between 15.2 and 16.0°C. Based on workshop floor area the heat load generated in the room was approximately 25–30 W m$^{-2}$. During the study some machines (Nos. 1 and 2) were on stand-by. The ventilation was designed for an unbalanced system to keep an inward air flow through the open doors.

## EXPERIMENTAL PROCEDURE

Several tracer gases have been used in the past for characterizing ventilation processes in buildings [10]. For this study $SF_6$ was chosen as it has desirable tracer gas characteristics, in terms of detectability, safety, and cost,

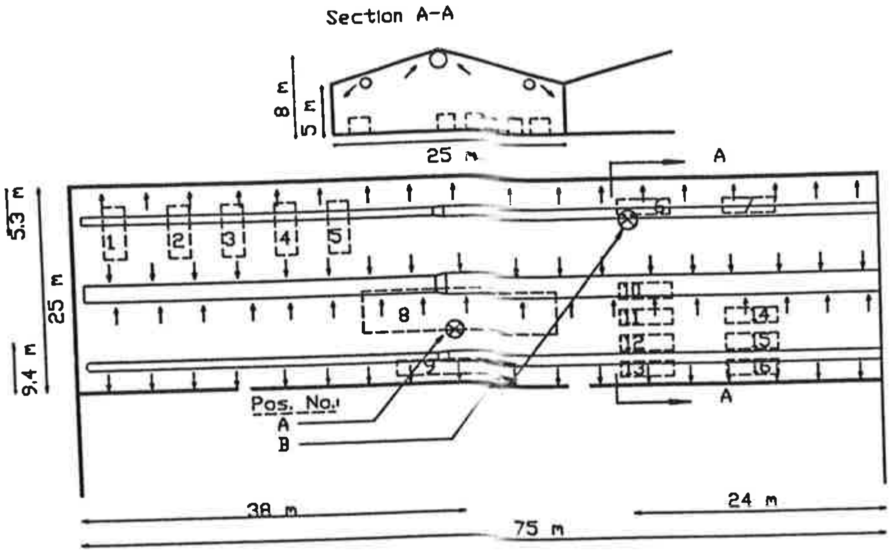


Fig. 1. Layout and cross-section of the workshop designed for mixing ventilation (not to scale).

and has been used successfully in several ventilation studies [2, 3, 5]. The tracer gas was injected from a pressurized bottle at a constant flow rate of 10 $cm^3\ min^{-1}$ controlled by a calibrated rotameter at an estimated accuracy of $\pm 3\%$. When measuring the local mean age of the fresh air, the tracer was injected into the air supply ducts at a distance more than 80 times the duct diameter from an inlet. When it enters the room the tracer may be considered homogeneously mixed with supply air [11].

To simulate a true contaminant by a tracer it is important that the tracer is discharged in a pattern similar to the contaminant origination pattern [12]. The density of the tracer is of importance if un-diluted tracer is injected as a simulated contaminant from a local source [8, 13]. With reference to air the neat $SF_6$ has a relative density of 5.0. To enhance the mixing of the tracer and air a large area device was used for emitting the tracer at a low flow rate (10 $cm^3\ min^{-1}$). The tracer was emitted into the convective plume above the heated barrel (0.15 m dia, 0.[illegible] m long) of the injection unit located at machine No. 11. The surface temperature of the barrel was 65°C.

Tracer gas concentrations and air temperatures were recorded at two test rig positions (Nos. A-B, Fig. 1) in the room. At test rig position A a propeller fan located 4.[illegible] m above the floor was blowing air downwards for cooling of the moulded products. At each test rig data were obtained at the following five different levels (given in m) above the floor: 0.5, 1.5, 2.5, 3.5, and 4.5. Data were also obtained from the air supply ducts. For practical reasons data were not obtained at the exhaust. In the second test period when measuring the concentration of the simulated contaminant data were also collected 2.0 m above the floor. Data at an accuracy of $\pm 5\%$ were collected sequentially with a 9 s sampling interval using a rapid response multipoint measuring system (Fig. 3) [14]. The system, having a detection limit of 0.03 ppb

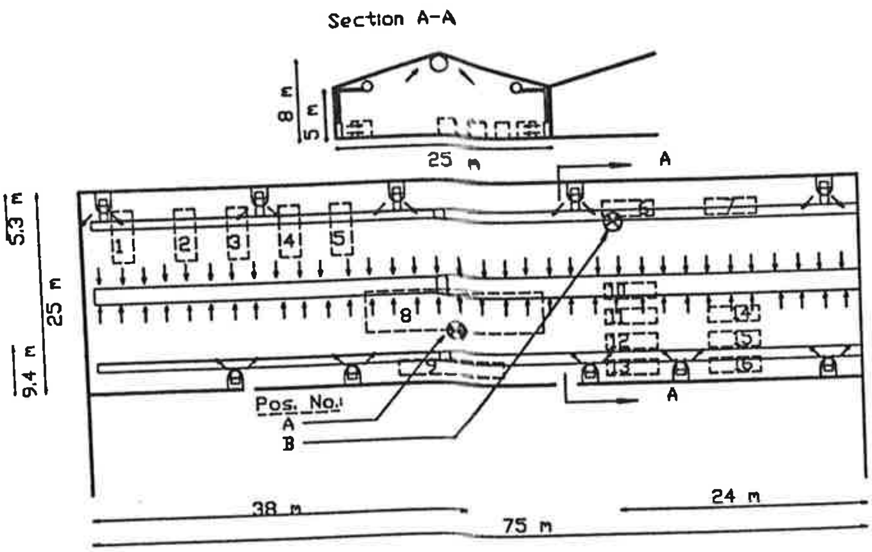


Fig. 2. Layout and cross-section of the workshop designed for displacement ventilation (not to scale).

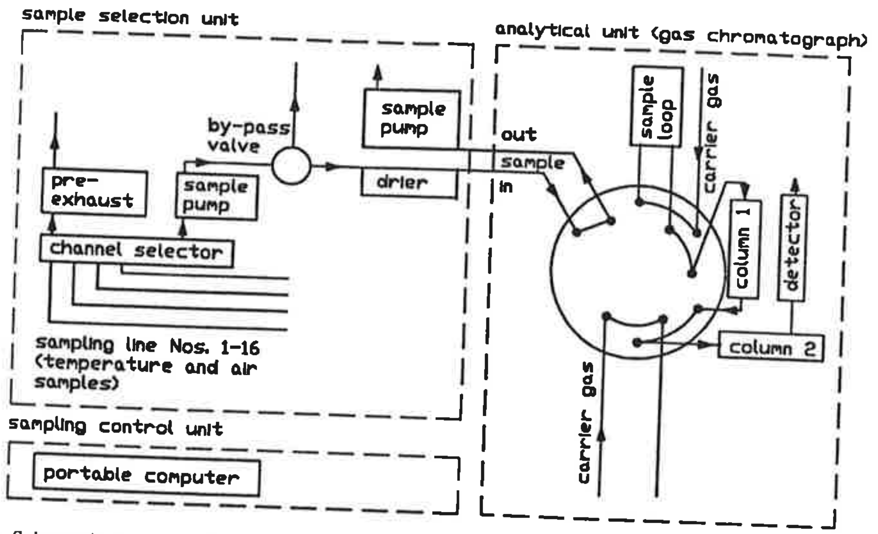

Fig. 3. Schematic diagram of the tracer-gas measuring unit (the gas chromatograph is shown in the load mode).

$SF_6$, was used successfully in a previous ventilation study [14]. However, to allow a more detailed spatial characterization the number of sampling lines was expanded from eight to sixteen. During an experiment the sequence of operations and data acquisition were run by menu-driven software using a portable computer. Data were collected on a hard disk of the computer. The cumulative age frequency distribution, $F_p(t)$, was estimated by an exponential curve-fitting procedure incorporated in the menu-driven software. Then $\mu_p$ was estimated by numerical integration.

The air temperature was measured (Pt 100 transducers) sequentially (9 s step period) using the multipoint measuring system. Based on tests in the laboratory the estimated accuracy was $\pm 0.2°C$. The air velocity was measured using a calibrated temperature compensated omnidirectional low-velocity flow analyser with technical characteristics meeting an accepted standard [15].

## RESULTS

In the first test period when measuring the local mean age of air the constant supply air tracer gas concentration was $C_m = 26.3$ ppb. The estimated air supply rate was $Q_m = q/C_m = 380$ m$^3$ min$^{-1}$. In the second test period when measuring the local mean age of air the constant supply air tracer gas concentration was 26.8 ppb. The estimated air supply rate was $Q_m = q/C_m = 370$ m$^3$ min$^{-1}$. Smoke testing at the open doors confirmed that $Q_e > Q_m$ during both of the test periods.

As an example of the data collected during the first test period tracer gas concentration against time at test rig No. B are shown in Fig. 4. For comparison data collected at the same position during the second test period are shown in Fig. 5. The estimated parameters characterizing the air flow patterns throughout all the experiments are summarized in Table 1. The estimated parameters characterizing the flow fields of the simulated air contaminant are summarized in Table 2. Data characterizing the thermal environment are summarized in Table 3.

## DISCUSSION

Data from the present study were collected under normal conditions of production with no attempt to reduce disturbances caused by moving machines, traffic in or out of the workshop, etc. In the first test period the ventilation was designed for a mixing system. Consistent with previous field studies [3, 5] and with data obtained from an experimental room under isothermal conditions [16] the mixing system succeeded in creating a basically uniform vertical distribution of the local mean age of air, the PR-index, the air temperature and the air velocity (Tables 1 and 3). However, it is noted that the conditions at test rig No. A were influenced by the propeller fan. As observed from Table 1 the circulating air flows were not sufficient on the horizontal level to create a uniform distribution of the local mean age of air. In the zone of occupancy the air temperature complied with the recommended [17] thermal comfort limit (20–24°C) for light, mainly sedentary activity, during summer. In general, air velocities were at (test rig No. B) or exceeded (test rig No. A) the comfort limit (0.15 m s$^{-1}$). The vertical air temperature gradient did not exceed the comfort limit (3°C m$^{-1}$). The spatial distribution of a contaminant and resulting concentration levels are a result of properties of the contaminant itself and the distribution of the supplied air within the room [8]. A vertical stratification of the contaminant concentration was observed only at test rig No. B (Table 2). It is noted that the mean age of the contaminant was low at floor level and high at ceiling level so indicating that the contaminants arrived first in the zone of occupancy, subsequently to rise towards the ceiling. Due to the propeller fan no vertical stratification was observed at rig No. A.

In the second test period the ventilation system was designed for a vertical (upwards) displacement air flow. From the age parameter of air delivered by the air supply terminals, two distinct flow regions of the workshop may be observed (Table 1, test rig No. B) with the front located at a level of 1.5 m to 2.5 m above the floor. Below the front the mean age of the air was 10 min; above the

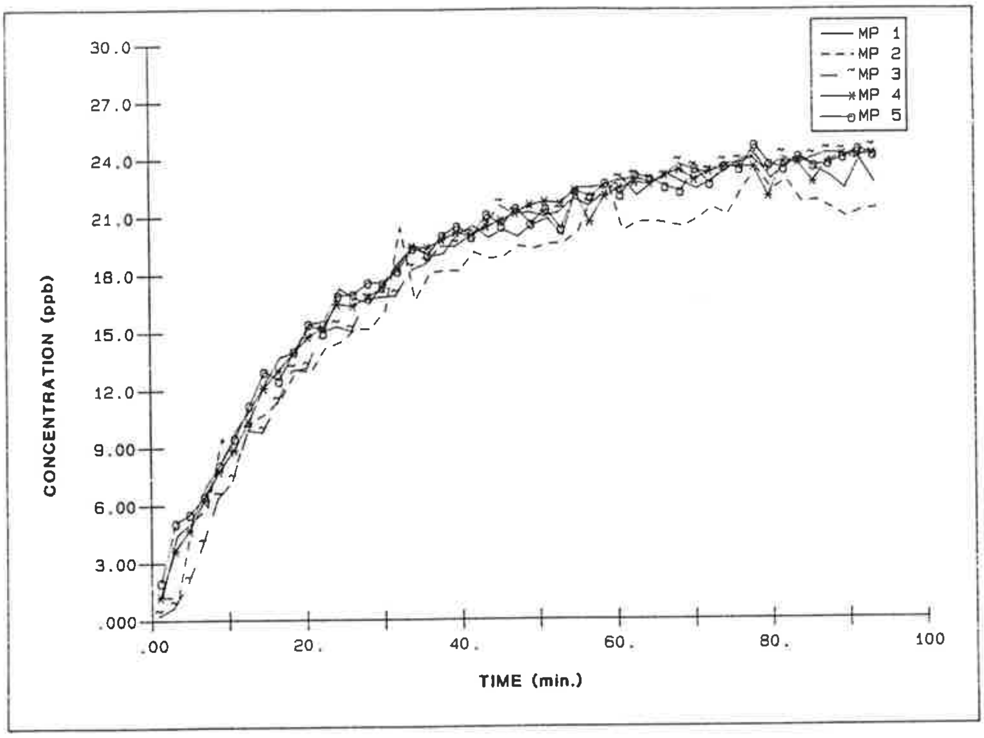


Fig. 4. Tracer gas concentrations during the first test period (mixing ventilation) at test rig No. B. Measuring point (MP) level above the floor: MP1 = 0.5 m, MP2 = 1.5 m, MP3 = 2.5 m, MP4 = 3.5 m, MP5 = 4.5 m.

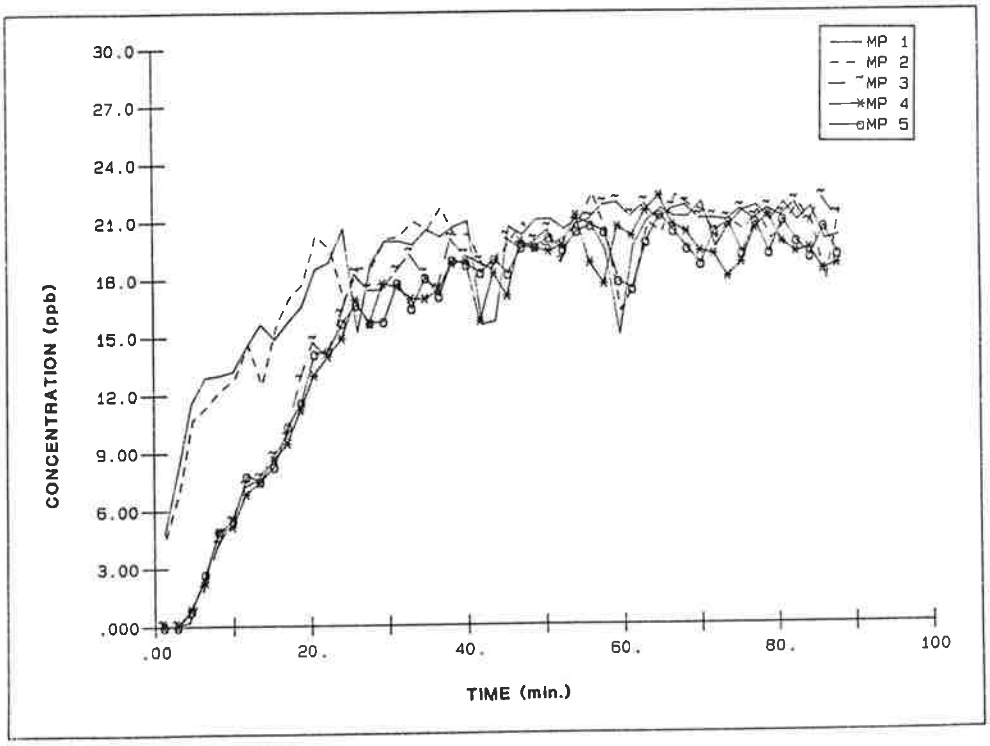


Fig. 5. Tracer gas concentrations during the second test period (displacement ventilation) at test rig No. B. Measuring point (MP) level above the floor: MP1 = 0.5 m, MP2 = 1.5 m, MP3 = 2.5 m, MP4 = 3.5 m, MP5 = 4.5 m.

Table 1. Estimated parameters characterizing the air flow fields

| | | Ventilation principle | | | |
|---|---|---|---|---|---|
| | | Mixing | | Displacement* | |
| Pos. | $H$† (m) | $\mu_p$ (min) | $PR_p$ (%) | $\mu_p$ (min) | $PR_p$ (%) |
| A | 0.5 | 18 | 92 | 22 | 81 |
| | 1.5 | 17 | 92 | 24 | 73 |
| | 2.5 | 18 | 93 | 24 | 73 |
| | 3.5 | 18 | 91 | 25 | 75 |
| | 4.5 | 19 | 81 | 27 | 77 |
| B | 0.5 | 25 | 94 | 10 | 76 |
| | 1.5 | 22 | 92 | 10 | 76 |
| | 2.5 | 22 | 92 | 20 | 76 |
| | 3.5 | 21 | 90 | 20 | 76 |
| | 4.5 | 22 | 84 | 21 | 81 |

* The propeller fan at test rig A was in function during the test period.
† Height above the floor.

Table 3. Data describing the thermal environment

| | | Ventilation principle | | | |
|---|---|---|---|---|---|
| | | Mixing | | Displacement* | |
| Pos. | $H$† (m) | $t$ (°C) | $v$‡ (cm s$^{-1}$) | $t$ (°C) | $v$‡ (cm s$^{-1}$) |
| A | 0.5 | 21.5 | 12 | 22.9 | 21 |
| | 1.5 | 21.5 | 22 | 23.1 | 48 |
| | 2.5 | 22.3 | 16 | 23.1 | 28 |
| | 3.5 | 22.8 | 12 | 23.3 | 10 |
| | 4.5 | 23.8 | 24 | 24.1 | 9 |
| B | 0.5 | 21.4 | 11 | 19.6 | 11 |
| | 1.5 | 21.7 | 14 | 21.4 | 15 |
| | 2.5 | 22.2 | 13 | 22.9 | 9 |
| | 3.5 | 22.3 | 9 | 23.5 | 8 |
| | 4.5 | 22.9 | 20 | 23.3 | 10 |

* The fan propeller at test rig A was in function during the test period.
† Height above the floor.
‡ Mean value of 3 min sampling period.

front the mean age was 20 min. This finding of a vertical stratification is consistent with recent tracer gas studies of displacement ventilation in workshops [5, 18] and in the laboratory [2]. No vertical stratification of the PR-index was observed (Table 1). The measured contaminant concentrations (Table 2) were stratified vertically consistent with the spatial distribution of the local mean age of air. A substantial increased concentration level was observed above the front. This finding is consistent with recent data on the spatial distribution of true contaminants in a workshop ventilated from the same design principle [14]. It is noted that the mean age of the simulated contaminant in the zone of occupancy was elevated compared to the conditions of the upper flow region. This finding, supported by the observed concentration levels of the occupied zone indicate recirculating air flow from the upper flow region down into the lower region. Data obtained from a fluid model have shown that the interface between the upper and lower

Table 2. Estimated parameters characterizing the flow fields of air contaminants

| | | Ventilation principle | | | |
|---|---|---|---|---|---|
| | | Mixing | | Displacement* | |
| Pos. | $H$† (m) | $\mu_p$ (min) | $C_p(\infty)$ (ppb) | $\mu_p$ (min) | $C_p(\infty)$ (ppb) |
| A | 0.5 | 26 | 9 | 30 | 4 |
| | 1.5 | 25 | 8 | 31 | 5 |
| | 2.0 | — | — | 24 | 9 |
| | 2.5 | 24 | 8 | 15 | 12 |
| | 3.5 | 26 | 7 | 14 | 16 |
| | 4.5 | 30 | 7 | 11 | 18 |
| B | 0.5 | 11 | 54 | 13 | 3 |
| | 1.5 | 15 | 45 | 20 | 7 |
| | 2.0 | — | — | 21 | 20 |
| | 2.5 | 15 | 32 | 5 | 31 |
| | 3.5 | 20 | 29 | 5 | 47 |
| | 4.5 | 49 | 22 | 6 | 58 |

* The propeller fan located at test rig A was put out of function during the test period.
† Height above the floor.

zone acts as a lock that will hinder the transport of contaminants between the zones. However, differences in temperature between the fluid and the surfaces were of importance for setting up boundary layer flows. It was observed that cool surfaces located in the upper flow region to a degree caused boundary layer flows from the upper flow region to penetrate down into the lower region [19]. Basically the recommended thermal comfort limits were not violated in the zone of occupancy (test rig. No. B).

The local mean age of air is a valuable tool for the evaluation of the performance of different ventilation systems. The technique has been used in experimental rooms [8] and in the field [3, 5] for characterizing flow fields of different ventilation design principles. Throughout the study all conditions except the ventilation process may be considered approximately constant. It is noted that the heat load in the second test period was elevated compared to the first test period. According to this age parameter, and consistent with previous field studies [3, 5], displacement ventilation was more efficient in distributing fresh air from the air supply ducts than mixing ventilation. Changing the ventilation design principle from mixing to displacement reduced the mean age of air in the zone of occupancy by a factor of 2 (test rig. No. B). This improvement of the air renewal may not apply if cooling fans are used (test rig No. A). In terms of air quality the change in ventilation design principle reduced the exposure levels to the simulated contaminant by a factor 1.6–18.

## CONCLUSION

The intervention study showed that in terms of supplying fresh air to the zone of occupancy the performance of displacement ventilation may exceed that of mixing ventilation by a factor of 2. This finding does not apply in situations where cooling fans are blowing air downwards. In terms of air quality the performance of displacement ventilation may exceed that of mixing ventilation by a factor of 1.6–18. Consequently, the displacement design principle has potential for improving air quality in high-ceiling workshops with a heat load.

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