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Efficient Work Environment Ventilation

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Abstract

A breakthrough in ventilation research was made once it was realized that ventilation principles based on mixed flow patterns are not optimal and that further energy savings can be achieved if an alternative technique could be developed. Several researchers, particularly in the Nordic countries, have shown by theoretical studies that replacing mixed ventilation flow by displacement flow increases ventilation efficiency. This also results in decreased air supply volumes and thus decreased energy requirements. In addition, lower air velocities may reduce problems of comfort and noise.
Soon, however, practical experience showed that a displacement ventilation system must be very carefully designed in order to work as theoretically expected. Not all systems were successful initially and difficulties were encountered in implementing the new technology.

This paper discusses the design basics for practical displacement ventilation systems. An example is taken from a plastics industry, where horizontal displacement ventilation is applied to a real work environment situation. Field measurements are made. Results of the field measurements, carried out in the plastics industry, are surprisingly good. For a channel flow ventilation situation, an air change efficiency of around 80 % has been achieved. It is also shown that worker exposure to styrene vapor can be kept within acceptable limits. The results are affected by the room geometry and the nature of the air supply. The energy gains from high air change efficiencies are discussed, particularly with regard to cold climates where there are large heating requirements.

Introduction

Displacement ventilation systems have become popular in a growing number of applications, because of their efficient contaminant removal and high air change efficiency. Conventional mixing ventilation can never be more efficient than 50%, in terms of air change efficiency. This is one of the main reasons for the development of displacement ventilation systems [1]. Another is the increasing demands for better thermal comfort conditions. In processes where contaminant emission and heat generation are not strongly coupled, momentum driven supply air can flow horizontally through a room [2]. Laboratory work for a better understanding of this type of displaced flow has been reported [3].The distribution of the supply air has been shown here to have a great influence on the flow pattern in a ventilated space.

Successful industrial applications of horizontal displacement ventilation have been recently reported [4,5]. These measurements are discussed further here and compared to numerical simulations of the same industrial work place situation. The process described is the manufacture of polyester products reinforced with glass-fibre. With mixing ventilation, this process requires extremely high air change rates to keep the styrene concentrations in the lamination hall within acceptable limits. This paper reports the results and discusses the research efforts for better ventilation principles in this type of industrial application.

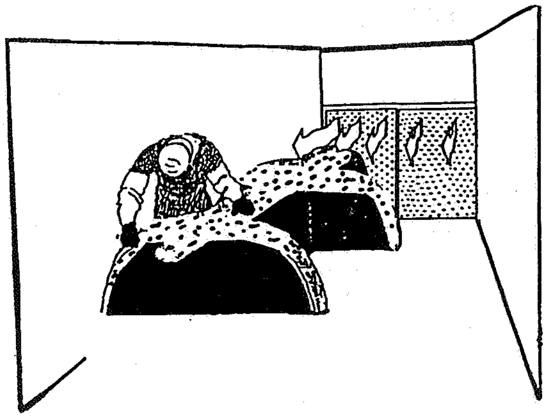


Figure 1. Worker exposure to styrene during lamination with polyester shows great variation with worker position and the principle of ventilation air transport.

Numerical simulations

Numerical simulations were used to analyze the air flow and concentration patterns. The governing equations to describe the three-dimensional transport mechanisms are the equations for continuity, momentum, concentration and the local mean age of air. The k-ε turbulence model, Jones and Launder [6], uses two additional equations, one for the kinetic energy of turbulence k and one for the dissipation of this energy ε. Both these equations are of the same type as those mentioned earlier. In general form, the partial differential equations are:

$$\frac{\partial(\rho u_i \phi)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \qquad (1)$$

where S_ϕ is the source term [7]. In these calculations, the dependent variable ϕ takes the forms: u, v, w, c, $\bar{\tau}_p$, k, ε. For the continuity equation, $\phi = 1$. The SIMPLE algorithm [8] and staggered grid arrangements were used to give discrete solutions of the equations. The discreet equations for each node point were derived by hybrid upwind central differencing [9]. A stable iterative solving of the equations was thus achieved. All calculations were performed in three dimensions on a 40*40*36 grid with constant spacing. The solution time on a Risc-6000 work station was around three hours per dependent variable ϕ.

Boundary conditions

All surfaces were assumed to be adiabatic (zero heat flux). Zero-gradient boundary conditions were assumed for incoming and outgoing air. The boundary conditions for the k-ε turbulence model used were given for isotropic turbulence. Wall functions of the conventional type were used. The ventilation flow rate was 1,9 m^3/s. In the numerical simulations, the styrene emission rate was set to 60 mg/s, and the location of the source was fixed. The laminar and turbulent Prandtl numbers (Schmidt numbers for concentration) σ and σ_ϕ were set to 0,72 and 0,90 respectively in all calculations. The diffusion coefficient, Γ_Φ, took the form:

$$\Gamma_\Phi = \frac{\mu_t}{\sigma_\phi} + \frac{\mu}{\sigma} \quad (2)$$

where μ and μ_t are the laminar and the turbulent dynamic viscosities [6]. A steady-state method was used for the numerical prediction of the mean age of air [7]. The final differential equation for the local mean age of air $\bar{\tau}_p$ includes the density ρ as the source term:

$$\frac{\partial(\rho u_i \bar{\tau}_p)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\Gamma_{\bar{\tau}_p} \frac{\partial \bar{\tau}_p}{\partial x_i}\right) + \rho \quad (3)$$

The boundary condition is $\bar{\tau}_p = 0$ at the inlet.

Measurements in the work environment

Some results from the plastics industry [4,5] are compared with the numerical simulations. The most useful result from the measurements was the extremely high air change efficiency found in a channel-shaped lamination room with horizontal displacement ventilation. The results also indicated how much air is required for ventilation if a mixing ventilation principle is used. These figures are compared to corresponding figures when horizontal displacement ventilation is used. A pattern of the channel flow is described by measured vertical velocity profiles.

Simulated air flows

The lamination chamber, including the air terminal for the one-way orientation of the ventilation flow, has been described [4]. All the dimensions are straightforward and easy to model in a computer program. The mold , however, is more complicated and its shape had to be simplified for the computer program. Figure 2 shows a plan view (x-y) of the simulated 6-metre long, simplified, double-cross mold model. The whole chamber width was open for supply and exhaust air, see Figures 1 and 2.

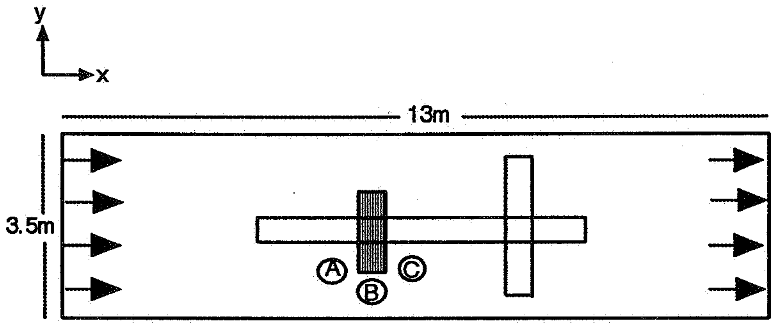


Figure 2. Plan view of the lamination hall, where both industrial measurements and numerical calculations were performed. Points A, B and C are worker positions. The styrene concentration was estimated at these points in the breathing zone 1,5 m above the floor. The simulated styrene emission (60 mg/s) was equally distributed over the striped source area.

The air supply velocity u_0 for both measurements and simulations was 0,3 m/s. The Reynolds number for the flow, Re, is based on the active (open) air supply area A and takes the form:

$$Re = \frac{u_0\sqrt{A}}{\nu} \tag{4}$$

where ν is the kinematic viscosity of the flow. For the displacement flow shown here, a Reynolds number of 55200 was used.

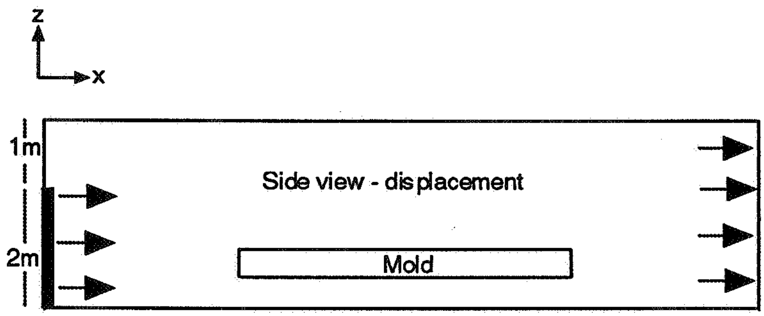


Figure 3. Elevation of lamination hall and mold. In the calculations, the mold geometry was simplified. In reality, the upper corners were a bit curved [4].

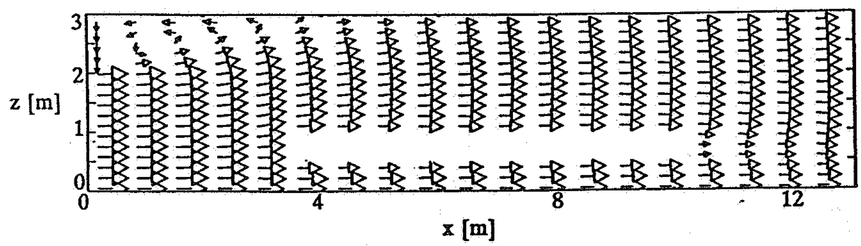
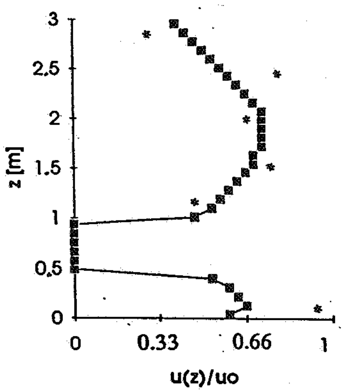


Figure 4. Simulated air flow structure in the lamination hall with horizontal displacement ventilation. Recirculation occurred only in a small area above the air supply. Center-plane crossing the mold; y =1,75 m.



* = measurements [4]

Figure 5. Numerically calculated vertical velocity profile over the mold compared to the measured profile (x = 7 m, y = 1,75 m). Supply velocity u_0 = 0,3 m/s. The velocity is blocked by the mold from z = 0,5 m to z = 1,0 m.

Simulated mixing ventilation as reference

Mixing ventilation was used as a reference and compared to the displacement flow. The comparison was done both for the work place and in the numerical calculations. The mixing model (plant) used was identical to the displacement model, except for the size and shape of the air supply inlet and exhaust outlet. Both were reduced here to a 0,35-m-high slit extending the full width of the room. A constant ventilation flow rate of 1,9 m^3/s was used. This gave supply and exhaust velocities almost six times greater than those in the displacement flow.

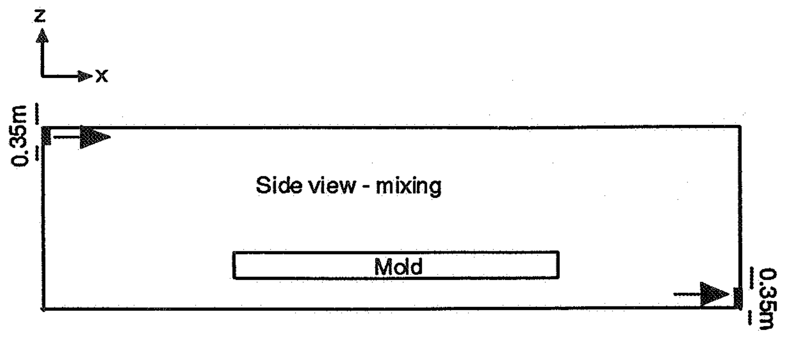


Figure 6. The mixing ventilation reference simulations were also performed in the 13-m-long chamber. The air supply and exhaust velocities were higher than for the displaced flow.

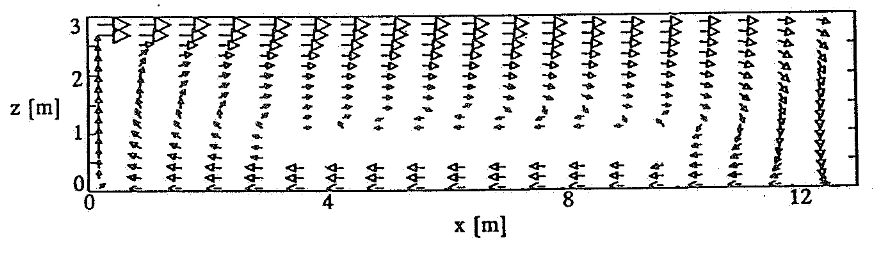


Figure 7. Simulated air flow structure in the lamination hall with mixing ventilation. Recirculation occurs both on the upper and lower side of the mold. Center-plane crossing the mold; y =1,75 m.

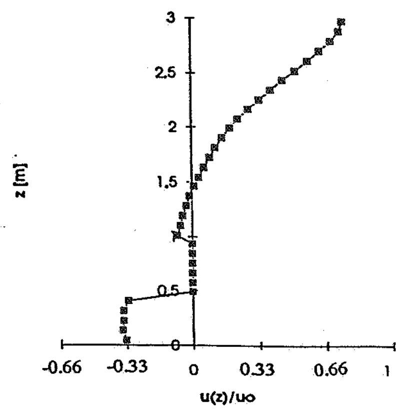


Figure 8. Simulated vertical velocity profile through the center line of the styrene-emitting area of the mold (striped area in Figure 2). A Reynolds number of 132000 was used.

Ventilation efficiency

A tracer gas pulse method was used to measure the mean age of air in the lamination hall [5]. In the simulations, equation (3) was used to calculate the local mean age of air at every grid point in the room. The room mean age, $<\bar{\tau}>$, was then calculated as the average value of the local mean ages. For the air change efficiency, ε_a, the following definition was used:

$$\epsilon_a = \frac{\tau_n}{2<\bar{\tau}>} 100\% \qquad (5)$$

where $\tau_n = V/q$ (=room volume/ventilation flow rate) is the nominal time constant of the room [10].

The contaminant removal effectiveness $<\varepsilon>$ is defined as the ratio between the steady-state contaminant concentration in the exhaust outlet and the steady-state average contaminant concentration in the room [11,12]:

$$<\epsilon> = \frac{c_e(\infty)}{<c(\infty)>} \left(= \frac{\tau_n}{\tau_e}\right) \qquad (6)$$

Equation (6) shows that $<\varepsilon>$ can also be expressed as the ratio between the nominal time constant of the room, τ_n, and the mean residence time of the contaminants, τ_e [13]. It is a measure of how quickly the contaminants are removed from the room. A comparison of measured and simulated ventilation efficiencies is given in Table 1.

Table 1. Measured and simulated ventilation efficiency measures.

	Ventilation efficiency			
	Air change efficiency, ε_a %		Contaminant removal effectiveness, $<\varepsilon>$	
	Displacement	Mixing	Displacement	Mixing
Simulated	85	41	1,36	0,54
Measured	81	51*	1,90	-

* The measured industrial plant for mixing ventilation [14] was different from the simulated mixing reference plant. An air change efficiency of over 50 % indicates some degree of displacement flow.

Table 1 shows that the agreement between simulated and measured air change efficiencies in the displacement chamber (Figures 1-3) is good. The measured contaminant removal effectiveness value is higher than the simulated. A probable reason for the deviation is the location of the contaminant source, which was 0,75 m closer to the exhaust outlet during measurement than in the simulations.

For mixing ventilation, the measured industrial plant was not identical to the simulated plant. The ventilation systems, including air terminals, were also different. The simulated system, shown in Figure 6, was inefficient. The supply air passed through the room without contacting the contaminants. The recirculated air came in close contact with the contaminants and thus blew contaminants the wrong way. This lead to a contaminant removal effectiveness of only 0,54.

By far most efficient ventilation was achieved with the channel-flow arrangement with horizontal displacement ventilation. A buoyancy influence from the exothermal lamination process, not considered in the simulations, could be one reason for the small difference in measured (81 %) and simulated (85 %) air change efficiencies.

Simulated exposure rates with different ventilation principles

Measured exposure rates are not directly dealt with in this report. The reason is that the appropriate measured data were not available. It was difficult to generate steady-state conditions in the lamination hall, and difficult and expensive to measure at many points simultaneously. A constant, time-independent styrene-emission from the mold is difficult to arrange. The comparison of exposure rates here is based on numerical simulations only. Constant boundary conditions are easy to arrange. So it is possible to have a fair comparison of the effects of different air flow principles on the exposure rates.

Table 2. Numerically calculated exposure rates with different ventilation principles at different worker place locations A, B and C. All three locations are 0,3 m from the mold and 1,5 m above floor (see Figure 2). The upper surface of the mold i.e. the styrene source, is 1 m above floor level.

Worker position	Simulated exposure rates [mg/m^3]	
	Displacement ventilation	Mixing ventilation
A	< 0,1	212
B	0,2	150
C	18	143

Displacement ventilation gave lower exposure rates than mixing ventilation, even at worker position C. This is because of a low vertical concentration diffusion, up to the breathing zone, with this type of ventilation flow. At the source level, 1m above floor level, dangerous concentrations occur. Table 2 shows that the two types of ventilation can be expected to give very different exposure rates.

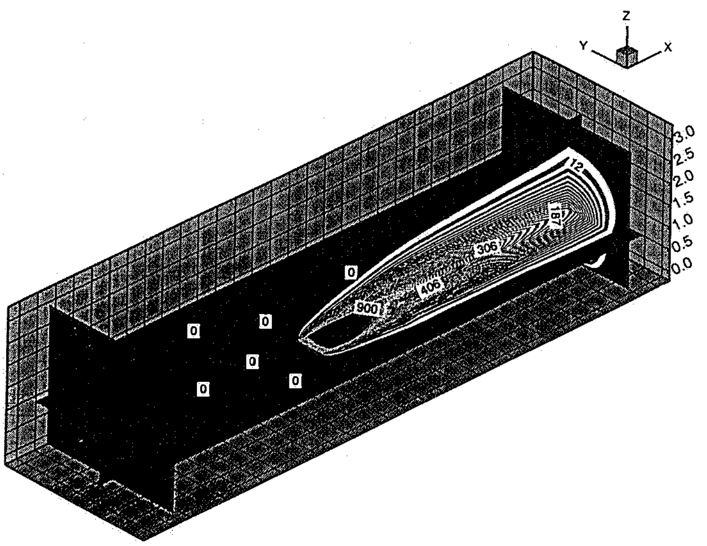


Figure 9. Styrene exposure rates [mg/m^3] with horizontal displacement ventilation. In the zero-concentration zones the exposure is guaranteed to be less than 6 mg/m^3. Fence plot showing center-planes for x-y, x-z and exhaust outlet plane for y-z.

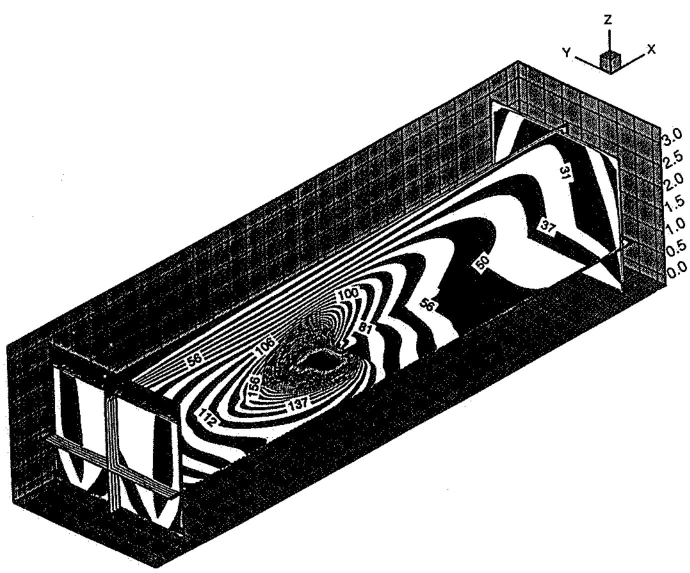


Figure 10. Styrene exposure rates [mg/m^3] with mixing ventilation. The value (value label) positions are the same as for the previous figure. This makes it possible to compare exposure rates with the two ventilation principles.

Energy and health aspects

For mixing ventilation, there is a well-known relation between the contaminant generation $\dot{m}_c$ and the ventilation flow rate q, which gives the contaminant concentration c in the room.

$$c = \frac{\dot{m}_c}{q} \tag{7}$$

There is a great influence of the ventilation efficiency ε and the spatial positions (x,y,z) on the styrene exposure rates. This leads to a new principal equation for the local styrene concentration c_p in the room:

$$c_p = \frac{\dot{m}_c \cdot f(\epsilon, p(x,y,z))}{q} \tag{8}$$

where the ventilation efficiency ε is a measure of the stored styrene quantity in the room, and p(x,y,z) indicates the spread of this quantity. The latter is a position factor that depends on the contaminant source position and the positions of the ventilation air supply and exhaust. Equation (8) states that in all parts of the room the concentrations are directly influenced by the total ventilation flow rate q. The term $\dot{m}_c \cdot f(\varepsilon, p(x,y,z))$ determines the level and the distribution of the local room concentrations.

For worker position C in Table 2, the mixing ventilation flow rate has to be increased by a factor of 8 to bring the exposure rate down to a level achieved by horizontal displacement ventilation (18 mg/m^3). Air consumption data from measurements [4] and concentration extrapolation show that 4-5 times more air has to be used with mixing ventilation. The differences in air change efficiency between displacement and mixing were: for the simulations 85% - 41% = 44%; and for measurements 81% - 51% = 30%, Table 1. The energy E, required to heat ventilation air is given by:

$$E = q \cdot \rho \cdot \Delta t \cdot c_p \cdot \Delta T \tag{9}$$

where
q = ventilation flow rate
ρ = density of air
Δt = heating duration
c_p = specific thermal capacity
ΔT = temperature rise of the heated air.

It has been shown that efficient ventilation means low ventilation flow rates, q. Equation (9) shows that this also means low energy requirements, E. In the equation no account is taken of the heat losses during heating and transport, the efficiency of the heater, or the energy used by the fans. Countries with high heating requirements for ventilation air normally also have a long heating season, which affects Δt, and a large indoor-outdoor temperature difference, which affects ΔT. To compensate for this, ventilation flow rates are often lowered. In this situation, efficient ventilation should be introduced for health reasons. Also, knowledge about correct work practice and favorable worker positions are important. Recently the exposure limits for styrene have been lowered; the eight-hour exposure limit in Sweden and Finland is today 20 ppm (90 mg/m^3). This once again puts higher demands on future ventilation systems.

Conclusions

Because of large variations in the size of manufactured products and the movable nature of work activities, general ventilation is frequently used for controlling airborne styrene. High exposure rates can be avoided by using horizontal displacement ventilation that is properly positioned in the room. Factors that need more study, because of their influence on the ventilation air flow, include room geometries, air supply and exhaust arrangements, and thermal loads. Concentrated air supply diffusers [15] can probably reduce costs further.

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