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EFFICIENCY OF VENTILATION SYSTEMS FOR AN INDUSTRIAL WATER TREATMENT HALL

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SUMMARY

This study brings useful information on the efficiency of ventilation principles usable for an industrial water treatment hall, wherefrom contaminated gases generated by an aerobic process have to be extracted. Three different ventilation systems are compared on the point of view of two efficiency criteria : the global ventilation effectiveness and the local air quality index in the occupied zone. Results are given both for laminar and turbulent flow conditions. It is shown that the nature of the flow has little influence on the global efficiencies and that an ideal ventilation system does not exist since the performances of each system can widely vary depending on the chosen criterion.

INTRODUCTION

This paper deals with pollutant removal from large industrial halls for water treatment where contaminated water contained in large horizontal tanks is treated through an aerobic process. Chemical reactions generate

various species which can be either toxic or simply nasty. The diffusion and transport of these chemical products throughout the hall must be controlled for two main reasons :

i) firstly, contaminants have to be extracted out of the hall for adequate treatment. If the ventilation system is not designed with enough care, large quantities of polluted air will have to be processed, which can lead to an oversizing of the treatment device. This first aspect concerns the point of view of the global ventilation effectiveness [1,2].

ii) secondly, it is necessary to maintain a good air quality in some occupation zones inside the hall. This is the point of view of the local air quality index as defined in [2].

The designer of the ventilation system has to take these two aspects into consideration and then to define the most suitable system.

The present study compares several ventilation systems with regard to these aspects. We shall see further that the solutions to these two problems can be contradictory and that a compromise is necessary.

FLOW DESCRIPTION

The industrial water treatment hall under consideration is defined in fig.1. It is characterized by a large horizontal tank containing contaminated water. An airflow is imposed through the whole depth of the water layer in order to allow aerobic reactions. Due to the large section of the tank, the air flowrate generates very low velocities normally to the free surface. We consider that the pollutant production rate is low and that, consequently, its production does not significantly affect the air density : the pollutant is a passive tracer. Moreover, it is assumed that there is neither heat production nor thermal gradient inside the enclosure : the flow is isothermal and buoyancy forces are neglected.

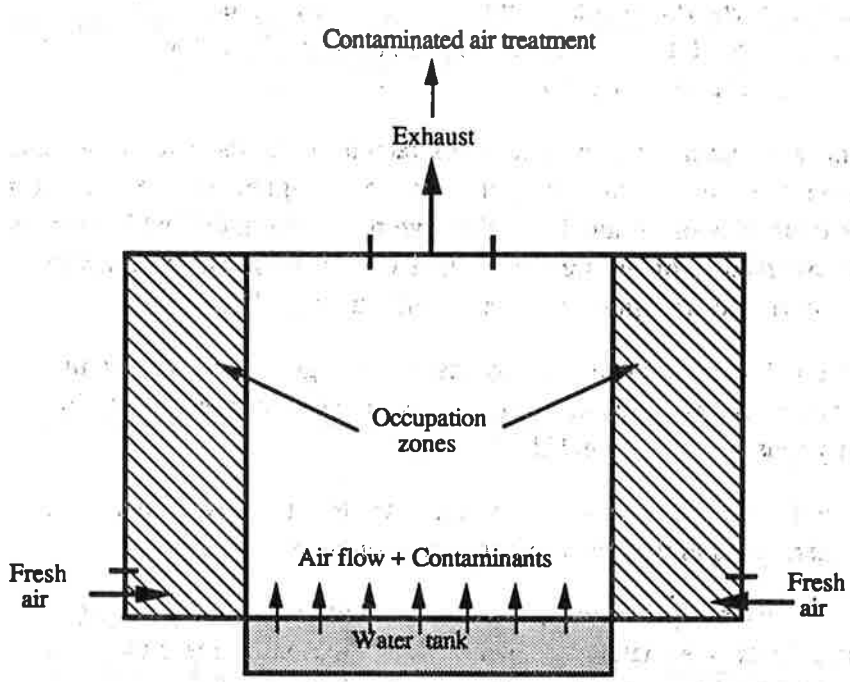


Fig.1 Diagram of the water treatment hall

Fresh air inlet(s) and exhaust(s) can be placed at several locations at the walls depending on the ventilation system under consideration. The different ventilation systems which were studied are presented in fig.2. In practice, rather small venting flowrates are needed due to the low contaminant flux density, which corresponds to inlet Reynolds number comprised between 10^3 and 10^4.

As said before, the ventilation efficiency has to be defined from the point of view of two criteria :

- the ventilation effectiveness which is the ratio of the steady state concentration of contaminant at the exhaust duct, C_e and the steady state mean concentration of contaminant inside the hall, C_m.

$$\varepsilon_V = \frac{C_e}{C_m}$$

- the local air quality index which is the ratio of the steady state

concentration of contaminant at the exhaust duct and the steady state mean concentration of contaminant inside the occupation zone, C_p.

$$\varepsilon_q = \frac{C_e}{C_p}$$

In this context, we tested three configurations derived from widely used systems (in fig.2, we represent the half volume because of symmetry).

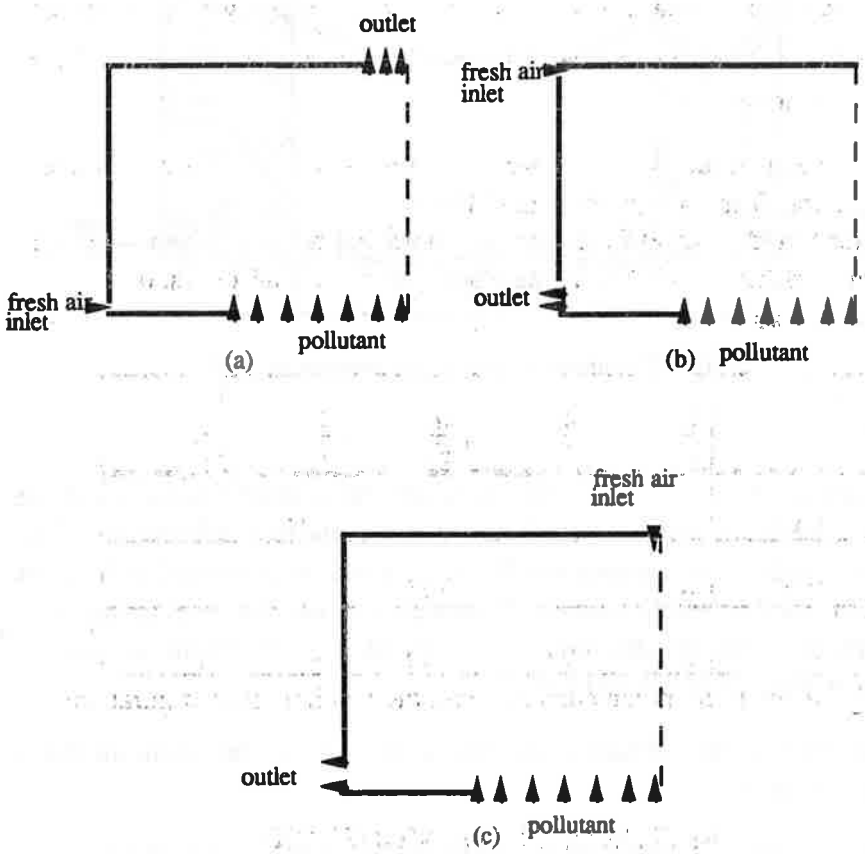


Fig. 2 Studied configurations

PHYSICAL PARAMETRES

This physical problem is completly defined by the following parameters defining the geometry and the flow :

The fresh air inlet jet is defined by its Reynolds number, $Re = \frac{U_0 d_i}{\nu}$. The contaminated air is characterized by the ratio $V_p = \frac{U_p}{U_0}$ where U_0 stands for the fresh air discharge velocity and U_p stands for the polluted air discharge velocity.

The geometrical quantities depend on the ventilation system under consideration. They are given in table1.
Hereafter L and H stand for the cavity width and height, d_i stands for the air inlet dimension, d_e for the exhaust dimension and d_t stands for the width of the water tank.

Config.	$\frac{d_i}{H}$	$\frac{d_i}{L}$	$\frac{d_t}{L}$	$\frac{d_e}{H}$	$\frac{d_e}{L}$	$\frac{L}{H}$
(a)	0.05	---	0.6	---	0.15	1
(b)	0.05	---	0.6	0.15	---	1
(c)	---	0.05	0.6	0.15	---	1

Table 1. Geometrical parameters defining the ventilation configurations

MATHEMATICAL MODELING

Equations

The steady state two-dimensional flow, is characterized by the following set of differential equations describing the conservation of mass, momentum and chemical species. Turbulence is taken into account by the

use of a low Reynolds number k-ε model.

Conservation laws describing the flow are the following :

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0$$

$$\frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho vu)}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}(\mu_e \frac{\partial u}{\partial x}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial u}{\partial y}) + \frac{\partial}{\partial x}(\mu_e \frac{\partial u}{\partial x}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial v}{\partial x})$$

$$\frac{\partial(\rho uv)}{\partial x} + \frac{\partial(\rho vv)}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}(\mu_e \frac{\partial v}{\partial x}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial v}{\partial y}) + \frac{\partial}{\partial x}(\mu_e \frac{\partial u}{\partial y}) + \frac{\partial}{\partial y}(\mu_e \frac{\partial v}{\partial y})$$

$$\frac{\partial(\rho uc)}{\partial x} + \frac{\partial(\rho vc)}{\partial y} = \frac{\partial}{\partial x}(\Gamma_e \frac{\partial c}{\partial x}) + \frac{\partial}{\partial y}(\Gamma_e \frac{\partial c}{\partial y})$$

μ_e and Γ_e are obtained from a two-equation turbulence model characterized by two more equations :

$$\frac{\partial(\rho uk)}{\partial x} + \frac{\partial(\rho vk)}{\partial y} = \frac{\partial}{\partial x}(\Gamma_k \frac{\partial k}{\partial x}) + \frac{\partial}{\partial y}(\Gamma_k \frac{\partial k}{\partial y}) + P_k - \rho\varepsilon - \rho D$$

$$\frac{\partial(\rho u\varepsilon)}{\partial x} + \frac{\partial(\rho v\varepsilon)}{\partial y} = \frac{\partial}{\partial x}(\Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x}) + \frac{\partial}{\partial y}(\Gamma_\varepsilon \frac{\partial \varepsilon}{\partial y}) + C_1 f_1 \frac{\varepsilon}{k} P_k - C_2 f_2 \rho \frac{\varepsilon^2}{k} + \rho E$$

with : $P_k = \mu_t \{ 2[(\frac{\partial u}{\partial x})^2 + (\frac{\partial v}{\partial y})^2] + (\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x})^2 \}$

$$\mu_t = \rho f_\mu C_\mu \frac{k^2}{\varepsilon}$$

$$\mu_e = \mu + \mu_t \qquad \Gamma = \frac{\mu}{\sigma} \qquad \sigma = \frac{\mu c_p}{\lambda} \qquad \Gamma_t = \frac{\mu_t}{\sigma_t} \qquad \Gamma_e = \Gamma + \Gamma_t$$

$C_\mu = 0.09$; $C_1 = 1.44$; $C_2 = 1.92$; $C_3 = 1$; $\sigma_t = 0.9$; $\sigma_k = 1.$; $\sigma_\varepsilon = 1.3$

Present calculations were carried out using the Jones & Launder turbulence model defined by the following functions [3].

f_μ	f_1	f_2	D	E
$\exp\left(\frac{-2.5}{1+\frac{R_t}{50}}\right)$	1	$1 - 0.3\exp(-R_t^2)$	$2\nu[(\frac{\partial\sqrt{k}}{\partial x})^2+(\frac{\partial\sqrt{k}}{\partial y})^2]$	$2\nu\nu_t[(\frac{\partial^2 v}{\partial x^2})^2+(\frac{\partial^2 u}{\partial y^2})^2]$

Table 2- Turbulence model functions

$R_t = \frac{k^2}{\nu\varepsilon}$; $R_n = \frac{nk^{1/2}}{\nu}$ where n is the normal distance from the wall

The problem is completely defined with the boundary conditions summarized in table 2 :

Boundary	u	v	c	k	ε
inlet	U_{in}	0	c_{in}	k_{in}	ε_{in}
walls	0	0	$\frac{\partial c}{\partial n} = 0$	0	0
outlet	$\frac{\partial u}{\partial n} = 0$	$\frac{\partial v}{\partial n} = 0$	$\frac{\partial c}{\partial n} = 0$	$\frac{\partial k}{\partial n} = 0$	$\frac{\partial \varepsilon}{\partial n} = 0$

Table 2 - Boundary conditions

It is worth saying that the choice of this turbulence model results from a study which was carried out for validation of several turbulence models. Four models — the Jones & Launder, the Launder & Sharma, the Lam &Bremhorst low Reynolds models as well as the standard k-ε model — were tested by comparison with experimental data obtained on a scale model [4] . Results show that in the isothermal case (and only for this case), the four models gave almost identical results in good agreement with measurements. For the present study, it has been made choice of the Jones & Launder model.

NUMERICAL METHOD

The previous set of equations was discretized with a finite volume method and the resulted one was solved using a hybrid numerical scheme. The pressure was iteratively treated with the SIMPLE algorithm.

In order to assume an accurate computation of the jet zones we used non uniform mesh grid. Several grids were tested and finally the 46 x 44 grid appeared to be a good compromise between computational time and results precision.

Laminar flow was calculated until variation in local friction coefficient at the walls was less than 0.5%. Should the occasion arise, turbulent computation was then continued using the chosen turbulence model.

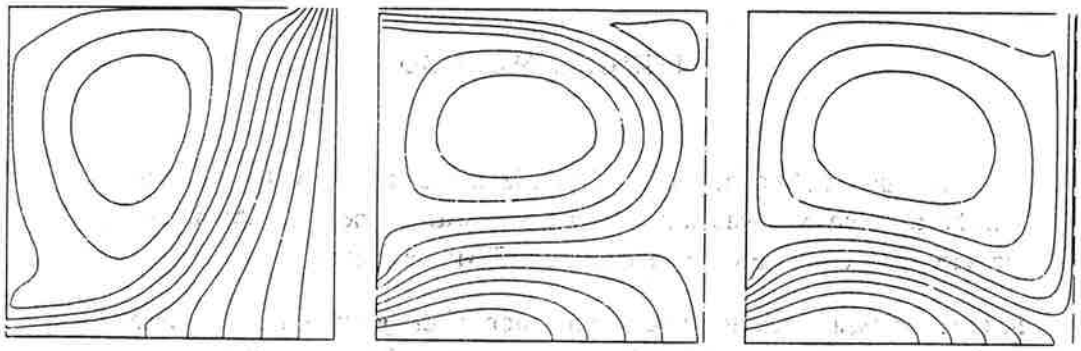
COMPARISON OF THE VENTILATION SYSTEMS

Flow and concentration fields

The different systems were compared from the point of view of the flow and concentration fields, the global ventilation effectiveness and the local air quality index concerning the occupation zones defined defined by $0<x/L<0.4$ as shown in fig.1. For each configuration, computations were carried out both for laminar (Re=100) and turbulent flow (Re=1000).

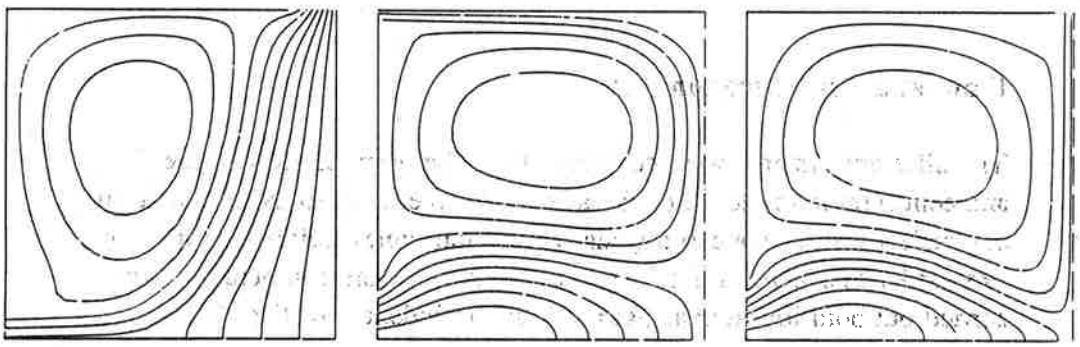
Fig.4a & 4b give the flow field respectively obtained for laminar and turbulent conditions. For both calculations Vp was taken equal to 0.2 and the Schmidt number was equal to 1.13. Results reveal no important qualitative difference between both flow conditions. Similar conclusions can be drawn from fig.5a & 5b showing the concentration fields.

- for the (a) system, the pollutant is concentrated in the central part of the hall, above the tank.
- for the (b) & (c) system, the pollutant concentration is located in the lower part of the hall but is important in the occupied zone.



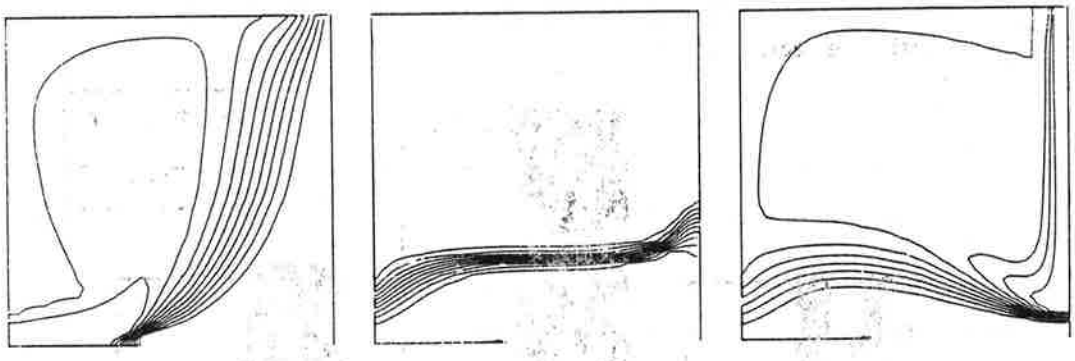
(a) (b) (c)

Fig. 4a Flow field - Laminar flow (Re=100) . Vp=0.2 - Sc= 1.13



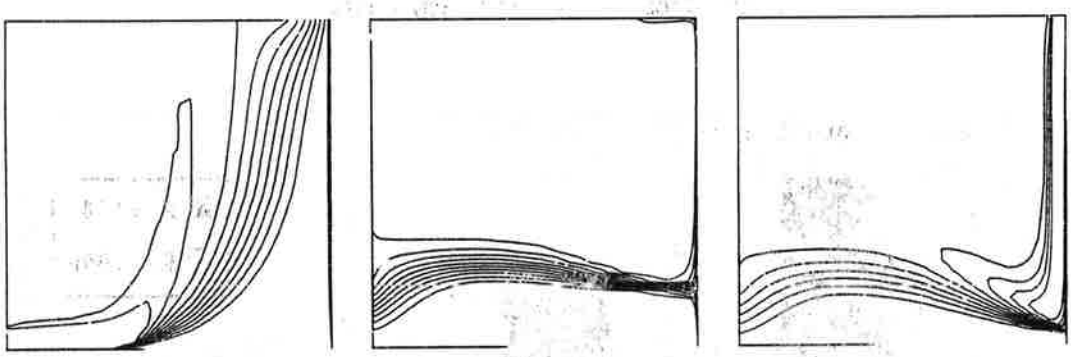
(a) (b) (c)

Fig. 4b Flow field -Turbulent flow (Re=1000) .Vp=0.2 - Sc= 1.13



(a) (b) (c)

Fig. 5a Concentration field - Laminar flow (Re=100) Vp=0.2 - Sc= 1.13



(a) (b) (c)

Fig. 5b Concentration field -Turbulent flow (Re=1000) Vp=0.2 - Sc= 1.13

System efficiencies

These three systems were also compared on the point of view of their efficiencies. Fig.6& 7 respectively give the global ventilation effectiveness and the local air quality index in the occupied zone ($0<x/L<0.4$) for both laminar (Re=100) and turbulent (Re=1000) conditions.

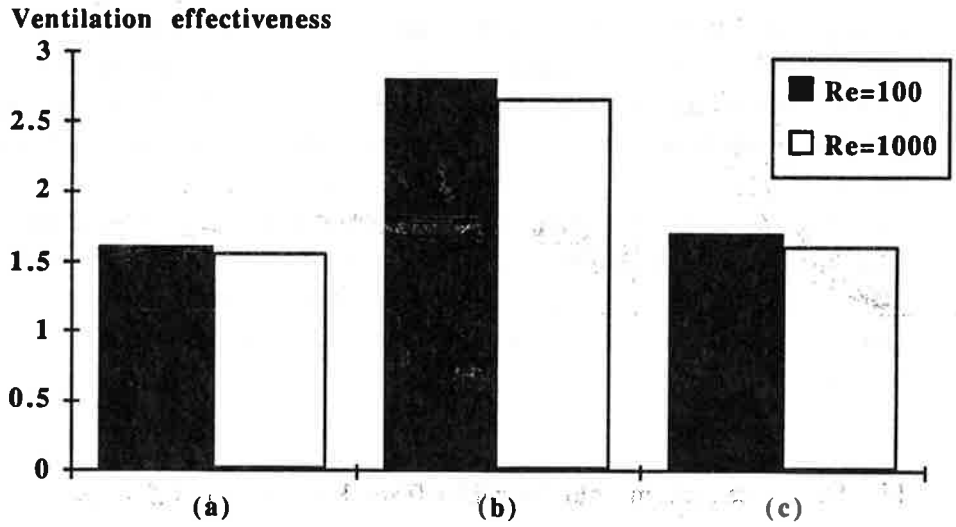


Fig.6 Global ventilation effectiveness

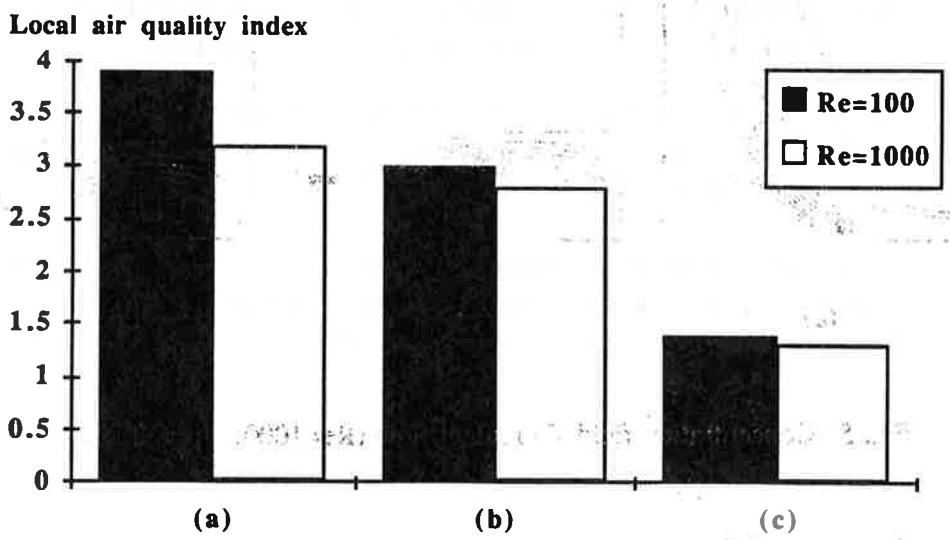


Fig. 7 Local air quality index

Two main conclusions can be drawn from these results :
Firstly, the nature —laminar or turbulent— of the flow has little effect on these efficiencies. This means that, for this type of isothermal flows, it makes sense to use numerical calculations of laminar flows to carry out parameter sensivity studies concerning the global characteristics of real flows.
Secondly, these results demonstrate that, in this case, the ideal ventilation system does not exist, since from the point of view of the ventilation effectiveness, the (b) system is the most appropriate but with regard to air quality in the occupied zone, the (a) system is the best.

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