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19th AIVC Conference

Ventilation Technologies in Urban Areas

Proceedings

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PREFACE

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty one IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D). This is achieved in part through a Programme of collaborative RD&D consisting of forty-two Implementing Agreements, containing a total of over eighty separate energy RD&D projects. This publication forms one element of this Programme.

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the Programme is maintained by an Executive Committee, which not only monitors existing projects but identifies new areas where collaborative effort may be beneficial.

To date the following have been initiated by the Executive Committee (completed projects are identified by *):

Annex 1	Load Energy Determination of Buildings*
Annex 2	Ekistics and Advanced Community Energy Systems*
Annex 3	Energy Conservation in Residential Buildings*
Annex 4	Glasgow Commercial Building Monitoring*
Annex 5	Air Infiltration and Ventilation Centre
Annex 6	Energy Systems and Design of Communities*
Annex 7	Local Government Energy Planning*
Annex 8	Inhabitant Behaviour with Regard to Ventilation*
Annex 9	Minimum Ventilation Rates*
Annex 10	Building HVAC Systems Simulation*
Annex 11	Energy Auditing*
Annex 12	Windows and Fenestration*
Annex 13	Energy Management in Hospitals*
Annex 14	Condensation*
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Annex V Air Infiltration and Ventilation Centre

The Air Infiltration and Ventilation Centre was established by the Executive Committee following unanimous agreement that more needed to be understood about the impact of air change on energy use and indoor air quality. The aim of the Centre is to promote an understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

The Participants in this task are Belgium, Denmark, Finland, France, Germany, Greece, Netherlands, New Zealand, Norway, Sweden, United Kingdom and the United States of America.

19TH AIVC CONFERENCE PROGRAMME
"Ventilation Technologies in Urban Areas"

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

A SEMI EMPIRICAL FLOW MODEL FOR LOW-VELOCITY AIR SUPPLY IN DISPLACEMENT VENTILATION

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A Semi Empirical Flow Model for Low-Velocity Air Supply in Displacement Ventilation

ABSTRACT

Similar to supply air jets in mixing ventilation this paper describes a comprehensive flow model for displacement ventilation derived from the integrated Navier-Stokes differential equations for boundary layers. A new test method for low velocity diffusers in displacement ventilation is developed based on this new flow model. Contrary to jet flow, it is shown that the only independent variable in the new model is the buoyancy flux. In addition to this variable the calculations need a single empirical constant, which is determined from a limited number of full-scale tests of a limited number of similar shaped diffusers of different size. There are made a number of tests to try out the new model.

The results are promising. For plane (two-dimensional) flow the velocity accelerates to a constant value. For radial flow there is also an acceleration zone, after which the velocity decays. Both theoretical and empirical data predicts that for similar shaped diffusers the width of the near zone (distance from the centre of the diffuser to a chosen velocity depends only on the buoyancy flux, not the dimensions of the diffuser (radius and height). One consequence of this is, among others, that the width of near zone cannot, for a certain air flow rate, be shortened by choosing a larger radius and a lower height of the diffuser. The diffuser constant K for radial diffusers has, however, turned out to be more or less dependent on the difference in temperature between the supply air and the room air, probably due to that the outflow is not ideally radial, and the effect of the temperature difference is to make the flow become more radial. The new model also enables the designer to calculate the near zone for arbitrary airflow rate, supply air temperature and arbitrary supply diffuser size of similar shaped diffusers. Practical benefits are, among other things, improved test standards and design methods for displacement ventilation.

LIST OF SYMBOLS

g - Acceleration due to gravity
h - Coordinate of boundary layer border
H - Height of diffuser
I - Profile integral
K - Performance constant
L - The horizontal perimeter of a diffuser
P - Pressure
q_V *- Flow rate*
x - x-co-ordinate
y - y-co-ordinate
T - Absolute temperature
U - Velocity in x-direction
V - Velocity in y-direction
ρ *- Density of air*
β *- Thermal expansion coefficient*
ϕ *- Angle of spread*

Subscripts

0 - location of outlet
1 - location of maximum velocity, classification of profile integral
2, 3 - classification of profile integral
m - maximum
p - plane
r - radial, in the room
f - in the flow
s - supply
U - a fixed arbitrary velocity

1 INTRODUCTION

There are lacking aero- and thermodynamic models for the near zone of the air supply diffusers in displacement ventilation. Because of lacking flow models it is expensive to give valid near zone data for all combinations of supply airflows and supply air temperatures. Manufacturers of the diffusers supply test data, which may be difficult to extrapolate to the actual situation. The aim of this paper is to supply new knowledge into this field and to present a more scientific based, and less expensive, test method and design guide for low velocity air diffusers in displacement ventilation.

2 Theory

2.1 Basics

The type of flow is a kind of boundary layer flow, fig.1, but it does not exhibit the features of self-preservation. Because it is a boundary layer flow the boundary layer momentum equations, neglecting surface friction, apply:

$$\left.\begin{aligned} &U\frac{\partial U}{\partial x}+V\frac{\partial U}{\partial y}=-\frac{1}{x^r}\frac{1}{\rho}\frac{d}{dx}\left(x^r P\right) \text{ (Mom.eqn.)}\\ &\frac{1}{x^r}\frac{\partial\left(Ux^r\right)}{\partial x}+\frac{\partial V}{\partial y}=0 \text{ (Cont.eqn.)}\end{aligned}\right\} \quad (1)$$

Plane flow: r=0

Radial flow: r=1

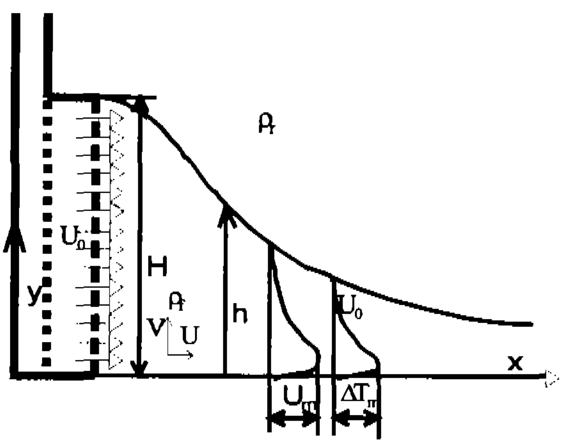

Fig.1 Flow pattern behind a low-velocity diffuser. Supply air temperature lower than room air temperature.

There is no external pressure gradient in the flow region so that the pressure gradient is totally governed by the internal (thermal buoyancy) forces. When the outflow has a lower temperature than the surrounding air, the equation for the static pressure difference between the location (x, y) and a point at the same level in the room outside the boundary region becomes:

$$\begin{aligned}\Delta P_{xy} &= g\int_y^{h(x)}\left(\rho_f(y)-\rho_r\right)dh(x)\\ &= g\rho_r h(x)\int_{y/h(x)}^{1}\frac{\left(\rho_f(y)-\rho_r\right)}{\rho_r}d\frac{y}{h(x)}\end{aligned} \quad (2)$$

Further relations are:

$\rho_f(y)-\rho_r=\rho_r\beta\Delta T_f(y)$

$\Delta T_f=T_r-T_f$ *=Temperature difference between the boundary flow and the room air*

$\beta=\frac{1}{T_r}$ *=Volumetric thermal expansion coeff.*

In the following we substitute h(x) with the parameter h. The final equation for the pressure difference in the boundary layer at a location (x, y) then becomes:

$$\begin{aligned}\Delta P_{xy} &= g\rho_r h\int_{y/h}^{1}\left(\beta\Delta T_f(y)\right)d\frac{y}{h}\\ &= g\rho_r\beta\Delta T_m h\int_{y/h}^{1}\left(\frac{\Delta T_f(y)}{\Delta T_m}\right)d\frac{y}{h}\end{aligned} \quad (3)$$

For the bottom streamline the pressure equation becomes:

$$\begin{aligned}\Delta P_{xy_{(y=0)}} &= g\rho_r\beta\Delta T_m h\int_0^1\left(\frac{\Delta T_f(y)}{\Delta T_m}\right)d\frac{y}{h}\\ &= g\rho_r\beta\Delta T_m h I_1(x)\end{aligned} \quad (4)$$

$I_1(x)$ is the integral if the dimensionless temperature profile, in principle a function of x.

Just for information the energy equation can be used to express ΔT_m in terms of q_v. Differentiating equation 4 with respect to x gives the pressure gradient along the bottom streamline:

$$\frac{d\Delta P_{xy}}{dx}_{(y=0)}=g\rho_r\beta\frac{d}{dx}\left(\Delta T_m h I_1(x)\right) \quad (5)$$

In most situations q_v will not vary much, but it is necessary to incorporate the initial induction when calculating ΔT_m. The energy equation tells us however that $(q_v\Delta T_m)$ is constant and equal to $(q_{v0}\ \Delta T_0)$ implicitly suggesting that the buoyancy flux, $g\beta\Delta T_m q_v$, does not vary with x.

Before solving the differential equation for the bottom stream layer it is convenient to change the continuity equation in equation 1 somewhat. We assume insignificant induction in the acceleration section and, integrating the continuity equation across the boundary layer, we can express the velocity at the bottom plane through the total flow rate, q_v, of the boundary layer. We get:

$$\frac{1}{x^r}\int_0^h \frac{\partial (Ux^r)}{\partial x} dh + \int_0^h \frac{\partial V}{\partial y} dh = \frac{1}{x^r}\frac{\partial}{\partial x}\left(U_m hx^r\right)\int_0^1 \frac{U}{U_m} d\frac{y}{h} = 0$$

For a constant flow rate, qv, V=0 both at the bottom plane and at the outer boundary. Then:

$$\frac{\partial}{\partial x}\left(U_m hx^r\right)\int_0^1 \frac{U}{U_m} d\frac{y}{h} = \frac{\partial}{\partial x}\left(I_2(x)U_m hx^r\right) = 0$$

$$\left(I_2(x)U_m hx^r\right) = const. = \frac{q_v}{\phi^r} \qquad (6)$$

$I_2(x)$ is the integral of the dimensionless velocity profile and is in principle a function of x

U_m is then given by:

$$U_m = \frac{q_v}{I_2(x)\phi^r hx^r} \qquad (7)$$

The momentum equation in equation 1 can now be transformed to be valid for a stream layer at the bottom plane:

$$U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y} = -\frac{1}{x^r}\frac{1}{\rho}\frac{d}{dx}\left(x^r \Delta P_{x0}\right)$$

$$(8)$$

$$x^r U\frac{\partial U}{\partial x} = -\frac{d}{dx}\left(x^r g\beta I_1(x)\Delta T_m h\right)$$

The justification for the expressions above is that $V\frac{\partial U}{\partial y} \approx 0$ at the bottom plane.

U is substituted by U_m in equation 7.

$$x^r \frac{q_v}{I_2(x)\phi^r hx^r}\frac{\partial}{\partial x}\left(\frac{q_v}{I_2(x)\phi^r hx^r}\right) = -\frac{d}{dx}\left(x^r g\beta I_1(x)\Delta T_m h\right)$$

Which further expanded yields:

$$\frac{1}{2}\frac{q_v^2 x^r}{\left(\phi^r\right)^2\left(I_2(x)hx^r\right)^3}\frac{d}{dx}\left(I_2(x)hx^r\right) = g\beta\frac{d}{dx}\left(I_1(x)\Delta T_m hx^r\right)$$

(9)

ΔT_m is a function of x because the profiles change with x. However through the energy equation we can link it to the inflow conditions:

$$\Delta T_m = \frac{q_{v0}}{q_v}\frac{I_2(x)}{I_3(x)}\Delta T_0$$

$I_3(x)$ is the integral of the product of the dimensionless velocity- and temperature profile and is in principle a function of x.

Substituting this expression in equation 9 we get:

$$\frac{1}{2}\frac{q_v^2 x^r}{\left(\phi^r\right)^2\left(I_2(x)hx^r\right)^3}\frac{d}{dx}\left(I_2(x)hx^r\right)$$
$$= g\beta\frac{d}{dx}\left(\frac{q_{v0}}{q_v}\frac{I_1(x)}{I_3(x)}I_2(x)\Delta T_0 hx^r\right)$$

Here q_v is the flow rate after the primary induction when for instance a perforated plate is used. Later the induction is small so we can put $\frac{q_{v0}}{q_v}$ constant. Another legal approximation is that $\frac{I_1(x)}{I_3(x)}$ may be considered constant.

Altogether an acceptable simplification will be that: $\frac{q_{v0}}{q_v}\frac{I_1(x)}{I_3(x)}$ *=constant.*

Finally we can rewrite equation 9 in the following way after gathering the derivatives:

$$\frac{d}{dx}\left(I_2(x)hx^r\right)\left(\frac{1}{2}\frac{q_v^2 x^r}{\left(\phi^r\right)^2\left(I_2(x)hx^r\right)^3} - \frac{(I_1)(x)}{(I_3)(x)}\frac{q_{v0}}{q_v}g\beta\Delta T_0\right) = 0$$

(10)

Eq. 10 has three possible solutions. Either the derivative is zero or the expression within the parenthesis is zero or both are zero. The first alternative means that $\left(I_2(x)hx^r\right)$ is constant. But this has no meaning in the acceleration section because this implies that the velocity is constant. Letting the value of the parenthesis expression be zero we obtain the lowest value of $\left(I_2(x)hx^r\right)$ or the value of $\left(I_2(x)hx^r\right)$ which gives the maximum value of the velocity. The derivative and the parenthesis may as a third alternative be zero simultaneously. Anyway we get:

$$I_2(x)hx^r = \left(\frac{1}{2}\frac{q_v^2 x^r}{\frac{I_1(x)q_{v0}}{I_3(x)q_v}\left(\phi^r\right)^2 g\beta\Delta T_0}\right)^{\frac{1}{3}}$$

Which substituted into eq. 7 gives: *(11)*

$$U_m = \left(2\frac{I_1(x)}{I_3(x)}\frac{g\beta\Delta T_0 q_{v0}}{\phi^r x^r}\right)^{\frac{1}{3}}$$

Equation 11 gives the maximum value of the maximum velocity at the bottom of the boundary layer. For a radial flow pattern this is a function of the distance x. The procedure outlined here does not allow us to establish an expression for calculating the distance x where the maximum value occur.

The result shows that the maximum velocity is uniquely determined by the buoyancy flux of the inflowing air $(q_{V0}g\beta\Delta T_0)$. The relevance of the buoyancy flux is among others discussed by Sandberg (1991).

Index zero refers to the outlet of the diffuser for which: $\Delta T_o = T_r - T_0$

T_0 = *Supply air temperature*

2.2 Plane flow

Plane flow is two-dimensional without any radial spread, fig 2, and is seldom realised in practice. In eq. 11 we put r=0 and we get the following expression for $I_2(x)h$ and U_m (from eq. 6 and 11). The buoyancy flux is given per unit length of the diffuser.

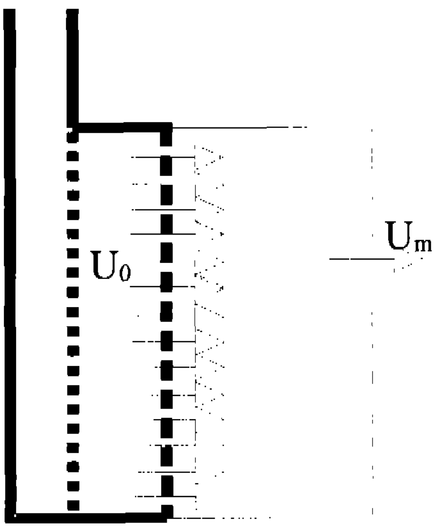

Fig.2 Plane, two-dimensional flow.

$$I_2(x)h = \left(\frac{1}{2}\frac{q_v^2}{\frac{I_1(x)q_{v0}}{I_3(x)q_v} g\beta\Delta T_0}\right)^{\frac{1}{3}} \qquad (12)$$

$$U_m = \left(2\frac{I_1}{I_3}g\beta\Delta T_0 q_{v0}\right)^{\frac{1}{3}} = K_p\left(g\beta\Delta T_0 q_{v0}\right)^{\frac{1}{3}} \qquad (13)$$

The order of magnitude for the constant K_p in equation 13 is found by putting actual numerical values into the expression:

$$K_p \approx \left(2\frac{I_1}{I_3}\right)^{1/3} = \left(2\frac{0,6}{0,368}\right)^{1/3} \approx 1,5$$

(We have used relevant values calculated for ordinary jet flows, Abramovich (1963).)

The analysis shows that the maximum velocity is uniquely determined by the buoyancy flux of the supply air. The result also shows that the velocity is constant in the far region, quite similar to what is found for buoyant plane plumes. The extension of the acceleration region has no analytical solution and consequently needs to be determined experimentally or numerically.

2.3 Radial flow

In practice most low velocity diffusers develop/exhibit a radial flow pattern, fig 3, 4 and 5. In eq.11 we put r=1 and we get the following expression for $I_2(x)h$ and U_m (from eq. 6 and 11). The buoyancy flux is given per unit length of the diffuser.

$$I_2(x)hx = \left(\frac{1}{2}\frac{q_v^2 x}{\frac{I_1(x)q_{v0}}{I_3(x)q_v}(\phi)^2 g\beta\Delta T_0}\right)^{\frac{1}{3}} \qquad (14)$$

$$U_m = \left(2\frac{I_1}{I_3}\frac{g\beta\Delta T_0 q_{v0}}{\phi x}\right)^{\frac{1}{3}} \qquad (15)$$

By multiplying the numerator and denominator in expression 15 with x_O, eq. 15 is transformed to:

$$U_m = \left(2\frac{I_1}{I_3}\frac{q_{V0}g\beta\Delta T_0}{\phi x_0}\right)^{1/3}\left(\frac{1}{\frac{x}{x_0}}\right)^{1/3} \qquad (16)$$

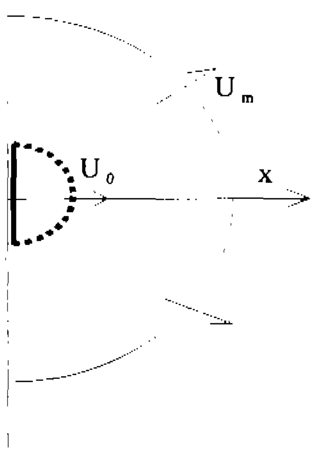

Fig. 3 Radial flow pattern.

Here $\frac{q_{V0} g \beta \Delta T_0}{\phi x_0}$ is the buoyancy flux per unit horizontal length of the diffuser front, because ϕx_0 is the circumference of the diffuser, which we may choose to denote L. When the radial flow has expanded to the side walls or symmetry boundary for neighbouring diffusers, x / x_0 acquires its largest value and the flow become two-dimensional (constant velocity for larger distances).

The solution gives the relation between buoyancy flux, distance from the diffuser and the maximum velocity (acceleration velocity) for radial flow. The solution does not give any information about the length of the acceleration region and no information about the rate at which the velocity decrease after the initial acceleration.

It is a problem in radial flow, contrary to two-dimensional flow where the x-value does not appear, that the x-value for maximum velocity cannot analytically be determined. However, tests have shown that the value for x / x_0 at maximum velocity does not vary very much for a set of similar shaped diffusers. The value seems to be within a span of 2,5 - 3,5. This implies that the maximum velocity can be determined by an expression of the type:

$$U_m = K_r \left(\frac{q_{V0} g \beta \Delta T_0}{L} \right)^{1/3} \qquad (17)$$

The order of magnitude for the constant K_r is found by putting relevant numerical values in eq.16:

$$K_r \approx \left(2 \frac{I_1}{I_3 \frac{x_m}{x_0}} \right)^{1/3} = \left(2 \frac{0,6}{0,368 \cdot 3} \right)^{1/3} \approx 1 .$$

We have here in the same way as for two-dimensional flow used values for the profile integrals from jet flow.

The buoyancy flux, $q_V g \beta \Delta T_m$, is however constant and can be set equal to: $q_{Vo} g \beta \Delta T_o$, where subscript o refers to the supply outlet where: $\Delta T_o = T_o - T_s$

T_s = *supply air temperature*

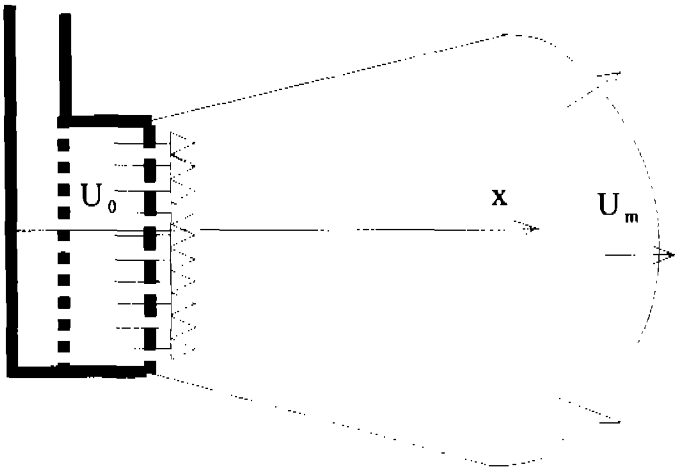

Fig 4. Horizontal flow pattern behind a plane diffuser.

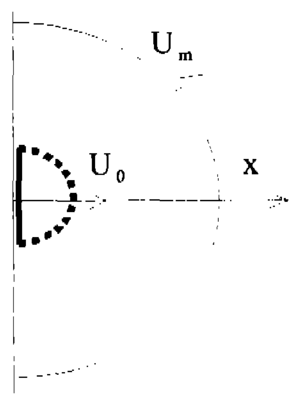

Fig.5. Horizontal flow pattern behind a radial diffuser.

3 VERIFICATION

Table 1

q_{v0} [m3/h]	q_{v0} [m3/s]	Δt [K]	L [m]	X_0 [m]	X_m [m]	T_s [K]	H_0[m]	Ar_0	Um [m/s]	K_r
200,	0,05556	3,0	0,785	0,25	0,85	292	1,0	20,1	0,300	1,56
311,	0,08639	3,1	0,785	0,25	0,56	292	1,0	8,6	0,440	1,95
420,	0,11667	3,2	0,785	0,25	0,47	292	1,0	4,9	0,640	2,54
310,	0,08611	5,2	0,785	0,25	0,77	290	1,0	14,6	0,455	1,70
309,	0,08583	8,2	0,785	0,25	1,09	286	1,0	23,5	0,510	1,63
309,	0,08583	3,1	0,785	0,25	0,59	292	0,8	4,5	0,450	2,00
310,	0,08611	3,2	0,785	0,25	0,57	292	0,5	1,1	0,515	2,26
179,	0,04972	3,1	0,393	0,125	0,38	293	1,0	6,5	0,620	2,63
292,	0,08111	5,1	0,393	0,125	0,37	291	1,0	4,0	1,020	3,10
399,	0,11083	8,1	0,393	0,125	0,50	287	1,0	3,5	1,430	3,35
309,	0,08583	7,3	0,785	0,25	1,03	282	1,0	21,2	0,480	1,59

Table 1 shows the result of laboratory tests, Wulff (1995), on a series of similar diffusers where K_r is calculated from eq. 17 based on test-data. As we can see the constant is not very much "constant". A comprehensive parameter study was performed. The Π-theorem in dimensionless analyses suggests that the Archimedes number is a significant dimensionless parameter if the length scale matters. Our results did not verify this assumption. The parameter study resulted in the following: The flow pattern is not strictly radial but is influenced by the airflow per unit length and the temperature difference. Increasing flow rates (inertial effect) reduces the radial effect, and an increased temperature difference (buoyancy effect) has an opposite effect. The constant K_r showed to be a function of the parameter $(\Delta T_0/(q_{v0}/L)^2)$ only. In fig. 6 is shown an example from a test series where airflow rates, diffuser height, diffuser radius and temperature difference were varied. The result could be uniquely correlated in the following function:

$$K_r = 25{,}4\left(\frac{\Delta T_0}{q_{v0}^2/L}\right)^{-0{,}44} \qquad (18)$$

Increased airflow rate increases the constant while increased temperature difference has the opposite effect.

For two-dimensional flow we found that the velocity is constant in the far region analogue to two-dimensional convection plumes. If in radial flow there is an analogy to three-dimensional convection plumes the velocity decay in the far region should obey the following relation:

$$U_m \approx (x_0 + x)^{-1/3}$$

Our test data, fig. 7, indicate that the decay exponent is closer to -1. Work carried out by Peter V. Nielsen et al in Denmark (1998, 1991 and 1992) also indicates that the exponent for x is more like -1 in the far region. We therefore adopt the exponent -1 in the far region for radial flow.

Then the following interesting relations can be derived for the far region:

$$\frac{U_x}{U_m} = \frac{x_1}{x} \qquad (19)$$

x_1 is the x-value where the decay curve crosses the ordinate value 1 ($\frac{U_x}{U_m} = 1$)

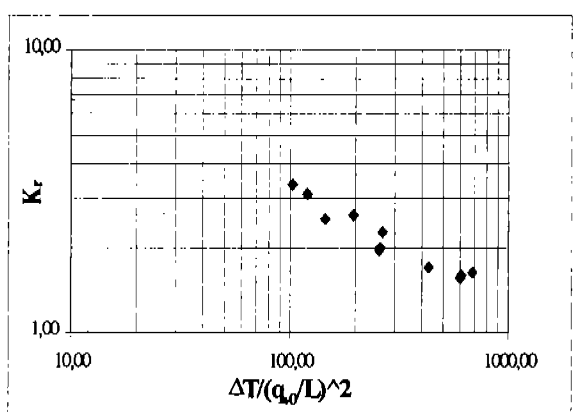

Fig. 6 Lab tests where K_r is corrected by the parameter $\Delta T_0/(q_{v0}/L)^2$

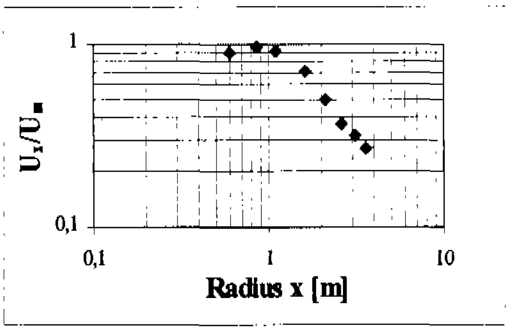

Fig. 7 Velocity as a function of x.

For a certain, arbitrary chosen, velocity the following relation is valid in the far region, based on eq. 16 and 19:

$$U_{x,U=konst} = \left(2\frac{I_1}{I_3}\frac{q_{V0}g\beta\Delta T_0}{L}\right)^{1/3}\left(\frac{1}{\frac{x_m}{x_0}}\right)^{1/3}\frac{x_1}{x_U}$$

$$= \left(\frac{x_1}{x_U}\right)^{\frac{2}{3}}\left(\frac{x_1}{x_m}\right)^{\frac{1}{3}}\left(2\frac{I_1}{I_3}\frac{q_{V0}g\beta\Delta T_0}{L}\right)^{1/3}\left(\frac{1}{\frac{x_U}{x_0}}\right)^{1/3} \qquad (20)$$

When the chosen constant velocity is f.ex. 0,2 m/s we can put $x_U = x_{0,2}$.

For a series of similar shaped diffusers, tests have indicated that the following relation holds:

$$\left(\frac{x_1}{x_U}\right)^{\frac{2}{3}}\left(\frac{x_1}{x_m}\right)^{\frac{1}{3}} \approx Constant$$

Then we can postulate that test data should obey the following relation in the far region for a set of similar shaped diffusers:

$$U_{U=konst} = K_{rU}\left(\frac{q_{V0}g\beta\Delta T_0}{L}\right)^{1/3}\left(\frac{1}{\frac{x_U}{x_0}}\right)^{1/3} \qquad (21)$$

For a product series of similar shaped diffusers it can be expected that test data should obey the following relation in the region with decreasing velocity between the distance from the diffuser and an arbitrary chosen constant velocity:

$$\frac{U_x}{\left(\frac{q_{V0}g\beta\Delta T_0}{L}\right)^{1/3}} = K_{rU}\left(\frac{1}{\frac{x_U}{x_0}}\right)^{1/3} \qquad (22)$$

The constant K_{rU} must be experimentally (empirically) determined.

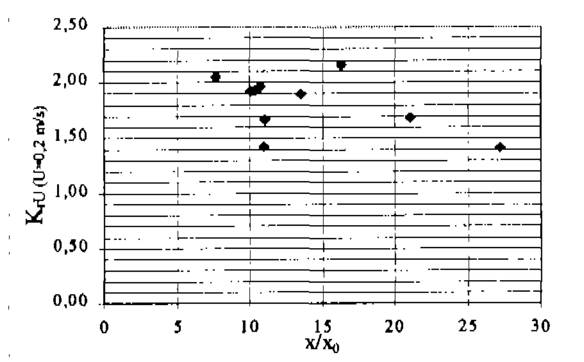

Fig. 8 $K_{r0,2}$ determined from tests using eq.22

In fig 8 the constant K_r is calculated from the same tests as shown in table 1 based on eq. 22. As we can see the constant is not very much "constant". A comprehensive parameter study showed that in this far region only the temperature difference matters. The explanation is the same as for the acceleration region that increasing thermal forces (buoyancy) increases the radial effect i.e. forces the flow more to the sides. The inertia forces have no effect in this region. Fig. 9 shows the temperature effect and the good correlation between the constant K_{rU} and the temperature difference. Again the height of the diffuser and, indirectly, the Archimedes number does not matter.

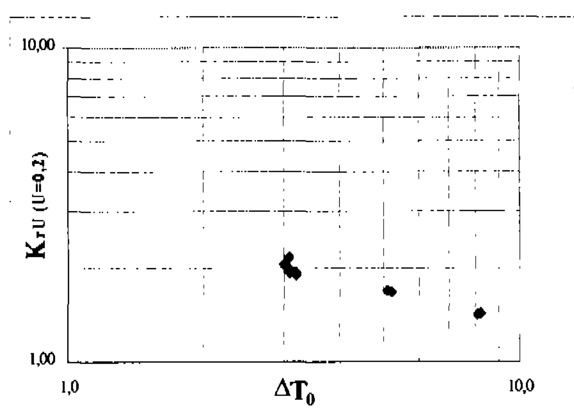

Fig 9. K_{rU} varies uniquely with ΔT_0.

The temperature function for this test series became:

$$K_{r0,2} = 3\Delta T_0^{-0,36} \qquad (23)$$

This temperature function (temperature and flow rate function for the maximum velocity) is different for different diffuser design. Having ideal diffusers with ideal radial flow, the exponents in these functions are expected to become zero. Then the constants in equations 7 and 22 are really constant. Low K-factor is equivalent to good lateral spread.

The relations shown indicate that for a set of similar shaped diffusers it is not necessary to test more than a few sizes and loads to determine K_{rU}. Further it is sufficient to document a few velocities like 0,15, 0,2 and 0,25 m/s. The formulas make it possible to calculate the distance from a diffuser, $(x_U - x_0)$, where a given velocity occur, using arbitrary combinations of temperature differences and air flow rates.

4 Performance documentation of air diffusion devices

Performance documentation should contain the following information as a function of size and flow rate:

- Maximum velocity for the near zone of the diffuser as a function of distance and the temperature difference between the supply air and the air temperature 1,1 m above floor level, i.e. K_r or K_p i.e. to determine K_r or K_p as a function of ΔT_0. A few terminal velocities like 0,15, 0,2 and 0,25 m/s should be tested.
- Isovel envelopes to show the spread of the air flow.
- The relative temperature increase in the near zone.
- Pressure drop and noise generation data.

5 Summary

A semiempirical flow model is developed for displacement ventilation.

- The formulas make it possible to calculate the distance from a diffuser, $(x_U - x_0)$, where a given velocity occur, using arbitrary combinations of temperature differences and airflow rates.
- Utilizing the model the manufacturer has a cheaper and more useful and versatile method to perform performance documentation for low velocity diffusers compared to existing methods.
- The semiempirical model developed makes it possible for a designer to choose the right diffuser for a certain application if proper performance testing is made by the manufacturer.
- Existing testing standards should be revised according to the new method.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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DEVELOPMENT OF INTELLIGENT ALGORITHMS FOR INDOOR AIR QUALITY CONTROL THROUGH NATURAL VENTILATION STRATEGIES

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ABSTRACT

Simulations have been performed to investigate the performance of intelligent algorithms for control of indoor air quality through natural ventilation strategies whilst simultaneously meeting the requirements of thermal and visual comfort. The proposed control algorithms are founded on the knowledge base of the building physics and support the control of natural ventilation through control of the window opening, whilst simultaneously controlling the lighting, heating and cooling systems of the building. The concentration level of CO_2 is taken as the indicator of indoor air quality, whilst predicted mean vote, interior illuminance levels and the daylight glare index have been adopted as the high level controlled parameters for thermal and visual comfort respectively. The impact of the controller on the overall indoor environment has been investigated.

INTRODUCTION

Energy conscious design and energy management in buildings is of paramount importance, both as a means of contributing to security and diversity of energy supplies and also as a means of combating the environmental impact of excess energy consumption on both a global and local scale. Energy consumption in buildings cannot be considered without also accounting for the well being and comfort of the occupants, and experience has shown that building occupants are often not satisfied with the strictly controlled conditions in well sealed buildings. Factors such as indoor air quality and the sick building syndrome, lead to a requirement for utilising the available environmental energy sources to a maximum whilst optimising energy consumption. The use of environmental energy sources is one of the major elements of energy conscious design in buildings, and with it come parallel implications for indoor air quality, thermal comfort and visual comfort.

Intelligent control techniques using fuzzy logic controllers offer the possibility of meeting the required indoor comfort conditions by selectively controlling the building plant and the opening of apertures. The challenge is to meet thermal comfort, visual comfort and indoor air quality requirements simultaneously, whilst minimising energy consumption. The control conflicts arising with respect to the requirements, together with those arising from the necessity to avoid rapid switching of the control system, therefore have to be addressed.

The procedure adopted for the development of the intelligent, rule-based fuzzy control algorithms is to use detailed and simplified models of the building environment, validate the simplified models via comparison with the detailed models, and to compare various fuzzy logic control strategies using these models. A detailed dynamic single zone building model including thermal and mass transport, daylighting and artificial lighting components has been implemented in the MATLAB/SIMULINK environment and the Fuzzy Logic Toolbox of MATLAB has been used to implement the fuzzy logic controller.

MATHEMATICAL MODELLING

A large number of tools are available for simulation of the indoor environment in buildings, however in order to facilitate the combined simulation of the controller and building, the building physics models have been implemented using open architecture programming in the SIMULINK environment.

The building model consists of a zone air temperature model, heat transfer through enclosure structures, heat transfer through fenestration, radiative heat exchange in the interior of the zone, radiative heat exchange at the exterior of the zone, natural ventilation model, CO_2 concentration model, relative humidity model, lighting model including daylight, artificial lighting and glare, and an outdoor environment model including solar radiation and temperature.

For the purpose of the development of the fuzzy rule base, a simplified empirical model can be used to offer a general correlation for calculation of the air flow rate, or the mean air velocity in a zone. This is important from the point of view of the thermal comfort rule base as well for dissipation of indoor pollutants. For the purpose of developing a control algorithm based on the possible or probable air flow rate, this method is therefore suitable.

The British Standards Method has been adopted in order to obtain a realistic representation of the air flow inside the zone with single sided ventilation, wherin ventilation can be due to the effect of the wind or to the temperature diference, Allard (1998). In the case of the former, the air flux Φ is:

$$\Phi = 0.025\ A\ V\ (m^3/s)$$

where:

A : opening surface (m^2)
V : wind velocity (m/s)

In the latter case the wind flux is:

$$\Phi = C_d\ \frac{A}{3}\ \sqrt{\Delta T g H_2 / T}$$

where:

C_d : decrease flux coeficient (-)
g : acceleration due to gravity (kg/m^2s)
ΔT : difference between indoor and outdoor temperature (K)
T : average of indoor and outdoor temperature (K)
H_2 : opening height (m)

A simplified calculation of C_d can be obtained from the equation:

$$C_d = 0.0835\ (\Delta T / T)^{-0.3}$$

and the heat transfer per unit time due to ventilation is:

$$\begin{aligned} Q &= m\ c_{pair}\ \Delta T = \rho_{air}\ V_{airchange}\ c_{pair}\ \Delta T\ /dt \\ &= \rho\ c_{pair}\ \Delta T\ \ V_{airchange}/dt \\ &= \rho_{air}\ c_{pair}\ \Delta T\ \ \Phi \end{aligned}$$

KNOWLEDGE BASE AND FUZZY INFERENCE SYSTEM

The objective of the research activity is to investigate the applicability of fuzzy rule-based controllers for global indoor environmental control and to determine the most appropriate fuzzification, inference and deffuzification procedures. The rule base for the controller can be constructed with two different approaches. The first approach is to combine sets of rules which address particular aspects of indoor environmental control. In this instance, one rule set concerns control of visual comfort, with another rule set for thermal comfort and a third rule set for indoor air quality. The defuzzification process is then used to address the conflicts which arise between the control actions for the three categories for indoor environmental control. A second option is to construct a set of rules incorporating knowledge of building behaviour which in effect will amount to the number of inputs to the controller multiplied by the number of fuzzy sets describing those inputs. A first set of membership functions and three parallel rule sets for indoor environmental control, based on the first option, are presented.

The concentration level of CO_2 in the indoor environment is commonly adopted as the indoor air quality index, despite the fact that the concentration of total volatile organic compounds and smoke which could be harmful to the occupants health could possibly remain high even at low CO_2 concentration levels. The natural variations and random fluctuations in the outdoor wind conditions rule out the use of classical control techniques such as ON/OFF and PID, where the goal of the controller would be to maintain the CO_2 concentration below a certain crisp set point defined by the user. The change in window opening area has been proposed as the output parameter for a fuzzy logic controller for indoor air quality control under natural ventilation, whilst two parameters are proposed as inputs to the fuzzy controller, these being the CO_2 concentration and its derivative; Bruant et al (1996), Dounis et al (1996). An alternative control parameter is the window opening area.

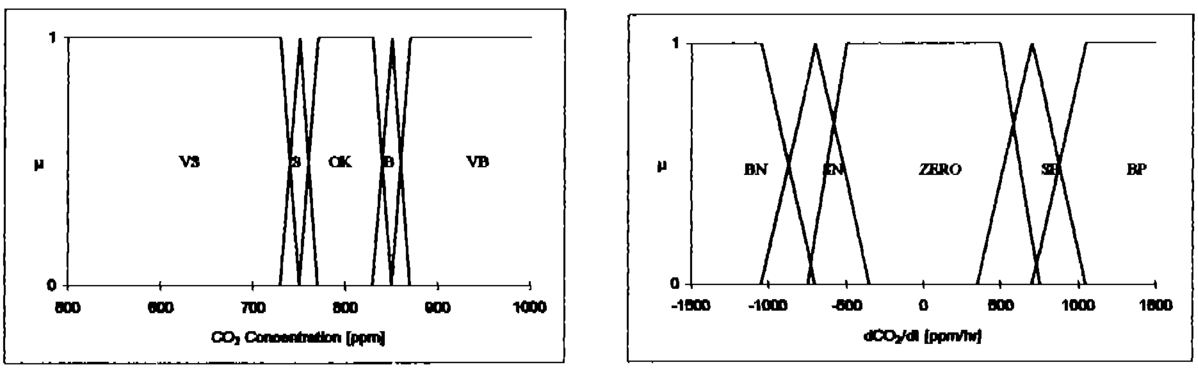

Fig. 1: Fuzzy set membership functions for indoor air quality control (controller input)

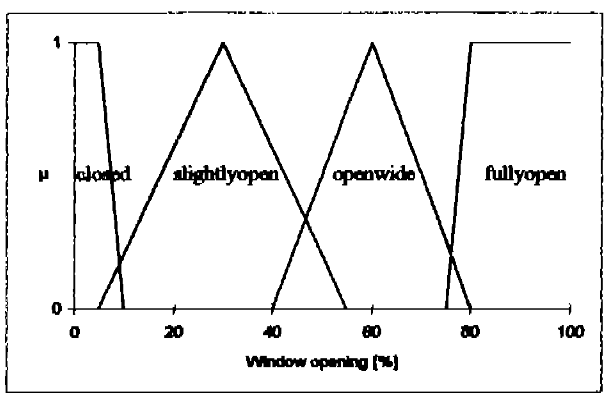

Fig. 2: Fuzzy set membership functions for window opening (controller output)

The rule set for indoor air quality control consists of the following 25 rules:

If (co2 is VS) and (dco2/dt is BN) then (window is closed)
If (co2 is VS) and (dco2/dt is SN) then (window is closed)
If (co2 is VS) and (dco2/dt is ZE) then (window is closed)
If (co2 is VS) and (dco2/dt is SP) then (window is closed)
If (co2 is VS) and (dco2/dt is BP) then (window is closed)
If (co2 is S) and (dco2/dt is BN) then (window is closed)
If (co2 is S) and (dco2/dt is SN) then (window is closed)
If (co2 is S) and (dco2/dt is ZE) then (window is closed)
If (co2 is S) and (dco2/dt is SP) then (window is slightlyopen)
If (co2 is S) and (dco2/dt is BP) then (window is slightlyopen)
If (co2 is OK) and (dco2/dt is BN) then (window is closed)
If (co2 is OK) and (dco2/dt is SN) then (window is closed)
If (co2 is OK) and (dco2/dt is ZE) then (window is closed)
If (co2 is OK) and (dco2/dt is SP) then (window is slightlyopen)
If (co2 is OK) and (dco2/dt is BP) then (window is slightlyopen)
If (co2 is B) and (dco2/dt is BN) then (window is slightlyopen)
If (co2 is B) and (dco2/dt is SN) then (window is slightlyopen)
If (co2 is B) and (dco2/dt is ZE) then (window is slightlyopen)
If (co2 is B) and (dco2/dt is SP) then (window is openwide)
If (co2 is B) and (dco2/dt is BP) then (window is openwide)
If (co2 is VB) and (dco2/dt is BN) then (window is slightlyopen)
If (co2 is VB) and (dco2/dt is SN) then (window is openwide)
If (co2 is VB) and (dco2/dt is ZE) then (window is openwide)
If (co2 is VB) and (dco2/dt is SP) then (window is fullyopen)
If (co2 is VB) and (dco2/dt is BP) then (window is fullyopen)

In a similar manner, membership functions and rule sets have been constructed for indoor thermal comfort and visual comfort control. Whilst indoor air temperature is one indicator of indoor thermal comfort, the actual response of a building occupant is a result of the temperature, mean radiant temperature, relative humidity and air movement, as well as the clothing and activity of the occupant. A more appropriate indicator of thermal comfort is the predicted mean vote (PMV). Using the PMV indicator (with thirteen fuzzy sets) as the input to a fuzzy logic environmental control system results indirectly in the regulation of the environmental variables, Dounis et al (1995). The controller can have PMV and outdoor temperature as its inputs, its outputs can be auxiliary heating, auxiliary cooling and ventilation window opening. In order to minimise the number of rules which are adopted for a prototype rule base for control of all three environmental categories, the number of fuzzy sets describing the PMV has been reduced to seven, resulting in a set of 35 rules for the control of thermal comfort.

The rule set consisting of thirty five rules for thermal comfort control takes the following form:

If (pmv is VN) and (tamb is L) then (ah is VB)(ac is OFF)(window is closed)
If (pmv is N) and (tamb is L) then (ah is PB)(ac is OFF)(window is closed)

. . . . . .

If (pmv is P) and (tamb is VB) then (ah is OFF)(ac is PM)(window is closed)
If (pmv is VP) and (tamb is VB) then (ah is OFF)(ac is ON)(window is closed)

Similarly, the illuminance levels and the daylight glare index have been proposed as controlled parameters in order to build visual comfort control with fuzzy reasoning, with outputs the window shading and the artificial lighting, Dounis et al (1992b). The fuzzy logic controller aims to maintain the illumination level within the desirable limits, which have been set by the user, whilst at the same time, glare must be controlled to fall within acceptable levels. A rule set consisting of twenty four rules for visual comfort control takes the following form:

If (dill is n) and (dgi is imp) then (shading is unshaded)(al is on)
If (dill is n) and (dgi is acc) then (shading is unshaded)(al is on)
. . . . . .
If (dill is vl) and (dgi is unc) then (shading is fullyshaded)(al is off)
If (dill is vl) and (dgi is unb) then (shading is fullyshaded)(al is off)

SIMULATION RESULTS

A series of simulations for both the heating and cooling season have been performed using a model of single building zone model with a south facing opening. The fuzzy controller attempts to maintain the PMV between –0.5 to 0.5 (the comfort zone) via regulation of auxiliary heating, cooling and window opening, whilst simultaneously controlling the lighting levels to 300 lux and the daylight glare index to 22 glare index units through regulation of the shading and artificial lighting. At the same time, the controller adjusts the window opening for control of CO_2 concentration levels by natural ventilation. The results of the simulations for the heating season are shown for three consecutive days in the Figure 4. The use of shading to control visual comfort, in particular glare, results in a reduction of the solar gains and the controller is unable to maintain thermal comfort, since the rule set for thermal comfort is based on the utilisation of thermal gains for minimisation of energy consumption and the heating system is sized accordingly. The CO_2 levels are adequately maintained close to the desired level of 800ppm.

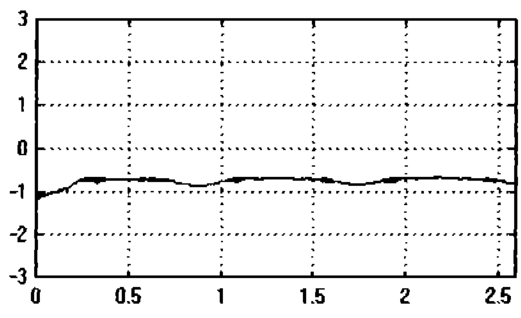

Fig.3a : PMV index vs time (x 10^4s)

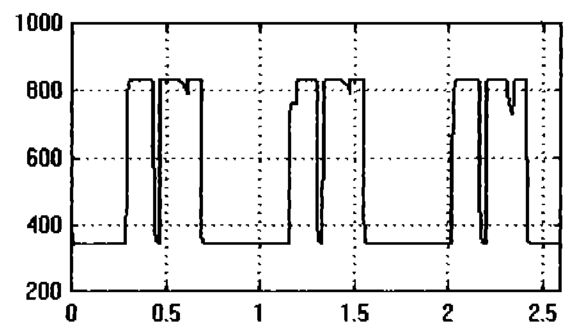

Fig.3b : CO_2 concentration (ppm) vs time (x 10^4s)

CONCLUSIONS

The simulations demonstrate the potential of the fuzzy controller to maintain indoor air quality close to the desired set point (800ppm) whilst simultaneously attempting to maintain thermal and visual comfort. Allowing for adjustment of the rule set to accommodate for the negative effects of the conflicts between the control requirements of the three indoor environmental parameters, it can be seen that a fuzzy rule-based controller is capable of maintaining the three global parameters close to there desired levels. The specific rule base has been developed to allow a maximum use of solar gains, but this is in conflict with the glare controller which imposes very high shading co-efficients. The result is that the PMV remains close to its lower desirable limit. The adoption of the rate of change of the PMV indicator as an input to the fuzzy controller is also proposed, since otherwise the indicator must be outwith the comfort zone before a control action is taken, Egilegor et al (1997). This is not necessarily the case for lighting control, since the natural variations in daylighting are more rapid than those of the thermal response of buildings, and this could lead to instability in the control. Further study of the controller performance with the proposed inputs, together with a combined building and plant model (in order to allow for plant response) is necessary in order to arrive at a final prototype controller for control of visual comfort, thermal comfort and indoor air quality. The case of a full rule base should also be studied.

Furthermore, intelligent control techniques offer the possibility of meeting the required levels of indoor air quality through selective exploitation of the potential for natural ventilation and the use of mechanical ventilation. The rule base should be further developed in order to study the performance of the controller with respect to the selective control of combined natural and mechanical ventilation.

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SIMULATION OF INFILTRATION HEAT RECOVERY

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Simulation of Infiltration Heat Recovery

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Abstract

Infiltration has traditionally been assumed to affect the energy load of a building by an amount equal to the product of the infiltration flow rate and the enthalpy difference between inside and outside. Results from detailed computational fluid dynamics simulations of five wall geometries over a range of infiltration rates show that heat transfer between the infiltrating air and walls can be substantial, reducing the impact of infiltration. The classical method for determining the infiltration energy load was found to over-predict the amount by as much as 95 percent and by at least 10 percent. However, in order to achieve significant heat recovery flow paths which are unlikely in adventitious leakage are required.

Nomenclature

c_p = specific heat capacity of air (1006 J/kg K)
c_{ps} = specific heat capacity of insulation solid component (1006 J/kg K)
c_{pw} = specific heat capacity of wall sheathing (1200 J/kg K)
g = gravity (9.81 m/s^2)
k = air thermal conductivity (0.025 W/m K)
k_{eff} = effective thermal conductivity of insulation (0.025 W/m K)
k_s = thermal conductivity of insulation solid component (0.041 W/m K)
k_w = wall sheathing thermal conductivity (0.13 W/m K)
Q = total (conduction and convection) heat load (W)
Q_{inf} = energy load due to infiltration (W)
Q_o = pure conduction heat load (W)
m = infiltration mass flow rate (kg/s)
p = air pressure (Pa)
t = time (s)
T = temperature (K)
T_i = inside air temperature (298 K)
T_o = outside air temperature (274 K)
T_s = temperature of insulation solid component (K)
T_w = wall sheathing temperature (K)

[1] LBNL-42098:This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Building Technology of the U.S. Department of Energy under contract no. DE-AC03-76SF00098.

u = air flow velocity in x-direction (m/s)
v = air flow velocity in y-direction (m/s)
x = horizontal co-ordinate (m)
y = vertical co-ordinate (m)

α = insulation permeability (10^{-8} m^2)
$\Delta T = T_i - T_o$ (24 K)
ε = infiltration heat exchange effectiveness (dim)
ϕ = mass fraction of air in wall insulation material (0.99)
μ = air viscosity (1.72×10^{-5} kg/m s)
ρ = air density (kg/m^3)
ρ_s = density of insulation solid component (70 kg/m^3)
ρ_w = wall sheathing density (544 kg/m^3)

Introduction

Air leakage through building envelopes, infiltration, is a common phenomenon, which impacts both indoor air quality and building energy consumption. Some researchers have studied the potential of reducing building energy consumption by intentionally incorporating this process into the building design (3,9,15). In this technique, known as dynamic insulation, air is drawn through the building envelope in a direction that opposes the natural conductive flow of energy, so that some portion of the energy ordinarily lost to conduction is recovered.

In the general case, however, infiltration is unintentional and uncontrolled. Claridge and Bhattacharyya (7) note that a great deal of work has been devoted to the prediction and measurement of infiltration in building systems, but little effort has been directed toward determining the actual energy impact of infiltration. Infiltration can contribute a significant amount to the overall heating or cooling load of a building, but the actual size of the effect depends on a host of factors, including environmental conditions, building design, and construction quality. Based on experimental measurements taken at 50 residential buildings, Caffey (5) concluded that up to 40 percent of the heating/cooling costs in the homes studied was due to infiltration. In another study of residential buildings, Persily (13) attributed about one-third of the heating/cooling requirements to infiltration. Sherman and Matson (14) examined measured leakage data and concluded that a high fraction of the space conditioning load in U.S. buildings was due to infiltration. The results of a recent study (12) of U.S. office buildings performed by the National Institute of Standards and Technology (NIST) show that air leakage accounts for about 15 percent of the heating load in office buildings nationwide and about 1 or 2 percent of the cooling load. By all measures, the impact of infiltration can be sizeable and, so, should be considered in calculations of building energy consumption.

$$Q_{\text{inf}} = \dot{m} c_p (T_i - T_o) \qquad (1)$$

The traditional method of accounting for the extra load due to infiltration is to simply add another term to the energy balance. The extra term, shown in equation 1, is the product of the infiltrating air mass flow rate, the specific heat capacity of air, and the temperature difference between inside and outside. Note this term does not include the effects of moisture in the air. This equation is appropriate if the leaking air does not interact themally with the

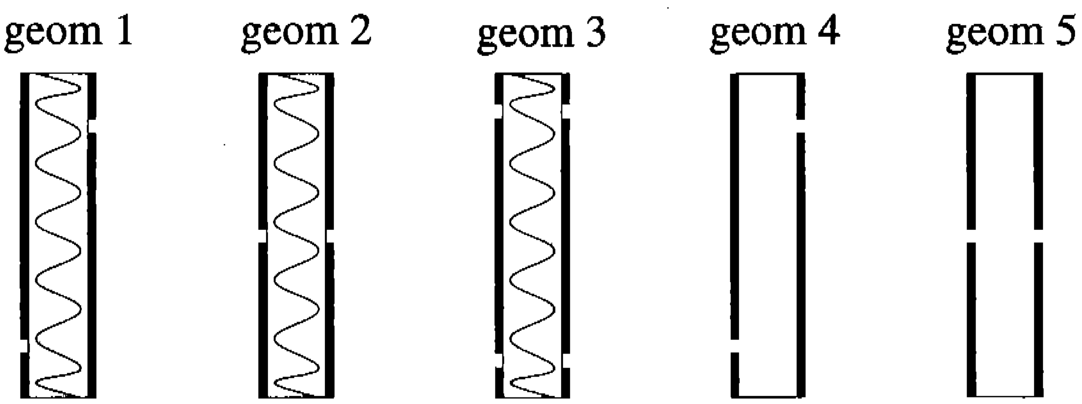

Figure 2: The five wall geometries examined; 1,2, & 3 are insulated and 4 & 5 are empty

Problem Formulation

The room represents a row-house inner unit (figure 1) and is composed of an infiltrating wall, a corresponding exfiltrating wall, and a ceiling, floor, front wall, and rear wall with no air leakage. The building envelope is separated into non-interacting wall elements, which are examined individually. Information from the individual walls is added together to determine the overall impact for a complete room system. The windward and leeward walls, both of the same geometry type, are matched by their air leakage rates and have crack lengths that extend the entire depth of the wall (10 m). The bulk air flow within the room is not represented, but this should not be a problem because, as Etheridge (8) notes, the internal room air flow has only a secondary effect on infiltration. The most important influences are wind-induced pressure differences and buoyancy of room air in the vicinity of the wall. Both are represented in these simulations.

The wall section is modeled as a two-dimensional, time-dependent system. Air flow and energy transport within the air are determined via the Navier-Stokes and energy equations (equations 2-5), respectively. A laminar representation is used for the flow. Solutions show this to be a reasonable assumption, as the Reynolds number at the crack under the highest pressure difference for the empty wall is only about 2000. Velocities elsewhere in the flow are much lower and would not provide the potential for turbulence. The plywood sheathing is represented as an impermeable, solid material. Energy transport within this material is calculated via the conduction equation (equation 6). Insulation in the wall is represented as a porous material. Air flow through the insulation is determined via Darcy's Law (equations 7 & 8), a common model for flow through porous media (4,10). Energy transport through the insulation is determined via a modified form of the energy equation (equation 9). In equation 9, an effective conductivity, given by equation 10, is used in the conduction flux term and the thermal inertia of the solid component is included in the transient term.

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}=0 \qquad (2)$$

$$\frac{\partial \rho u}{\partial t}+\frac{\partial \rho uu}{\partial x}+\frac{\partial \rho uv}{\partial y}=-\frac{\partial p}{\partial x}+\mu\frac{\partial}{\partial x}\left[2\frac{\partial u}{\partial x}-\frac{2}{3}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right)\right]+\mu\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right) \quad (3)$$

$$\frac{\partial \rho v}{\partial t}+\frac{\partial \rho uv}{\partial x}+\frac{\partial \rho vv}{\partial y}=-\frac{\partial p}{\partial y}+\rho g+\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)+\mu\frac{\partial}{\partial y}\left[2\frac{\partial v}{\partial y}-\frac{2}{3}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right)\right] \quad (4)$$

$$c_p\frac{\partial \rho T}{\partial t}+c_p\frac{\partial \rho uT}{\partial x}+c_p\frac{\partial \rho vT}{\partial y}=k\frac{\partial^2 T}{\partial x^2}+k\frac{\partial^2 T}{\partial y^2}+\frac{\partial p}{\partial t}+u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y} \quad (5)$$

$$\rho_w c_{pw}\frac{\partial T_w}{\partial t}=k_w\frac{\partial^2 T_w}{\partial x^2}+k_w\frac{\partial^2 T_w}{\partial y^2} \quad (6)$$

$$\frac{\partial p}{\partial x}=-\frac{\mu}{\alpha}u \quad (7) \quad , \quad \frac{\partial p}{\partial y}=-\frac{\mu}{\alpha}v \quad (8)$$

$$\frac{\partial}{\partial t}\left(\phi\, c_p\rho T+(1-\phi)c_{ps}\rho_s T_s\right)+c_p\frac{\partial \rho uT}{\partial x}+c_p\frac{\partial \rho vT}{\partial y}=k_{eff}\frac{\partial^2 T}{\partial x^2}+k_{eff}\frac{\partial^2 T}{\partial y^2}+\frac{\partial p}{\partial t}+u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y} \quad (9)$$

$$k_{eff}=\phi k+(1-\phi)k_s \quad (10)$$

Thermal gradients in the system develop due to the difference between indoor and outdoor conditions giving rise to natural convection. As mentioned previously, it is important to represent the effects of buoyancy on the flow to properly determine infiltration rates and the heat flux at the wall, so buoyancy is included in these simulations. A simple, temperature-dependent empirical equation of state for the fluid density, coupled with the body force term in the fluid y-momentum equation introduces the effects of buoyancy into the flow.

Results and Discussion

Simulations are performed for the five wall geometries under wind-induced pressures ranging from 0.1-10 Pa with a constant temperature difference of 24K between inside and outside. Due to the complexity of the problem, it was not possible to achieve a converged solution using the steady-state equations. Therefore, the time-dependent equations were integrated in time until steady-state was reached. Comparison of results from simulations using a coarse computational grid (33,000 nodes) and a fine grid (140,000 nodes) for two different wall geometries showed that the coarse grid provided a grid-independent solution. All results presented here are steady-state solutions from simulations using a 33,000 node grid.

The main point of interest is the extra energy load introduced by infiltration. This is determined by first calculating the heat flux through the room walls with no air leakage, designated as Q_o. Then, the energy flux is determined for the same wall types with air leakage. The difference between the two values is the infiltration-induced energy load. The convection and conduction energy fluxes across the external (outside) face of each wall are calculated for infiltrating and exfiltrating configurations. Using the external building face as the system control volume boundary is an arbitrary choice, the interior face could be used as well. However, it is important from an organizational standpoint that the energy accounting be performed at a consistent location.

For a given wall geometry, the infiltration air flow rate and energy flux vary with environmental conditions. The infiltration rate versus wind-induced pressure is shown in figure 3. In all cases, infiltration increases with wind pressure, but the actual values vary between geometries due to different flow resistances. The walls containing insulation (1-3) show a linear relationship between pressure and flow rate at pressures above about 0.5 Pa. This is the expected behavior, because the primary flow resistance in these cases, the insulation, is represented by linear flow-pressure relation, Darcy's Law. The empty walls (4-5) show a power law relationship with a flow exponent of about 0.5 at pressures above about 0.5 Pa. The literature (1,8,11) shows that for typical residential dwellings flow exponents range between one half to three quarters with averages typically about two-thirds. Although exponents up to unity are possible, they rarely play a significant role in adventitious leakage.

Below 0.5 Pa, the influence of the stack-induced pressure is comparable to that of the wind-induced pressure. It is interesting to note that for geometry 3, there is a low pressure plateau induced by the stack effect and leak geometry.

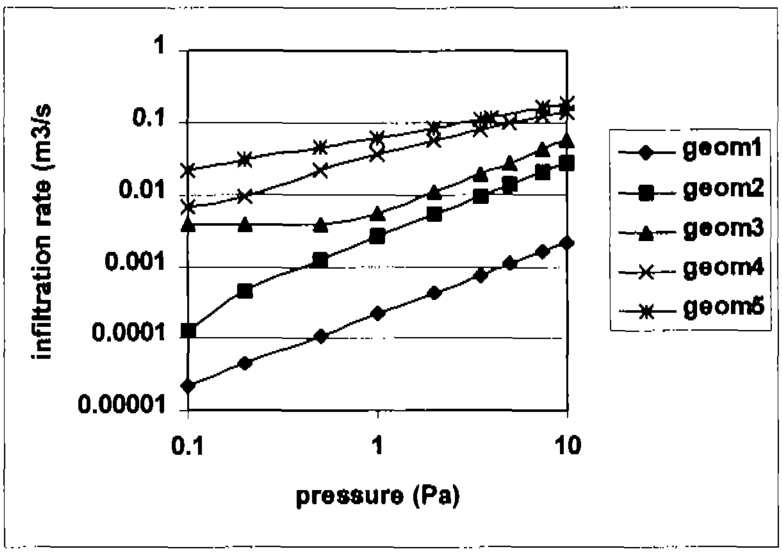

Figure 3: Infiltration rate vs. wind-induced pressure (ΔT=24K, crack length=10 m).

The extra energy load due to infiltration is given as a fraction of the classical load. The extra load, calculated via equation 11, uses the infiltration heat exchange effectiveness, ε, a non-dimensional factor introduced by Claridge (2,6,7), given in equation 12.

$$Q_{\inf} = (1-\varepsilon)\dot{m}c_p(T_i - T_o) \quad (11)$$

$$\varepsilon = 1 - \frac{Q - Q_o}{\dot{m}c_p \Delta T} \quad (12)$$

Figure 4 shows ε for the five wall geometries at various wind-induced pressures. In all cases, the heat recovery decreases with increasing flow rate. This is true for a given wall geometry over the range of pressures or in a comparison of different wall geometries at a given pressure. The heat transfer becomes less efficient at high flow rates because there is less time for energy to be transported from the walls to the infiltrating air and less conducting energy available to recover.

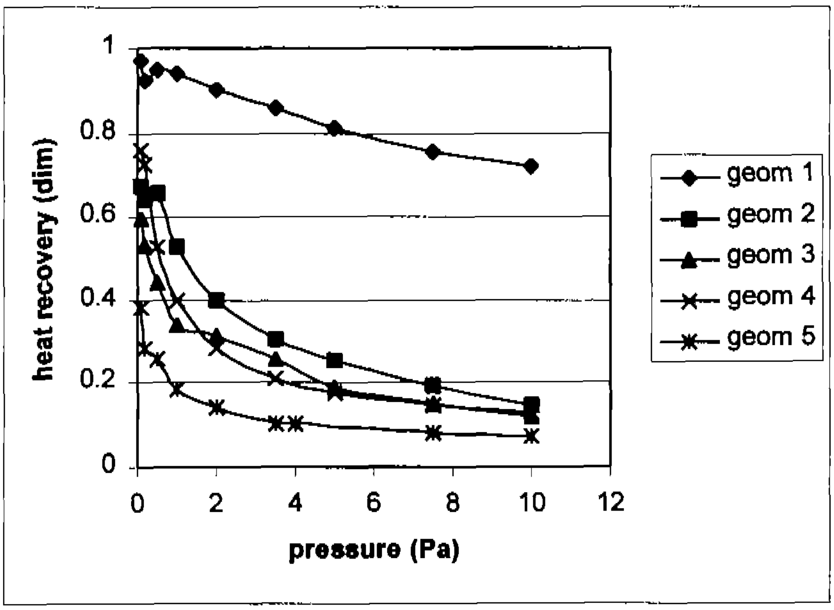

Figure 4: Heat recovery vs. wind-induced pressure for the five walls (ΔT=24 K).

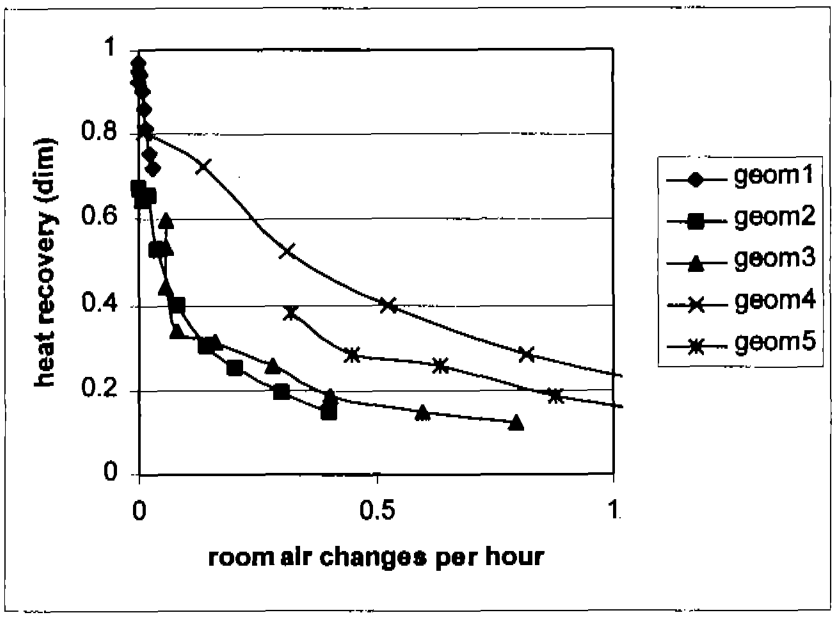

Figure 5: Heat recovery vs. infiltration rate in ACH.

Figure 5 shows that for a given infiltration rate (expressed here in room air changes) and wall type, insulated or empty, the heat recovery increases with infiltration path length. For example, wall geometries 4 and 5 are empty walls with long and short air flow paths, respectively. At a given infiltration rate the configuration with the long air flow path, geometry 4, has a higher heat recovery. Again, the increased heat recovery is due to longer transit times for infiltrating air in the wall. The same trend is true for geometries 1 and 2, but is more difficult to see on the graph. It is interesting to note that the three insulated walls all seem to fall on a common curve, suggesting that a scaling law may apply. Future work on other configurations and geometries will be needed to explore this notion further.

An interesting point is revealed in comparison of geometries 2 and 3. Note that geometry 3 is similar to 2, except there are two holes instead of one. At pressures above about 0.5 Pa, geometry 3 has twice the flow as geometry 2, but nearly the same heat recovery. This indicates that in geometry 3 there is little interaction between the two holes, which is due to the large flow resistance of the insulation separating them. A wall of this design may not need to be modeled in its entirety. However, preliminary studies of this wall with an empty cavity show that there is a significant amount of interaction between the high and low holes, so this may not be a universal trait for all such wall designs.

In one sense, our results compare well to the experimental measurements of Claridge and Bhattacharyya (7). They calculated a maximum heat recovery of about 0.8 for a "diffuse" leakage path, which corresponds most closely to our geometry 1. This was nearly the average value determined in this study, as can be seen in figure 4.

In other ways, our results are not entirely comparable. In our simulations, we subjected each configuration to a range of pressures that are representative of the wind-induced pressures that real dwellings experience. For a given pressure, infiltration rates vary depending on the flow resistance (determined by the wall construction and environmental conditions), as can be seen in figures 3 and 5. In contrast, Claridge and Bhattacharyya adjusted the driving pressure to provide the same range of infiltration rates for each configuration. This technique is useful for some purposes, but the flow rates are too low to be representative of infiltration in most real dwellings, like our row-house scenario. When plotted against air change rate, all of their heat recovery values would be at very low air change rates, like our geometry 1 data. The infiltration rates for the configurations with "concentrated" leakage paths would be much higher (orders of magnitude) for realistic driving pressures.

Conclusions

Though still requiring substantiation, these results show the potential importance of infiltration heat recovery. In some circumstances, particularly in cases with low flow rates and long air flow paths, the heat recovery can be substantial, up to 95 percent. In these cases, the classical method will greatly over-predict the extra heating load due to infiltration. Even when the heat recovery is at the lowest level calculated, about 0.1, the classical method will over-predict the infiltration load by 10 percent. All leakage paths have not been represented in our simulations, but it seems that some modification should be considered to the classical method to increase its accuracy.

In reality, the importance of infiltration heat recovery will be determined by the particulars of the problem. For example, Sherman and Matson (14) found infiltration rates in typical U.S. housing stock to be around 1 ach. Our results suggest that about 10-20 percent of the heat would be recovered at these flow rates, so it is unlikely that this mechanism plays a large role in the rather leaky envelopes of U.S. housing stock. In new construction, where infiltration rates can be quite low, infiltration heat recovery could be a significant effect, provided the infiltrating air goes through the insulating layers and not just directly through holes. However, this leakage scenario is associated with high flow exponents, which are not observed in most housing stock. Consequently, we would not expect this leakage scenario to occur, except in cases where it has been included in the design, as in dynamic insulation, and, so, the infiltration heat recovery would not be large.

The results in this report are limited to just a few test cases, but future work will include other wall geometries, more diverse environmental conditions, and integration of these findings into whole-building energy analysis models.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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A SOFTWARE APPROACH FOR ECONOMIC OPTIMISATION OF ENERGY USE IN BUILDINGS

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A software approach for economic optimisation of energy use in buildings

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SYNOPSIS
When designing a new, or retrofitting an existing building it is desirable to minimise the heating/cooling load, total energy use and emissions from combustion. Solutions to accomplish this has to be held up against investment costs, maintenance costs, longevity and of course indoor climate (among other things). Optimisation between these different and often competing criteria is complex, and involves a lot of parameters. In real life such an optimisation process is often done in a superficial way, or designers often use well-known climatization solutions without evaluating alternatives.

Energy in Buildings 2.0 (EiB) is a very user-friendly Windows application, applying the graphical user interface possibilities in Windows 3.1/Windows95. It simulates temperature, power demand, energy use and emission release from combustion. It is also possible to do profitability analysis of measurements/alternatives. *EiB* can be used in design of new buildings or retrofitting of existing buildings.

Simulations in this paper shows that smart energy design in commercial buildings , often using building dynamics, can reduce the energy consumption dramatically. Measures like : daylight utilisation, use of thermal mass, hybrid ventilation systems and demand controlled heating, ventilation and cooling, are very effective for reducing the energy use in commercial buildings. Profitability analysis indicate that many of these energy efficiency measures are cost effective.

1 Introduction

Optimisation between competing criteria (e.g. indoor climate and energy use) when designing the climatization concept of a building, can be done by profitability analysis, comparing discounted running costs (salary, energy cost, maintenance cost, etc.) with investment costs. This is possible providing you have a model (or models) that can predict heating and cooling load, total energy use and thermal comfort. In order to do such model predictions in reasonable time with necessary accuracy a computer program has to be used, which take into account all the essential factors like : Insulation level of the building fabric, fabric tightness, internal loads, occupant behaviour, solar gain, thermal storage capacity, ventilation system solutions, energy generation plant efficiency, distribution systems, operating times, etc.

Experience shows that such computer programs has to be very user-friendly to be used by designers. Too many complex computer tools have been developed, only to be used by a few researchers for academic purposes.

Energy in Buildings 2.0 (EiB) is a very user-friendly Windows application, applying the graphical user interface possibilities in Windows 3.1/Windows95. It simulates temperature, power demand, energy use and emission release from combustion. It is also possible to do profitability analysis of measurements/alternatives. *EiB* can be used in design of new buildings or retrofitting of existing buildings.

EiB has been widely used in Norway for three years. From user feedback a more versatile, accurate and user-friendly version 2.0 has been developed. Based on a dynamic simulation (hour by hour calculation), more accurate prediction of cooling load, solar gain, ventilation plant and use of thermal storage (passive solutions) has been possible.

The program is element based, that is. you can describe as many ventilation plants, facades, windows, internal loads and energy plants as desirable. This gives a very flexible program which can simulate a wide variety of buildings, from simple residentials to large complex commercial buildings.

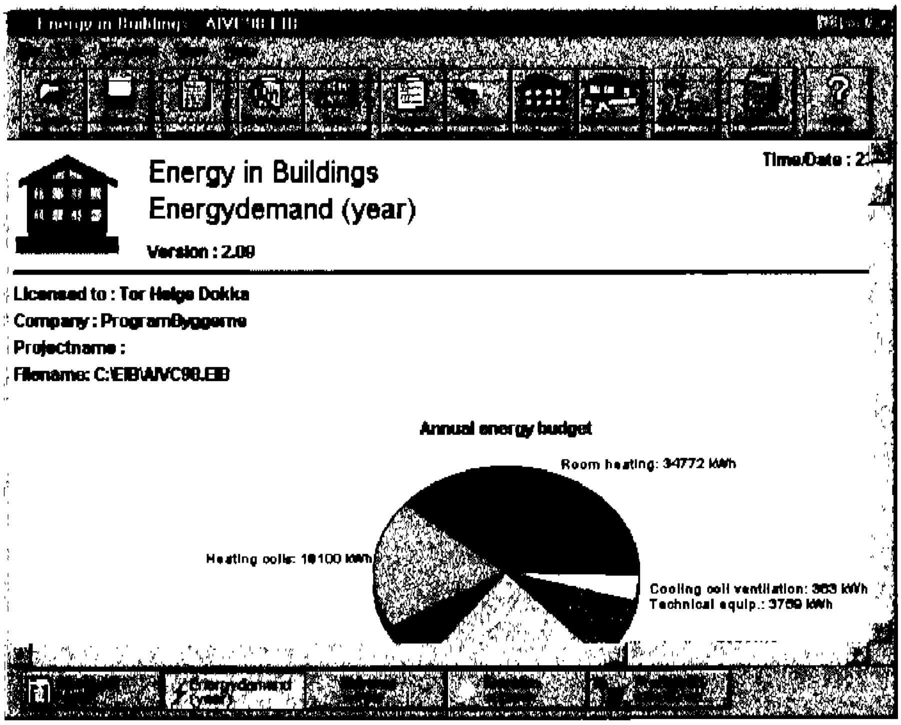

Figure 1 : The user interface in EiB

2 Models

EiB consist of several integrated submodels where the most important is the thermal model. The most significant models in *EiB* is given a description below.

2.1 Thermal model

This is the main "engine" in *EiB*. The simulation of temperature, cooling and heating load on a hourly basis (one hour timestep) is based on a one mass model. This means that all the internal effective heat capacity of a building is "concentrated" in a single mass with the same temperature.

All loads (heat gains, external temperature, solar gains, etc.) is approximated as step-functions (constant during the timestep of one hour). This makes it possible to solve the governing differential equation (from the heat balance) analytically. This analytic expression is used to calculate indoor temperature, and evaluate heating and cooling load to keep a desired setpoint temperature.

2.2 Ventilation and infiltration models

The ventilation model for balanced ventilation is based on submodels for heat exchanger, heating coil, cooling coil and fans. Output from the heat exchanger to the supply air is calculated from the exchangers temperature efficiency. The heat exchanger and heating coil is assumed regulated in sequence, that is : the exchanger delivers its maximum capacity before the heating coil kicks inn. The exchanger can be regulated to a desired setpoint temperature or be non-regulated. The heating coil is regulated after a desired setpoint temperature, and it is assumed that the coil has sufficient capacity to reach this temperature. The cooling coil is also regulated after a desired setpoint temperature, and has sufficient cooling capacity to reach this temperature. If the dewpoint temperature in the supply air dips below the apparatus dewpoint (ADP) of the cooling coil, the necessary cooling capacity to remove the latent heat of evaporation is calculated and added to the "dry" cooling load. The ADP temperature is calculated from the user given efficiency (contact-factor) of the cooling coil. Placements of fans can be either before or after the heat exchanger. The fans are described with a constant temperature rise of the supply and extract air. Its possible to describe multiple ventilation system in the same building(or energy zone).

Infiltration is either described with a user given constant air change per hour (ACH), or calculated with the Lawrence Berkley Laboratory (LBL) model. In the LBL model the infiltration air exchange is calculated from temperature difference inside/outside (stackeffect), the buildings stackheight, wind speed, shielding from terrain, distribution of air leakage, total building tightness and unbalance in the mechanical ventilation system. Air exchange in naturally ventilated buildings with evenly distributed openings can also be estimated with the LBL-model.

2.3 Window and solar model

Heat loss through windows is calculated using the total U-value for the window. Solar heat gain through windows is based on an algorithm which take into account :

- Reduction in solar intensity through panes, described by the total solar heat gain coefficient (direct and secondary heat gain). A semi-empirical formula that estimates the reduction in solar gain with increased incident angle is also implemented
- Reduction in solar gain due to artificial solar shading devices is calculated with a solar shading coefficient
- Geometric models for shading from overhangs, fins, nearby buildings and the horizon is implemented. Shading is calculated on a hourly basis from the sun position in the sky
- Solar flux on facades is calculated from solar altitude, azimuth angle, wall orientation and inclination, atmospheric transmissivity and ground reflection. Solar altitude and azimuth is calculated from time of year, time of day, latitude, longitude and timezone of the location. On cloudy days solar flux reduction (compared to sunny days) is calculated by the cloud cover factor (0-1), and division of direct and diffuse radiation is calculated from the cloud cover index. The solar model is mostly taken from Kimura, \2\ and Duffie&Beckman, \3\.

2.4 Simulation of monthly and annual heating and cooling load

To avoid large hour by hour climate files, a simplified climate and energy simulation model has been developed. Each month is first split into typical sunny days and typical cloudy days. Temperature data for these two days has to be given for each month; mean temperature and temperature amplitude (assumed a sinusoidal variation). The number of sunny days each month must also be known/estimated. Four days each month is simulated : A sunny day with normal operation schedule, a sunny day with weekend/holiday operation schedule, a cloudy day with normal operation schedule and a cloudy day with weekend/holiday operation schedule. Each day is simulated several times to reach diurnal stabile conditions. The temperature, heating and cooling load the last day is recorded. To get the monthly heating and cooling loads the four days are weighted according to the number of sunny and cloudy days and the number of normal operational days and non operational days each month. Annual heating and cooling load is of course the sum of the monthly loads.

2.5 Calculation of energy use, energy costs and emissions

The calculated heating and cooling load is the internal demand of the building, and can be quite different from the total energy consumption of the building. This is due to the efficiency of fuel burning devices, loss in distribution system, and coefficient of performance of cooling machines and eventual heat pumps. In *EiB* different models to account for this is implemented.

Heating

Different energy sources (e.g. heat pump and electric heating) can meet the demand for room heating. Each energy source has to be given a maximum heating capacity and efficiency or COP. The heating load for the heating coil and the tap water, is assumed covered by one of the described energy sources for room heating.

Cooling

The electric energy use for the local cooling system (e.g. chilled ceiling) and cooling coil(s) is calculated from the given COP for the cooling system. It is possible to have multiple local cooling systems and cooling coils with different COP values.

Energy costs

Energy costs for the building are calculated from the gross energy use and energy price per kWh for the different energy sources. Electricity is possible to describe with a seasonal varying tariff, while the other energy sources is given a constant energy price.

Emissions

Emission of CO_2, SO_2, NOx and particles is calculated from the gross energy use for each energy source. Both local generated emissions from fuel burning devices in the building and emissions from energy plants delivering electricity or district heating are taken into account.

2.6 Energy efficiency measurements and profitability analysis

To estimate the profitability of measurements on existing buildings or evaluating different design alternatives on new buildings, an economic profitability model has been implemented in *EiB*. Elements from the existing building is replaced by new elements (e.g. new windows, more efficient ventilation, more efficient boiler, etc.). An annual simulation for both the existing and "new" building are done, and the difference in energy costs are calculated. This difference, that is the annual saved amount, is used to calculate present value, present value quotient, internal rate of return and payback period. Investment, lifetime and increase/reduction in running and maintenance costs for each measure has to be given.

3 Case : Optimisation of energy use in a new office building

To illustrate the use of *EiB*, a new office building in Oslo have been simulated with different design alternatives. All design alternatives comply to the new Norwegian building code (of 1997).

3.1 Demands in the Norwegian building code

The Norwegian building code sets an upper limit for the U-value for walls, floor. roof and windows. The upper limit for commercial buildings are given in table 1.

Table1 : Upper limit for the U-value in the Norwegian building code

	Wall	Floor	Roof	Windows
U-value (W/m^2K)	0.22	0.15	0.15	2.0

Demands for ventilation rates are : 7 l/s per person, and in addition 0.7 l/s per square meter floor area when low emitting building materials is used, 1.0 l/s m^2 where building materials with normal emission is used and 2.0 l/s m^2 when building materials with undocumented emissions is used.

Alternative 1 : ordinary design

A three storey building is located in Oslo. In the ordinary design, alternative 1, it has the following data :

- Heated floor area and air volume : 2100 m^2 and 6300 m^3
- U-values for walls, windows, floor and roof is complying the Norwegian building code (0.22, 1.6, 0.15, 0.15).
- The total window area is 20 % of the floor area (30 % facing South, 30 % facing North, and 40 % East and West)
- Balanced ventilation with cooling coil, heating coil and heat exchanger : 3.33 l/s m^2 in normal operation schedule (12 hours a day) and 1.67 l/s m^2 outside normal operation (night and weekends). Specific fan power : 3 $kW/m^3/s$. Heat exchanger efficiency. : 50 %. Supply temperature : 18 °C (constant).
- Internal loads : Lighting 14 W/m^2 (10 hours a day), Equipment 10 W/m^2 (8 hours a day). On average 150 occupants per day 8 hours a day.
- Heating : Radiators with setpoint temperature 22 °C. Efficiency heating : 100 %.
- Cooling system : Chilled ceilings with setpoint temperature 22 °C. COP cooling system : 2.0.
- Operation : 5 days week, with Christmas and Easter vacation. Normal operation in summer months.

This is a rather usual design of an office building in Norway, where no special effort has been done to design the building energy efficiently, besides complying with the building code. The annual net energy use for this building is simulated to 292 kWh per square meter floor area. The annual energy budget is given in table 2.

Alternative 2 : Energy efficient design
In this design more energy efficient ventilation and night setback is implemented to reduce energy use :

- Use of rotating wheel heat exchanger with a temperature efficiency of 70 %
- Low pressure design of ventilation system with a reduction in specific fan power to 2 $kW/m^3/s$
- Use of building materials with normal emission, reducing the ventilation demand to 2.77 l/sm^2
- More seasonal adapted supply temperature, 19 °C in winter months and 17 °C in summer months
- Night setback of the heating system to 17 °C, 8 hours each night and in weekends.

With this design the net energy use is reduced to 210 kWh/year m^2.

Alternative 3 : Optimum energy design
The third alternative is an energy optimised design, with the following measures :

- Low pressure hybrid ventilation system , with specific fan power of : SFP = 1.0 $kW/m^3/s$. Heating coil is removed. Use of cooling coil with a COP = 3.0
- Low pressure heat exchanger with temperature efficiency of 60 %
- Use of low emitting building materials, and demand controlled ventilation, reduce the ventilation rate to 1.94 l/sm^2, 10 hours a day. Reduces to half the air flow outside normal operation schedule.
- Use of energy efficient lighting with daylight utilisation : 5 W/m^2, 9 hours a day
- Use of energy efficient equipment (computers and printers) : 4 W/m^2, 8 hours a day
- Removal of the local cooling system (chilled ceiling). Acoustic ceilings are removed and flooring material with small heat resistance are used to "expose" the concrete floors. This utilisation of thermal mass reduce cooling demand and overheating in the summer.
- Use of a heat pump system for room heating and tap water heating, with a COP = 3.0. Peak heating load is covered by an electric boiler.

With these measures the net annual energy use is reduced to 113 kWh/m^2. This is the net (internal) energy demand. The gross demand (energy delivered to the building), taking account the COP of the heatpump system, is down to the low value of 57 kWh/m^2yr

Table 2 : The net energy budget for the three design alternatives

Annual net energy budget	Ordinary	Energy eff.	Energy optimised
1. Room heating	65	30	63
2. Heating coils	64	41	0
3. Water heating	12	12	12
4. Fans/pumps	54	33	11
5. Lighting	45	45	15
6. Equipment	27	27	11
7. Room cooling	20	17	0
8. Cooling coils	6	5	1
Sum item 1-8	**292**	**210**	**113**

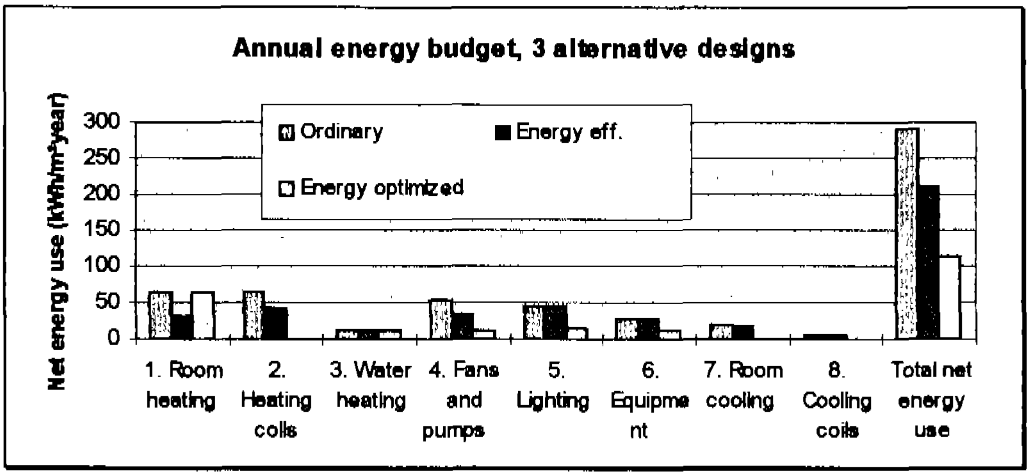

Figure 2 : Net energy budget for the three design alternatives

Profitability

It's obvious that measures in alternative 3 reduce the energy use considerable, but is this measures profitable for the building owner? To answer this question a profitability analysis has to be undertaken.

The measures in alternative 3 are merged into three measures :

Measure one : New hybrid ventilation system, and use of low emitting building materials. The measure reduce the duct and air handling unit costs with approximately 11 000 Euro (99 000 NOK). The costs for more expensive control system and heat exchanger (run around type) is estimated to 15 000 Euro (135 000 NOK). Extra costs in the building fabric due to the hybrid system is estimated to 12 000 Euro (108 000 NOK). The net investment costs compared to the ordinary design is then 16 000 Euro (256 000 NOK). Maintenance cost is assumed to be the same as with the balanced system, and the economic lifetime is set to 15 years.

Measure two : Use of energy efficient lighting and equipment. The daylight utilisation control system is estimated to 23 000 Euro (207 000 NOK). The extra cost for more energy efficient computers, monitors and printers is calculated to 60 000 Euro (540 000 NOK). Maintenance costs is the same as for ordinary design, and economic lifetime is set to 5 years.

Measure three : Use of heat pump system and removal of local cooling system. The reduction in cost of removing the local cooling system is 30 000 Euro (270 000 NOK). Installation of the heat pump system is estimated to 75 000 Euro (675 000 NOK). Maintenance cost for the removed cooling costs and the increased costs for the heat pump system is calculated to be the same. Economic lifetime is set to 15 years.

The interest rate for all measures are set to 4 %, and the electric energy price is 0.055 Euro/kWh (0.5 NOK/kWh).

Table 2 :Profitability of energy efficiency measures, alternative 3.

Energy Efficiency Measures	*Present value (Euro)*	*Present value quot. (-)*	*Internal rate (%)*	*Payback period (years)*	*Energy savings (kWh/yr)*	*Saved amount (Euro/yr)*
Hybrid ventilation	138 424	9.65	86.7	1.2	24 8019	13 889
Heatpump + no local cooling	57 617	2.28	19.0	5.5	16 4813	9 230
Reduced internal loads	-62 575	0.25	-31.8	32.8	8 1927	4 588
Sum measures	133 466	1.93	13	5.9	49 4759	27 707

4.0 Discussion, conclusions and further work

- There is a big demand for user friendly simulation programs that evaluate different designs to get energy efficient and environmental friendly buildings
- Programs has to be designed with the practical designers needs in mind
- Simulations shows that other measures than better insulation and high efficiency heat exchangers can be effective in reducing the energy use in commercial buildings
- Simulations and profitability analysis indicate that measures like : daylight utilisation, use of thermal mass, hybrid ventilation and demand controlled heating, ventilation and cooling, is cost effective.
- To be used extensively, designers also has to be sure that these programs produce accurate results. It is therefore very important to validate the models in the best possible manner. *EiB* is soon to be tested against the BESTETST procedure, \4\. This validation procedure will be published in a later paper.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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MODELLING SUPPLY DEVICES IN ORDER TO PREDICT IMPROVEMENTS IN INTERNAL AIR QUALITY

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Modelling Supply Devices in Order to Predict Improvements in Internal Air Quality

1 Synopsis

The air distribution effects of floor mounted swirl diffusers are investigated and described in this paper. Results are based on a case study of an office typical of those in urban commercial environments. The effects of the swirl applied to the supply air as well as temperature differentials between supply and room air are explored. The investigation is restricted to situations where cooling is required.

The results of the work, which is undertaken by way of CFD analysis, are presented in terms of appropriate ventilation effectiveness parameters. It is observed that variations to the form of swirl devices, changes to temperature differentials and the presence of internal heat sources combine to affect the internal air distribution to such an extent that its nature may vary from displacement to characteristics approaching those of mixing systems. Air quality at head level can change from very good to poor in reponse to relatively small changes in some of the parameters.

2 List of Symbols

Symbol		Units
ε_a	air change efficiency	
ε_p	local air change index at point p	
τ_n	nominal time constant	s
$\bar{\tau}_p$	local mean age of air at point p	s
$<\bar{\tau}>$	room mean age	s

3 Introduction

It is often assumed that displacement systems are superior to mixing systems in providing good air quality in the occupied zone of a ventilated space. This is especially so in large spaces when there is a cooling requirement. In such cases, fresh air is normally supplied at floor level so that it displaces heat and pollutants upwards into the unoccupied space above. Such an arrangement helps to minimise the air flow rate, thereby saving energy. However, the distribution of fresh air in such a system, and the air quality in the occupied zone, are affected by a number of factors, including:-

(i) the discharge characteristics of the floor mounted diffusers;

(ii) the presence of any momentum generating sources within the space, especially heat sources;

(iii) obstacles at or near ground level.

There are two types of floor mounted diffuser in common use in displacement systems. The tower type consists of a perforated cylinder, typically one metre high, supplied by ductwork, and located around the perimeter of the space and at other convenient points. Exit air velocities are of the order of 0.25 ms^{-1}. The flush type consists of circular grilles arranged on a grid over the whole floor area, and most often supplied by an under-floor plenum. Exit air velocities are of the order of 0.9 ms^{-1}, and these diffusers are often provided with swirl vanes in order to enhance their ability to create a plane of upward moving air. Indeed, swirl supply terminals are frequently specified for the provision of mechanical ventilation and, depending on the terminal exit angle and temperature difference, the characteristics of the resulting air distribution may approximate to either displacement or mixing ventilation. Neither type of diffuser can produce a perfect displacement air flow; this would require air to be supplied evenly over the whole floor plane, as is the case in laminar flow clean rooms.

Because the velocity and momentum of the air at entry are low, the second and third factors are relatively more important in displacement systems than in mixing systems. This became apparent in a recent study of the application of ventilation effectiveness parameters to some design studies [1]. The present study, therefore, examines the effect on air quality of the flow characteristics of the flush mounted floor diffusers with varying degrees of swirl, and the presence of heat sources. This is done by studying a simplified but typical design problem, and using the appropriate ventilation effectiveness parameter to evaluate air quality.

4 The Design Case and the Method of Assessment

The internal space on which the study is based is an office located in a typical urban environment. The office, which is shown in Figure 1, is 4.8m by 3m in plan and 3.2m high. One of the 3m x 3.2m walls is an external wall with a single glazed window, which extends the full 3m width of the room and vertically 2.2m from a 1.0m high cill. All other walls, the floor and ceiling are assumed to be backed by similar spaces at the same temperature. The office has two occupants and two PC's as heat sources. Ventilation is provided by six 150mm diameter floor mounted swirl diffusers and these are balanced by two ceiling mounted extract units. The dimensions and performance data for the diffusers were chosen to be typical of commercially available units. The ability of the system to produce good air quality is assessed by determining velocity vectors, temperatures and ventilation effectiveness throughout the space. The most appropriate ventilation effectiveness parameter in this situation was considered to be the Local Air Change Index (LACI, see appendix A for definitions), as contours of this parameter give a more realistic picture of the distribution of fresh air than either of the other parameters.

The discharge characteristics of the diffusers were investigated by testing the effect of different degrees of swirl. Rather than associate a swirl number with the diffuser, it was found to be more convenient for modelling purposes to specify a discharge angle. An angle of zero is taken to be the case where the air leaves the diffuser vertically, perpendicularly to the floor surface, in which case it may be expected to behave as a simple circular jet with zero swirl.

The effect of varying the temperature differential of the air at the inlet had, of necessity to be linked to adjustments in the external temperature, as these two factors interact. Each combination of factors was arranged to give approximately the same average space temperature.

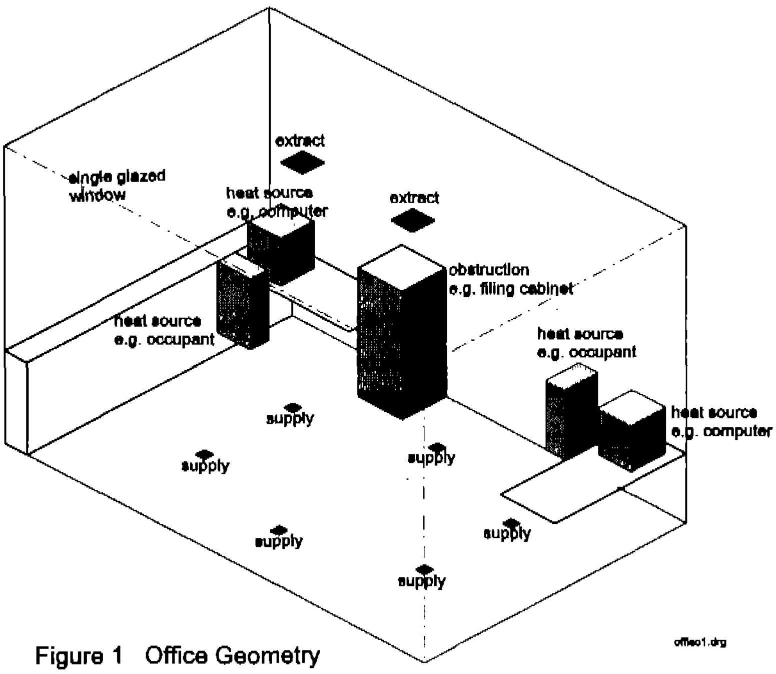

Figure 1 Office Geometry

5 Modelling Procedure

The problem was treated by means of Computational Fluid Dynamics, using the 'Flovent' CFD model by Flomerics. This provided the velocity vectors and temperature contours. The LACI contours were obtained by post-processing the Flovent solution, as has been previously reported [2], except that the post-processing software has now been rewritten in order to remove restrictions on the size of the model, and to compute contaminant removal effectiveness as well as air change efficiency parameters. The computed parameters are returned to Flovent for subsequent analysis and display.

The method of modelling of the diffusers required careful consideration. 'Flovent' is restricted to a rectangular Cartesian grid, whereas the diffusers are circular, and the addition of swirl imparts a circular rotational pattern to the jet. This problem was overcome by dividing the equivalent square representing the diffuser into four quarter segments through which air was supplied vertically upwards. A two dimensional horizontal thrust was then applied to the air as it left the diffuser in order that the resultant was directed at 45^{o} to the co-ordinate axes in plan and at the required angle to the vertical, and with the required discharge velocity of $0.9ms^{-1}$. This created the desired type of jet pattern, as is shown in Figure 2 by the velocity vectors on a horizontal plane 0.125m above floor level for a typical configuration.

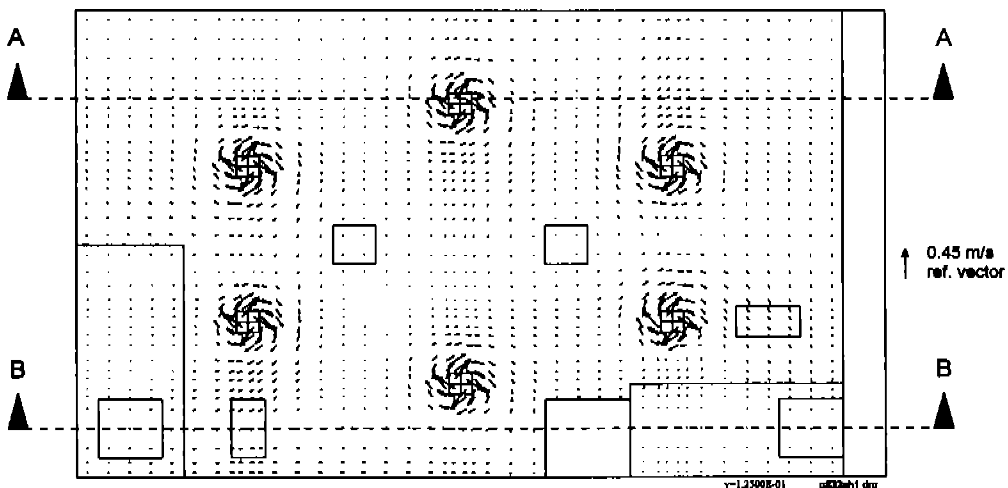

Figure 2 Velocity Vectors 0.125m. Above Floor Level
(Case 1: Supply at 60 Deg. from Vertical,)

For the purposes of this study, analysis has been restricted to a flow rate of $0.00952kgs^{-1}$ through each of the 6 floor mounted supply terminals, corresponding to a nominal time constant of 16 minutes, or 3.8 air changes per hour. The consequences of a variety of swirl angles, external ambient temperatures and supply air temperatures have been explored. Table 1 summarises the combinations which have been tested and are reported here. All are cooling cases with 180W of internal gains.

Table 1 Summary of Test Cases

Case no.	Diffuser discharge angle	Condition	Internal heat gains	External ambient temperature	Supply air temperature	Results plotted in figure nos.	Air Change Efficiency %
1	60°	Cooling	180W	25°C	19°C	3,4,5	77.8
2	40°	Cooling	180W	25°C	19°C	6	58.7
3	20°	Cooling	180W	25°C	19°C	7,8	51.7
4	60°	Cooling	180W	23°C	20°C	9	51.9
5	60°	Cooling	180W	21°C	21°C	10	53.9

6 Consideration of Results

The air distribution pattern from the diffusers is determined by three competing effects:

(i) the inertial force due to the jet exit velocity, tending to produce a vertical plume,
(ii) the buoyancy force due to the temperature difference, causing the air (in all the cooling cases) to form a layer at floor level, and
(iii) the coanda effect, also pulling the air towards the floor surface.

The addition of swirl is expected to reduce the effect of the first of these, and to help to create a displacement flow. As the swirl angle decreases, there may be a point at which the flow switches from a stratified displacement pattern to a predominantly vertical jet. However, entrainment, recirculation, internal heat sources, and internal room geometry all complicate the situation. The results for each of the cases given in Table 1 have been plotted to show contours of LACI, as this gives the distribution of fresh air. Referring to figure 2, the vertical sections are plotted either on plane A-A where there is one diffuser and no disturbing features, or on plane B-B through one of the heat sources. Table 1 also gives the air change efficiency for each case. It should be noted that an air change efficiency of 100% correponds to perfect displacement ventilation, whereas 50% corresponds to perfect mixing ventilation.

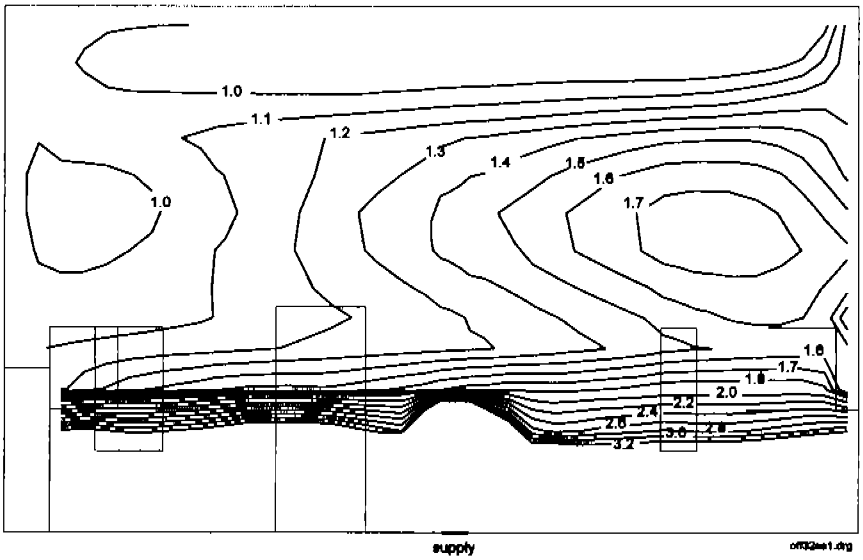

Figure 3 Case 1, L.A.C.I. Contours on Plane A-A.
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Suppl19 Deg.C.)

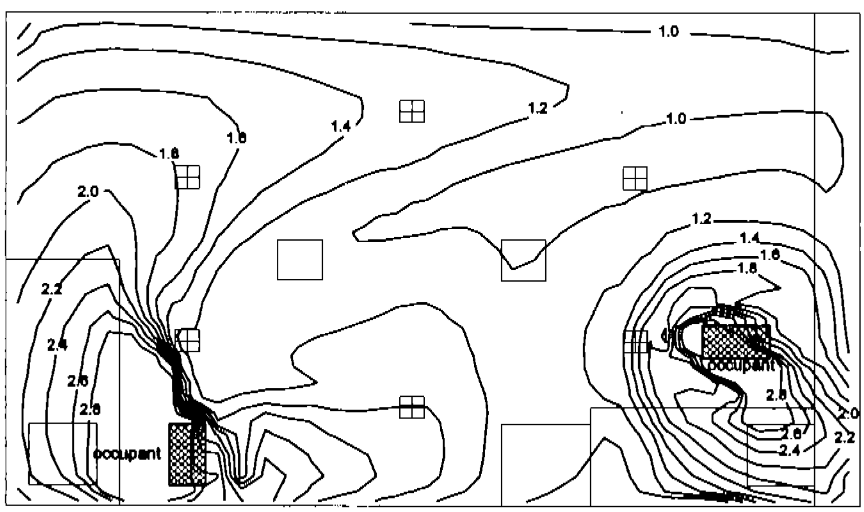

Figure 4 Case 1, L.A.C.I. Contours1.5m Above Floor Level.
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.)

Case 1 - swirl angle 60° - external temperature 25℃, supply temperature 19℃.
Figure 3, a section at A-A, shows a clear displacement pattern. Incoming air is trapped at floor level, with substantially horizontal LACI contours decreasing in value with height. Figure 4 shows the LACI at 1.5m above floor level, equivalent to head height for a seated person, with

values greater than 1 over most of the plane. It also shows islands of much higher values around the two heat sources (occupants), showing that the plumes induced by these sources are sucking fresh upwards from below. The vertical section at B-B in Figure 5 shows one of the plumes more clearly, but also shows that it induces a slight recirculation which brings some old air back down to lower levels. Despite this it will be seen from Figures 3, 4 and 5 that high values of LACI extend over much of the volume. The air change efficiency is very high at 77.8% indicating a successful displacement ventilation system.

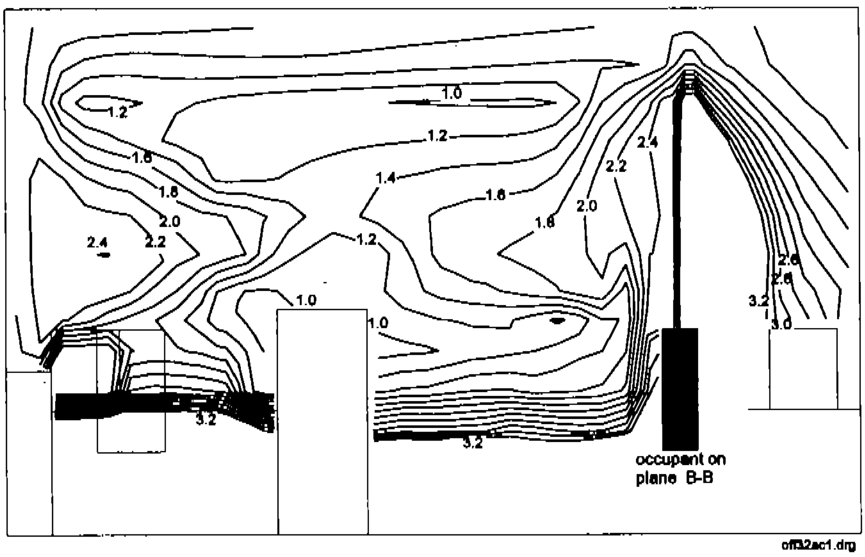

Figure 5 Case 1, L.A.C.I Contours on Plane B-B
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.)

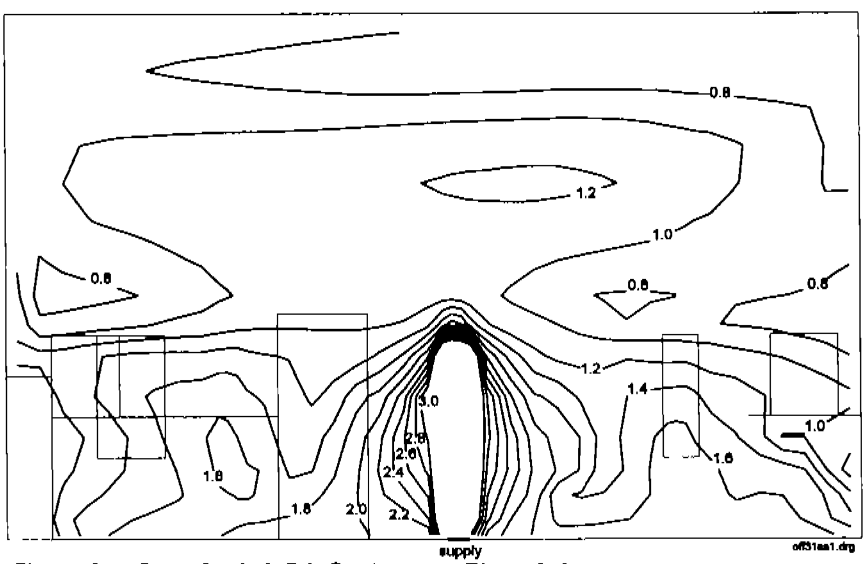

Figure 6 Case 2, L.A.C.I. Contours on Plane A-A.
(Supply at 40 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C)

Cases 2 and 3 - swirl angles 40° and 20° - external temperature 25 °C, supply temperature 19 °C.

Figure 6 shows that at a swirl angle of 40° the air enters as a jet before stratifying at about head height. The LACI contours now have a vertical bias, with high values clustered around the jet and much lower values at head level. Figure 7 for a swirl angle of 20° is very similar, suggesting that the expected transition has taken place between 60° and 40°. Figure 8, which is a horizontal section at head height for the 20° case, is significantly different to Figure 4 for 60°, and shows islands of high LACI above the diffusers.

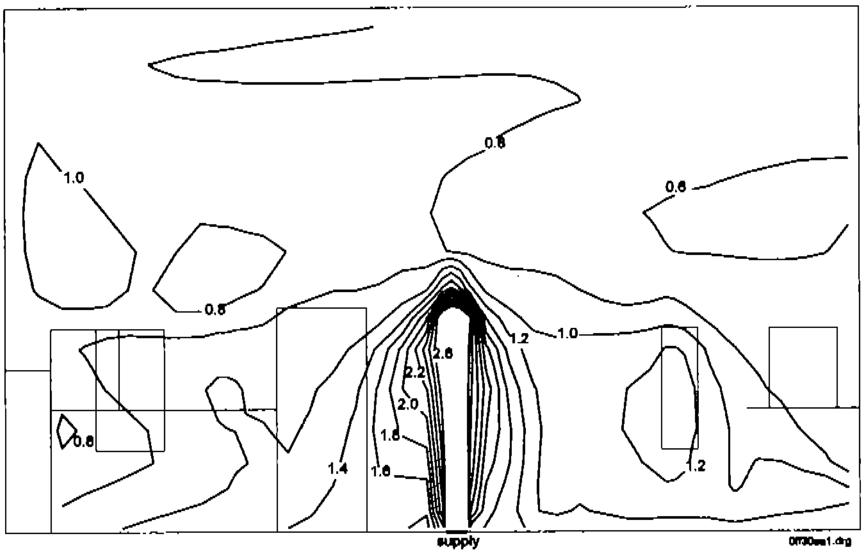

Figure 7 Case 3, L.A.C.I. Contours on Plane A-A.
(Supply at 20 deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.

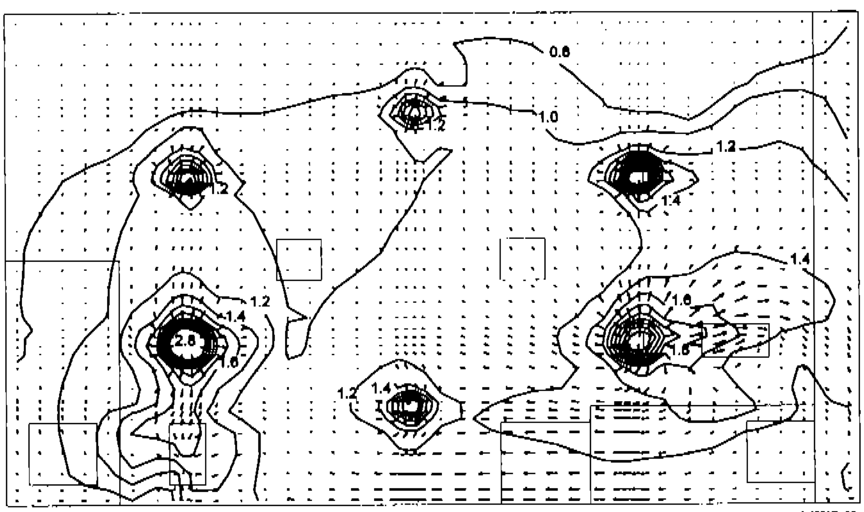

Figure 8 Case 3, L.A.C.I Contours and Velocity Vectors 1.5m Above Floor Level
(Supply at 20 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg. C.)

The air change efficiency values of 58.7% and 51.7% for Cases 2 and 3 respectively are considerably less than for Case 1, and when considered in conjunction with the LACI distribution are indicative of a mixing system rather than a displacement sytem.

Figure 9 shows a section at B-B for Case 3 and, like Figure 5, it shows a vertical plume of air convected upwards. However, in this case, the LACI at the breathing zone is only approximately 1.4 as opposed to a value in excess of 3.

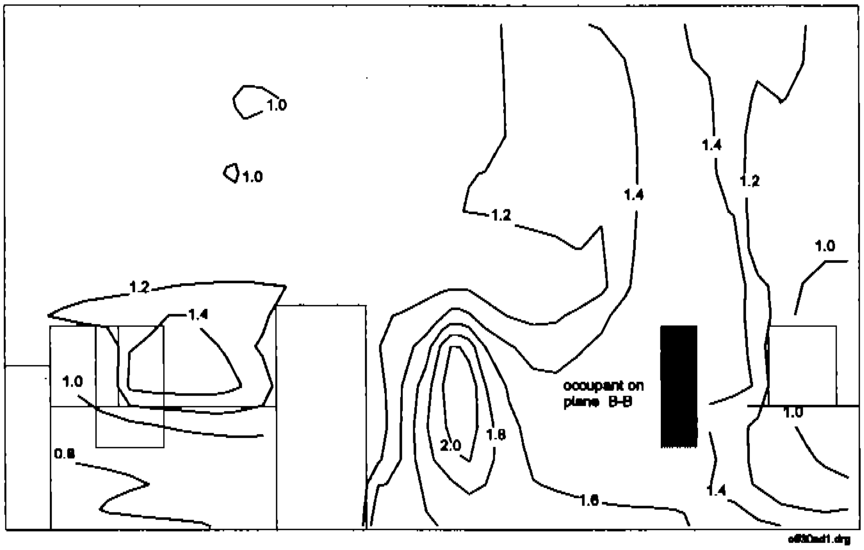

Figure 9 Case 3, L.A.C.I. Contours on Plane B-B
(Supply at 20 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg. C.)

Case 4 - swirl angle 60° - external temperature 23 °C, supply temperature 20 °C.
The distinctive displacement pattern of Figure 3 is partly due to the negative buoyancy of the 6°C external to internal temperature difference. Reducing this to 3°C, as is shown in Figure 10, compresses the contours downwards with values of considerably less than 1 at head level, which indicates poor air quality and is the opposite of what might be expected. Stratification is much more apparent than in Case 1 (Figure 3) and a consequence of the large volume of air with an LACI of less than 1 is that the air change efficiency of this case at 51.9% is much less than that of Case 1. It appears that the plumes of warm air over the heat sources are pulling a large quantity of fresh air to high level, effectively short-circuiting the rest of the space.

Case 5 - swirl angle 60° - external temperature 21 °C, supply temperature 21 °C.
With a temperature differential of zero, the only buoyancy forces are those due to the internal heat sources. It will be seen from Figure 11 that the result is similar to Case 4 although the contour lines of low LACI representing old air are less compressed. Again, it is probable that the heat sources are providing a convective route for fresh air to high level resulting in large volumes of old air near the centre of the room and so leading to a relatively low air change efficiency of 53.9%.

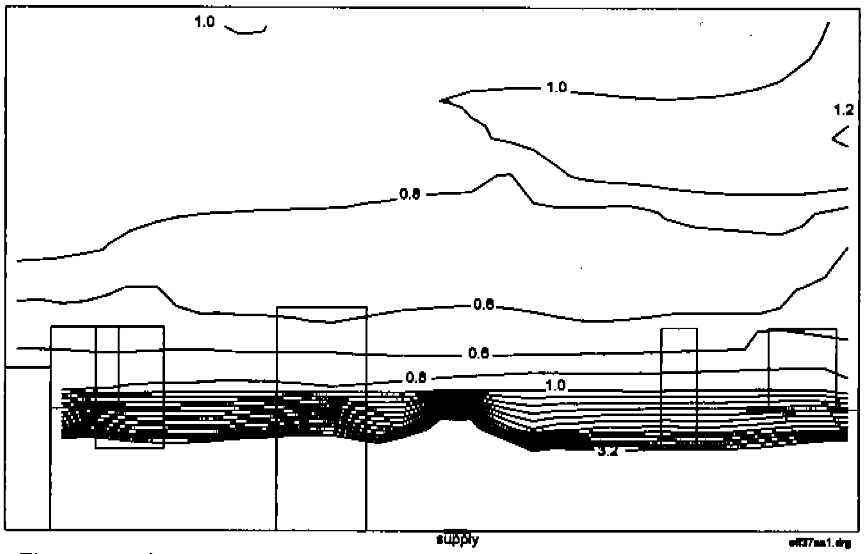

Figure 10 Case 4, L.A.C.I. Contours on Plane A-A
(Supply at 60 Deg. from Vertical. Ambient 23 Deg.C, Supply 20 Deg.C.)

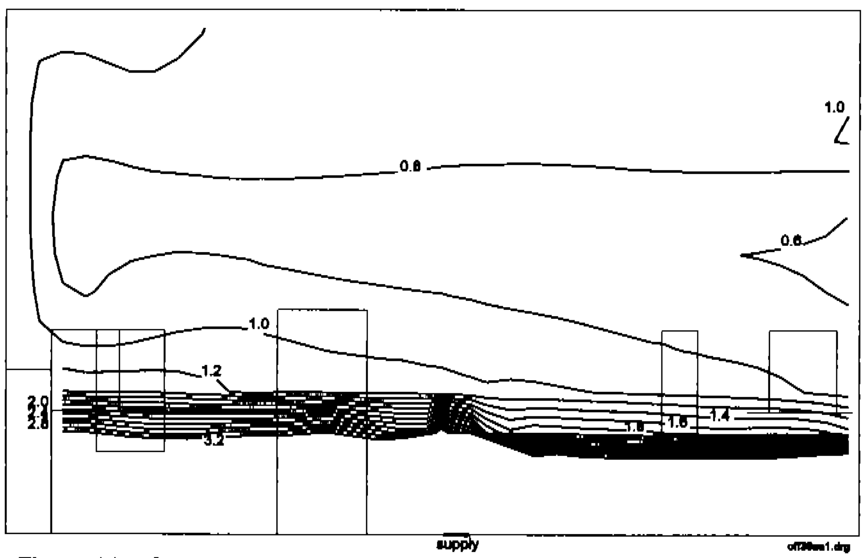

Figure 11 Case 5, L. A. C. I. Contours on Plane A-A.
(Supply at 60 Deg. from Vertical. Ambient 21 Deg.C, Supply 21 Deg.C.)

7 Conclusions

The first conclusion is that the addition of swirl to floor mounted diffusers enables them to create a true displacement ventilation pattern, provided that the amount of swirl is sufficient. Comparison of cases 1,2 and 3 show that in this particular example, a swirl angle of at least 60° to the vertical is necessary to overcome the inertial force of the jet.

The second conclusion is that even relatively low power heat sources can have a significant effect on the movement pattern of air from low to high level. This is because the pool of fresh air at low level has very little momentum. The plume which is set up over the heat source contains most of the fresh (i.e. 'young') air. This is most noticeable in Case 4, where the LACI has high values within the plume, but low values elsewhere. If the plume is due to an occupant, then it can be argued that the occupant is effectively bathed in a plume of high quality air. This may be a particularly fortunate characteristic of displacement ventilation. However, if the heat is due to some other type of source, the net effect could be for the occupant to find himself breathing the poor quality air at head level in other parts of the space.

The overall conclusion of this particular case study is that a system which is designed to provide a displacement pattern may sometimes behave quite differently, with a dramatic change in the fresh air quality at head level.

8 Appendix - Definition of Ventilation Effectiveness Parameters

i Local Mean Age of Air, $\bar{\tau}_p = \int_0^\infty t.A_p(t).dt$ where $A_p(t)$ is the age distribution curve for air arriving at point *p*.

ii Room Mean Age, $<\bar{\tau}>$ The average value of the local mean ages of air for all points in a room.

iii Local Air Change Index (LACI), $\varepsilon_p = \dfrac{\tau_n}{\bar{\tau}_p}$ where τ_n is the nominal time constant of the room. The nominal time constant is the reciprocal of the fresh air change rate. A value of LACI greater than 1 indicates that a point is receiving air more efficiently than it would with a perfect mixing system.

iv Air Change Efficency, $\varepsilon_a = \left(\dfrac{\tau_n}{2<\bar{\tau}>}\right)100\%$

9 References

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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Introduction to the Design of Natural Ventilation Systems Using Loop Equations

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Synopsis

The design of natural, including passive, ventilation systems assumes one of two generic forms: the nasty design problem where the designer seeks to size ventilation openings given climatic conditions and thermal comfort criteria or the nice design problem where the designer seeks to size ventilation openings given climatic conditions, indoor temperature distributions, and specified airflow rates – presumably determined from separate thermal or air quality considerations. The nasty form of design demands consideration of the complex dynamic coupled interaction of a building's airflow systems, thermal characteristics and airflow and thermal excitations – a challenge that only the most advanced simulation programs have been able to address and one were few, if any, can claim real expertise at this time. The nice form of ventilation design, on the other hand, is quite tractable and may be approached using existing, relatively simple and intuitively direct theory. Yet it is commonly perceived to be of a nasty character demanding iterative and approximate techniques for its solution.

This paper will present an 'exact' approach to the *nice* design problem that may be considered to be a more complete formulation of the approximate approach recently published in the CIBSE Application Manual AM10:1997 [1]. The approach presented is based on so-called *loop equations* that are commonly used in flow network simulation in the hydraulics field but have been largely ignored in the building ventilation field. It allows direct sizing of a variety of airflow components and the direct and unambiguous consideration of both stack-driven and wind-driven flows without resorting to simplifying approximations. Yet, the approach is developed in such a way as to enable building designers to identify a full range of feasible design configurations so that other, nontechnical design constraints may be included in the process of seeking a design solution – an example of such a design scenario will be presented.

Introduction

The benefits of ventilation have been know to humankind for all time – as a means to displace stale or contaminated air and to cool advectively or convectively. We can easily imagine the design of ventilation systems for air quality control began when the first individual attempted to redirect the smoke from a cooking fire to avoid its personal impact and, for cooling, when another moved to a shaded hill top location to gain the benefits of cooling breezes. With this extraordinarily long history in mind one should reasonably ask why do we now consider the design of natural ventilation systems such a challenge? The short answer to this questions is that now the stakes are higher – there is a significant energy penalty, and the consequent environmental impact, associated with over-ventilation [2, 3] and, of course, comfort and potentially health problems with under-ventilating buildings. Hence we seek to ventilate buildings with greater control and precision than in the past.

How then does a building designer or architect approach the design of building ventilation systems to achieve these more demanding objectives? Five distinct design tasks can be identified:

- *Establish Global Geometry* The designer must initially set the global geometric configuration of the system – e.g., siting of the building and landscape configuration, overall building form, and positions of fresh air inlets and stale air exhausts.
- *Establish System Topology* The designer must layout the airflow paths from inlet to outlet that will achieve the desired airflow objective – e.g., cooling, moisture control, CO_2

control, etc. – and select the types of airflow components – e.g., windows, doors, vents, etc. – that will provide the control of airflow desired.

- *Component Sizing* The designer must then size the components of the airflow system considering reasonable and relevant climatic conditions – the *design conditions* – and appropriate *design criteria*.
- *Control Strategy* The designer must develop a strategy to control the ventilation flow rates to achieve the objective design criteria and select hardware and, possibly, software to implement the strategy.
- *Detail and Assembly* Finally, the designer must develop detail and assembly drawings so that the system can actually be built.

This paper addresses only Task 3 – the task often identified as "design" in the engineering community – up-to-date guidance for Tasks 1 and 2 for both domestic and non-domestic buildings may be found in the recent publications of the British Research Establishment (BRE) and the Chartered Institution of Building Services Engineers (CIBSE) [1, 4].

In the allied fields of building technology, component sizing methods are invariably based on the same theoretical principles used to predict building performance – i.e., to *simulate* building behavior – although, recognizing the time and financial constraints designers face, these methods are generally simplified inversions of the mathematical methods used for whole system simulation. In the simulation of building ventilation systems, two classes of analytical problems may be distinguished – simple *network airflow analysis* where ventilation airflows are predicted for selected climatic conditions given temperature distributions within the building and the more complex *coupled network airflow/thermal analysis* where both building temperature distributions and airflows are predicted for selected climatic conditions.

Consequently, we may distinguish two generic classes of component sizing problems – the *nasty* and the *nice* "design" problems as articulated above in the **Synopsis**. The difficult *nasty* "design" problem was considered in an earlier paper [5]. The more tractable *nice* form of ventilation "design", on the other hand, is the object of this paper.

Irving and his colleagues have developed an approach to the *nice* "design" problem [1, 6, 7] that is coincidentally similar to an approach I've presented to architectural design students at MIT and Yale. Both approaches are based on the same fundamental theory – the basis of methods of network airflow analysis – and may be considered to be formulated as simplified *loop equations* for these networks. For the professional building designer, however, there is little penalty and much to be gained through the use of the more general and complete method based on an <u>exact</u> formulation of *loop equations*. This paper will present this formulation and, building on the work of Irving and his colleagues, demonstrate its application and utility.

Basic Theory

A building system may be idealized as a collection of control volumes (e.g., rooms, zones, or joints of a duct network system) linked by discrete airflow components (e.g., windows, doors, cracks, or duct segments of a duct network). For the representative *macroscopic idealization* illustrated below, a single room is modeled with an inlet opening at *a*, a room outlet grill at *b*, and a ducted stack running from *b* to *c* terminated by a stack terminal device at *c* – the control volumes considered in this case are the room and the stack duct and the discrete flow components: the inlet, outlet, and stack terminal devices.

Discrete locations or nodes are identified – e.g., the black dots above – with which values of temperature and pressure are specifically defined and the form of the variation of temperature and pressure within the control volumes is assumed and directly related to the nodal values – e.g., most commonly, but not necessarily, the temperature is assumed to be uniform and the pressure assumed to vary hydrostatically within the control volume. One of several pressure-flow relations is then associated with each discrete flow component to complete the building idealization task.

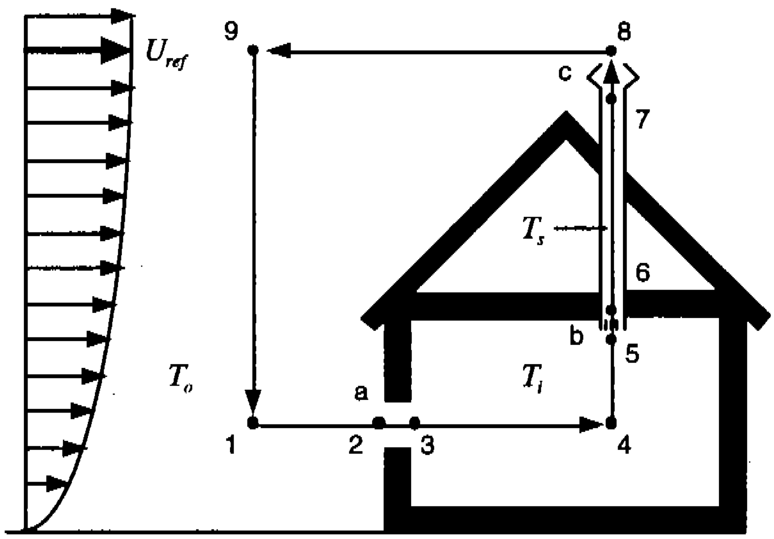

Figure 1 Representative macroscopic idealization of a building.

With such a *macroscopic idealization* in hand, systems of nonlinear algebraic equations – the *node equations* – can then be formed, by demanding the conservation of air flow in-to and out-of each control volume, and solved to determine nodal pressures and, subsequently, the airflows in each of the linking discrete airflow components.

Alternatively, one may approach the analytical problem by summing equations describing the changes of pressure as one traverses a continuous loop that follows possible airflow paths from node to node in the building idealization returning to the original starting node. With reference to Figure 1, one such loop is possible following the nodal path 1-2-3-4-5-6-7-8-9 and back to 1. These changes of pressure must, of course, add up to zero upon completion of the loop. While these *loop equations* are a bit tricky to form automatically for simulation purposes, they are well-suited to design investigations since the designer must design the loops – i.e., specify the global geometry and topology of the ventilation airflow paths that define the loops and size the discrete components along the path.

Wind & Hydrostatic Pressures

The equations needed to form the loop equations for a given building idealization are familiar. At surface locations external to a discrete flow component "*e*" wind-driven pressures P_{eo} are related to the ambient pressure $P_o(z_e)$ at the component level z_e and the dynamic pressure of the oncoming wind defined in terms of a wind pressure coefficient C_{Pe} and an associated dynamic pressure of a reference wind speed U_{ref}:

$$P_{eo} = P_o(z_e) + C_{Pe}\frac{\rho U_{ref}^{\ 2}}{2} \tag{1}$$

Pressure changes $\Delta P_i(\Delta z_{ij})$ due to elevation changes Δz_{ij} are defined by the discrete form of the hydrostatic equation for the usual assumption of uniform temperature distributions:

$$\Delta P_i(\Delta z_{ij}) = -\rho_i g \Delta z_{ij} \qquad (2)$$

or by the integral form for nonuniform temperature distributions:

$$\Delta P_i(\Delta z_{ij}) = -\int_{z_1}^{z_2} \rho_i(z) g dz \qquad (3)$$

Flow Component Relations

Finally, the pressure change, ΔP_e, along a discrete flow component "e" is related to the volumetric air flow rate Q_e and a characteristic design variable associated with the component ϕ_e as – in general, functional notation:

$$\Delta P_e = f(Q_e, \phi_e) \qquad (4)$$

Here, four pressure-flow relations will be considered, the first, useful for larger openings such as windows and doors experiencing unidirectional flow, is based on the orifice equation:

Orifice Relation $$\Delta P_e = \frac{\rho Q_e^2}{2C_d^2 A_e^2} \text{ where } \phi_e \equiv A_e \qquad (5)$$

where the discharge coefficient C_d may be expected to have a value close to 0.60 for flow intensities of interest in most situations. The second relation, often empirically fitted to measured behavior of adventitious openings, is based on the so-called power law relation:

Power Law Relation $$\Delta P_e = \frac{Q_e^{1/n}}{C_e^{1/n}} \text{ where } \phi_e \equiv C_e \qquad (6)$$

The third relation, appropriate for very narrow crack openings, is based on a linear relation:

Linear Relation $$\Delta P_e = \frac{Q_e}{C_e^*} \text{ where } \phi_e \equiv C_e^* \qquad (7)$$

Finally, the fourth relation is based on the familiar relation for flow in ducts:

Duct Flow Relation $$\Delta P_e = \frac{1000 fL}{D_h} \frac{\rho Q_e^2}{2A_e^2} \text{ where } \phi_e = D_h A_e^2 \qquad (8)$$

where f is a friction factor that varies from 0.01 to 0.05 for likely flow intensities encountered, L is the length of the duct, D_h is the hydraulic diameter, and A_e is the duct cross-sectional area. Details relating to these relations may be found in Awbi [8] and the ASHRAE Handbook of Fundamentals [9].

Bousinesq Approximation

The subscript on the air density variable ρ in Equations 1 and 5 has been omitted to acknowledge that the uncertainty associated with the use of these equations does not warrant precision in the specification of air density. When applying Equation 1 and 5 sufficient accuracy, especially for design calculations, may be obtained by using an air density representative of the range of values associated with the problem at hand (e.g., a mean value). On the other hand, precision is required in the hydrostatic equations, Equations 2 and 3, thus the subscript is retained. These approximations, which are a discrete form of the Bousinesq assumption used in *microscopic* analysis, are not theoretically required but simply ease the burden of computation.

Loop Equations

Using the equations enumerated above, we may directly form the loop equations for a given building idealization. For the representative idealization shown in Figure 1, we begin at the ambient pressure node 1, P_{1o}, and move forward around the loop adding first the increase due to the wind acting on the wall at node 2, then the pressure drop ΔP_a along the flow link a, the hydrostatic decrease resulting from the elevation change from 4 to 5, and so on to obtain:

$$+C_{P2}\frac{\rho U_{ref}^{2}}{2}-\overbrace{\Delta P_a}^{\text{window}}-\rho_l g\Delta z_{45}-\overbrace{\Delta P_b}^{\text{room outlet}}-\overbrace{\Delta P_{bc}}^{\text{stack duct}}-\rho_s g\Delta z_{58}-\overbrace{\Delta P_c}^{\text{stack terminal}}-C_{P8}\frac{\rho U_{ref}^{2}}{2}+\rho_o g\Delta z_{91}=0 \quad (9)$$

where ρ_l and ρ_s are the air densities in the room and the stack respectively. Substituting the flow relations for each of the components – e.g. here using the orifice relation for the window, the power law relation for the room outlet and the stack terminal, and the duct relation for the stack duct – yields the final explicit form of the loop equation. For a more complex building additional flow paths and, hence, loop equations may be formed.

Presented as in Equation (9), the loop equations may seem rather formidable, but on closer examination it may be seen that these equations involve simply a) a summation of hydrostatic changes that define a stack-driven pressure difference ΔP_s, b) windward and leeward pressures that define the wind-driven pressure difference ΔP_w, and c) a summation of pressure drops of each of the flow component ΔP_l along the loop that has the general form:

General Form

$$\overbrace{\sum f(Q_l)}^{\Delta P_l}=\overbrace{g\sum \rho_i\Delta z_{ij}}^{\Delta P_s}+\overbrace{(\Delta C_P)\frac{\rho U_{ref}^{2}}{2}}^{\Delta P_w} \quad (10)$$

where $f(Q_l)$ is positive for *forward* flow along the loop, Δz_{ij} is positive for increases in elevation along the loop, and ΔC_p is the algebraic sum of pressure coefficients with C_P summed positively when traversing the loop from the exterior to a wall surface location and negatively other wise.

For loops involving orifice flow links only, the loop equations assumes the following form:

Orifice-Only Loop

$$\overbrace{\sum\frac{\rho Q_l^{2}}{2C_d^{2}A_l^{2}}}^{\Delta P_l}=\overbrace{g\sum \rho_i\Delta z_{ij}}^{\Delta P_s}+\overbrace{(\Delta C_P)\frac{\rho U_{ref}^{2}}{2}}^{\Delta P_w} \quad (11)$$

This relation is essentially identical to that developed by Irving and his colleagues although their model was limited to control volumes with single inlets and outlets [1, 6, 7].

At any given stage of design all variables in a loop equation will be known, with the exception, of course, of the flow component design parameters (e.g., the opening areas A_ℓ for orifice components), and may be substituted directly. In general, however, one will not be able to define a sufficient number of loop equations to determine a unique solution. In natural ventilation design many alternative *feasible design solutions* may be identified. This apparent problem, we shall see, presents the designer with the opportunity to include nontechnical considerations in the design process if the problem is approach correctly. This may be made clear by consideration of a more complex example.

CIBSE Example Application

Consider the design of the hypothetical three story building, based on the example considered by Irving [1, 6, 7], sketched below. In this sketch, discrete flow components are identified by alphabetic labels – i.e., components *a, b, ... g* are distinguished – and four zones are considered – zones G, 1, 2, and s (i.e., stack). In addition, relevant pressure nodes and elevation increments are indicated. This example differs from that presented in the CIBSE application manual on natural ventilation [1] in that internal flow resistances – i.e., components b, d and e – are included.

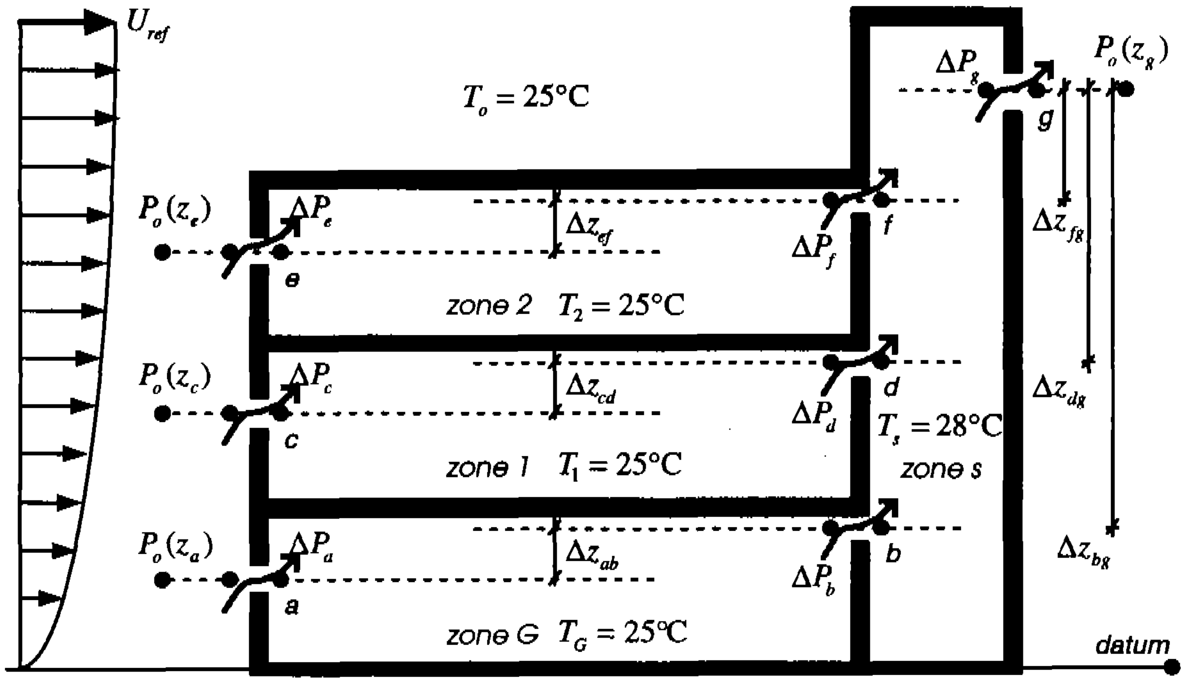

Figure 2Problem based on CIBSE example problem 1 [1]

Global Geometry & Topology

At this stage in design the global geometry and the ventilation system topology – i.e., the three flow paths, *a-b-g*, *c-d-g*, and *e-f-g* – have been defined by the designer. Regarding the former each story of the building is 3.25 m high, the inlet openings are centered 1.85 m above floor level, the room outlets at 2.85 m, the stack exhaust is centered 11.5 m above the ground, and each room has a volume of 322.9 m^3.

Climatic Conditions & Design Criteria

The climatic conditions and design criteria will be based on Irving's data [1]. The design is to provide a ventilation flow rate of 5 ACH for an outdoor ambient temperature of 25°C and a reference wind speed of 3 m/s. Windward wind pressure coefficients will be assumed to be +0.58 and leeward –0.20, internal room temperatures will be taken to be 25 °C and the mean air temperature of the stack will be assumed to be 28°C.

To transform the ventilation design objective into specific component airflow rates we simply demand continuity of flow – here, $\rho_o Q_a = \rho_G Q_b = \rho_o Q_c = \rho_1 Q_d = \rho_o Q_e = \rho_2 Q_f = \rho_s Q_g / 3$ – or, applying the discrete Bousinesq approximation, for the desired 5 ACH (0.448 m^3/s per room) ventilation flow rate the specific component flow rates required are:

$$Q_a = Q_b = Q_c = Q_d = Q_e = Q_f = 0.448 \text{ m}^3/\text{s} \ \& \ Q_g = 1.344 \text{ m}^3/\text{s} \tag{12}$$

Loop Equations

To proceed, the designer now seeks to size the seven openings associated with the modeled flow components using the loop equations. Three loops are particularly relevant to the designer in this case: the first defined by the ventilation flow path a-b-g and the return loop to a; the second c-d-g-c; and the third e-f-g-e. Given the building geometry and topology, climatic conditions, and design criteria presented above, all loop equation parameters may be directly determined and the loop equations formed:

$$\text{Loop } a\text{-}b\text{-}g\text{-}a \qquad \frac{0.32893}{A_a^{\,2}}+\frac{0.32893}{A_b^{\,2}}+\frac{2.9604}{A_g^{\,2}}=\overbrace{4.1418}^{\Delta P_w}+\overbrace{1.0003}^{\Delta P_g} \tag{13a}$$

$$\text{Loop } c\text{-}d\text{-}g\text{-}c \qquad \frac{0.32893}{A_c^{\,2}}+\frac{0.32893}{A_d^{\,2}}+\frac{2.9604}{A_g^{\,2}}=\overbrace{4.1418}^{\Delta P_w}+\overbrace{0.62446}^{\Delta P_g} \tag{13b}$$

$$\text{Loop } e\text{-}f\text{-}g\text{-}e \qquad \frac{0.32893}{A_e^{\,2}}+\frac{0.32893}{A_f^{\,2}}+\frac{2.9604}{A_g^{\,2}}=\overbrace{4.1418}^{\Delta P_w}+\overbrace{0.36427}^{\Delta P_g} \tag{13c}$$

Feasible Design Surfaces & Their Asymptotes

A combination of opening areas satisfying these loop equations will satisfy the design airflow objective for the assumed design conditions or, from a mathematical perspective, the loop equations define *feasible design surfaces*. Furthermore, *feasible design surfaces* are implicitly defined for both no-wind condition (i.e., $\Delta Pw = 0$) and with-wind conditions. For example, for loop *a-b-g-a* we obtain the surfaces shown below:

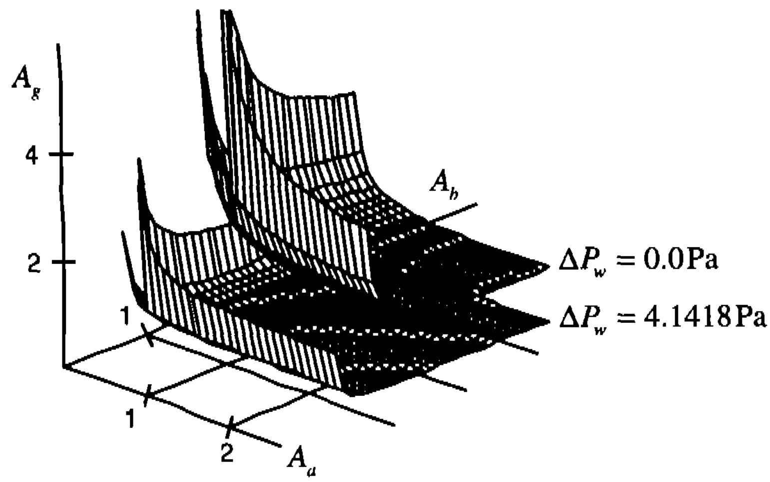

Figure 3 Feasible design surfaces for loop a-b-g-a for no-wind and with-wind conditions

For this example, and in general for *forward-flow loops*, the feasible design surface is hyperbolic in form and bounded by planar asymptotes oriented parallel to the principal axes. These aymptotes define the physically limiting conditions where the resistance of a single flow component governs the flow rate. As such, the asymptotes can prove very useful to the designer – e.g., while there is an unlimited number of feasible openings for this first loop, the opening areas must exceed these limiting values to achieve the design objective of 5 ACH under no-wind conditions. These asymptotes may readily be determined by systematically considering cases where all but one flow link is allowed to approach a negligible resistance (i.e., here an infinite opening area). For example, the A_a asymptote of the without-wind (ΔP_w = 0) loop *a-b-g-a* plotted above is found as:

$$A_a \text{ asymptote} \quad \lim_{\substack{A_b \to \infty \\ A_g \to \infty}} \left(\frac{0.32893}{A_a^{\ 2}} + \frac{0.32893}{A_b^{\ 2}} + \frac{2.9604}{A_g^{\ 2}} \right) = \frac{0.32893}{A_a^2} = \overbrace{1.0003}^{\Delta P_g} \tag{14}$$

$$A_a = 0.573 \ m^2$$

The with-wind surface is also limited by planar asymptotes that establish minimum openings for the second, with-wind, design condition. The *design surfaces* for the other two forward-flow loops will be of similar form and again minimum openings may be evaluated. The results of these straightforward evaluations are tabulated below.

Table 1 Design surface asymptotes for the example problem

Opening	Without-Wind	With-Wind
A_a	0.573 m2	0.25 m2
A_b	0.573	0.25
A_c	0.63	0.26
A_d	0.63	0.26
A_e	0.94	0.27
A_f	0.94	0.27
A_g	1.53 loop a-b-g-a 1.90 loop c-d-g-c 2.82 loop e-f-g-e	0.74 loop a-b-g-a 0.77 loop c-d-g-c 0.81 loop c-d-g-c

The maximum value of these minimum openings establishes the minimum full-opened area of each opening needed to achieve the design airflows. As expected, the without-wind case and, for the stack exhaust, the upper loop, governs. (It is interesting to note that the without-wind design solution opening for the ground floor level presented in the CIBSE manual [1] – 0.579 m^2 – is very close to the limit of 0.573 m^2 tabulated above thus this single opening is critically designed in the CIBSE manual.)

Inclusion of Nontechnical Design Constraints

With these limits and the loop equations in hand, the designer can then proceed to select specific openings to serve both the airflow design objectives and other design constraints he or she faces. For example, the designer may have to select a stack opening from a limited set of off-the-shelf values, let's say a 4.0 m^2 full-opened opening is so selected. In addition, let's assume the designer prefers to use the same window for the windward wall and the room-to-stack openings (i.e., $A_a = A_b = A_{ab}$, $A_c = A_d = A_{cd}$ and $A_e = A_f = A_{ef}$). Recognizing the without-wind case is governing, the designer may then substitute these constraints into the loop equations to finalize design decisions – i.e., for the given constraints we solve:

$$\textit{Loop a-b-g-a} \quad \frac{0.32893}{A_{ab}^{\ 2}} + \frac{0.32893}{A_{ab}^{\ 2}} + \frac{2.9604}{4.0^2} = +\overbrace{1.0003}^{\Delta P_g} \text{ or } A_{ab} = 0.898 \ \text{m}^2 \tag{15a}$$

$$\textit{Loop c-d-g-c} \quad \frac{0.32893}{A_{cd}^{\ 2}} + \frac{0.32893}{A_{cd}^{\ 2}} + \frac{2.9604}{4.0^2} = \overbrace{0.62446}^{\Delta P_g} \text{ or } A_{cd} = 1.224 \ \text{m}^2 \tag{15b}$$

$$\textit{Loop e-f-g-e} \quad \frac{0.32893}{A_{ef}^{\ 2}} + \frac{0.32893}{A_{ef}^{\ 2}} + \frac{2.9604}{4.0^2} = \overbrace{0.36427}^{\Delta P_g} \text{ or } A_{ef} = 1.916 \ \text{m}^2 \tag{15c}$$

This feasible design solution establishes the full-opened opening areas needed for the with-out wind design condition – it is important to stress that other feasible solutions exist as well. The design solution for the with-wind case will, presumably, be achieved by closing down these openings. The with-wind loop equations may then be applied in a similar manner to determined the reduced size of openings to maintain the desired airflow objective.

Conclusion

This paper has outlined the basic tasks of natural ventilation design and presented an approach to the sizing of ventilation components based on the formulation of pressure loop equations that is:

- *exact* – i.e., based on fundamental theory without the need for simplifying assumptions,
- *complete* – i.e., allows the complete and unambiguous consideration of wind and buoyancy effects and enables the design of ventilation flow paths assembled from a variety of different flow components including but not limited to orifice, linear, power law, ducts and fittings, and, possibly, assist fans,
- *inclusive* – i.e., allows nontechnical considerations to be included in the selection of components recognizing ventilation design has no unique solution in general.

A more detailed and complete presentation of this approach will be presented in an AIVC Tech Note "A Practical Guide to Passive Ventilation Air Quality Control in Houses" presently being drafted by the author. This document will present additional worked examples, means to account for infiltration, recently developed self-regulating inlet vents [10-11], and assist fans, and present an approach to deal with the more challenging *nasty* "design" problem.

Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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THEORETICAL AND FIELD STUDY OF AIR CHANGE IN INDUSTRIAL BUILDINGS

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ABSTRACT

The air leakages can have a large impact on heating needs and thermal comfort in industrial buildings. This is sometimes poorly taken into account, both due to the lake of theoretical approach and knowledge of the air tightness. We present the application of the calculation code SIREN95 in this field and its validation against field measurements.
The field study concerns five average industrial buildings, in which different tasks have been carried out : air tightness measurements, using pressurisation method, two series of measurements of air change with a tracer gas method (decay), field measurements during a whole heating season.
Weekly energy balances were calculated using the results of field measurements - and the air changes calculated by SIREN95. They showed a good agreement between heat gains (internal and solar) and heat losses (through the envelope, air change).

KEYWORDS

Air leakage
Air change losses
Stack effect
Pressurisation method
Tracer gas method

INTRODUCTION

The air leakages can have a large impact on heating needs and thermal comfort in industrial buildings.
A study has been undertaken to propose a method of determination of air leakage in this field. It is based on the aeraulic model SIREN95 developed in CSTB. SIREN95, firstly used for residential buildings, has been adapted for industrial buildings in order to calculate:
- the specific ventilation air flow rate,
- the air flow rate due to stack effect and wind effect through air leakages and air inlets,
- the air flow rate through doors when they are open,
- the heat losses corresponding to this air flow rates,

taking into account :
- the characteristics of the specific ventilation system,
- the geometric characteristics of the building,
- the air leakages,
- the doors opening.

This paper presents the application of the calculation code SIREN95 in this field and its validation against field measurements.

METHODS

The validation of SIREN95 concerns five average industrial buildings and is based on weekly energy balances constituted as follows :

- internal and solar gain and energy losses through the envelope of the buildings, as a result of field measurements and thermal modelling of the building,
- the air change rates calculated by SIREN95, which entering datas as a result of field measurements too.

Consequently, different tasks have been carried out in the studied buildings :

• field measurements during a whole heating season providing internal gains, solar gains, heat losses through the envelope, internal temperature at several levels,

• air tightness measurements, using pressurisation method, to assess the air leakages used as an entry in SIREN95. The air leakage is expressed as an equivalent hole in [m²],

• two series of measurements of air change rate with a tracer gas method (decay), to know how the leakage are distributed on the envelope ;

- the first one without wind and with high difference between indoor and outdoor temperatures,
- the second one with wind and low temperature difference

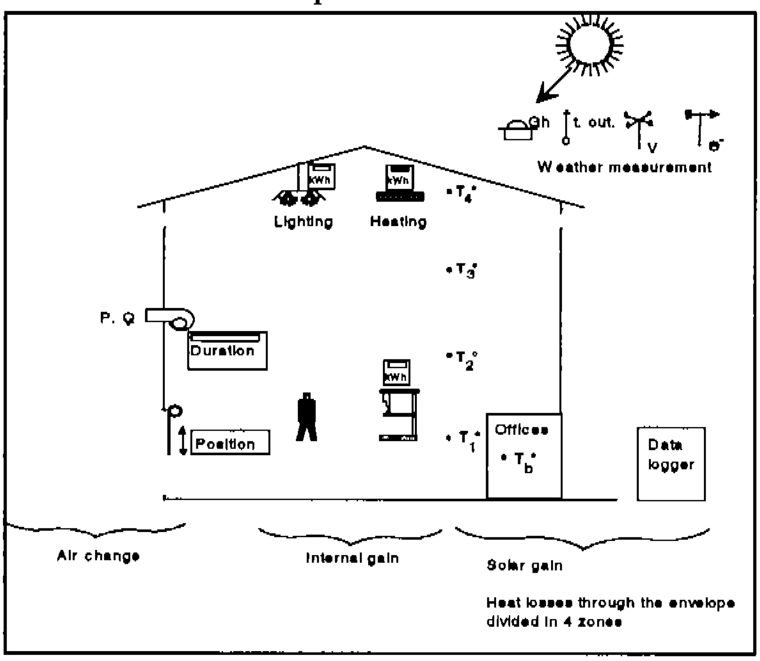

Figure 1 : field measurements

The repartition is made comparing the measured air change rate to the air change rate obtained with SIREN95

The aim of field measurements is to obtain the elements of energy balances, which are as follow (see figure 1) :

internal gain + solar gain = air change losses + heat losses through the envelope

- internal gains : energy of lighting, heating and process are measured,
- occupancy : averaged,

- solar gain : solar irradiation in a horizontal surface is measured and used to calculate energy through the envelope,
- heat losses through the envelope : calculated from the characteristics of envelope with the difference between outdoor an indoor temperatures,
- air change losses. Specific air change rate is calculated knowing the working time of fan, air flow rates being measured once.

The air change rate due to doors is calculated with the position of doors (open, half open, closed), known with a specific device.

Air leakages are measured. Air change rate is calculated with SIREN95 from specific air change rate, position of doors and air leakage.

The following tables and figures show the main characteristics of the buildings and examples of field results.

	1	2	3	4	5
Construction	1992	1990	1981	1988	1990
Area * m^2	695	811	671	1347	558
Volume m^3	2967	4234	3086	6957	2500
H $W/m^3.K$	0,42	0,40	0,55	0,23	0,36
$H_{réf}$ $W/m^3.K$	0,28	0,34	0,41	0,23	0,32

Table 1 : main characteristics of buildings

The constructions have the same metallic structure with incorporated insulation (mineral wool) for walls and insulated roof. Their shape is simple (parallelepiped) to make easier the field measurements and their comparison with the results of modelling.

The equipment of air tightness measurement limits their volume at 8000 m^3.

		10 Pa		50 Pa		n	
		I	D	I	D	I	D
1	m^3/h	12393	11287	30227	32378	0.55	0.65
	Ach	4,18	3.80	10,19	10.91		
3	m^3/h	9162	8984	27637	26860	0.68	0.68
	Ach	2.97	2.91	8.96	8.70		
4	m^3/h	6558	6656	23705	23837	0.79	0.79
	Ach	0,94	0,96	3,41	3,43		
5	m^3/h	12075	12435	44930	46723	0.81	0.82
	Ach	4,83	4,97	17.97	18.69		

Table 2 : results of air tightness measurements

I : measurements made during an increasing pressure cycle

D : measurements made during a decreasing pressure cycle

These results are presented for two pressure differences :
- 10 Pa : the air flow rate in m^3/h under 10 Pa is quite expressed by the same number as the hole area in cm²,
- 50pa is the reference in Europe.

It is to notice that the measurement has not been carried out in the building number two because of a too large air leakage, the pressure difference generated was too small to be significant.

The exponent for building number one in not the same during increasing and decreasing cycle : it is supposed the building could have been bent under pressure.

		Measurement location		
		Point 1	Point 2	Point 3
1	Ach	0,32	0,30	0,32
	CC*	0,996	0,996	0,997
2	Ach	without heating 0,61 heating 1,65	without heating 0,74 heating 1,51	without heating 0,58
	CC*	without heating 0,994 heating 0,997	without heating 0,987 heating 0,978	without heating 0,994
3	Ach	0,63	0,58	0,68
	CC*	0,998	0,998	0,998
4	Ach)	0,22	0,20	-
	CC*	0,998	0,997	-
5	Ach	0,67	0,68 air extraction 1,20	-
	CC*	0,995	0,995 air extraction 0,994	-

*CC : correlation coefficient

Table 3 : results of measurement of air change rate with tracer gas without wind and with high difference between indoor and outdoor temperatures

In each building the measurements have been carried out in several locations (point 1 to 3) at different highs, doors and windows closed.
The results show that :
- the air change rate doesn't vary with the high of measurement, then table 3 presents only the average of air change rate at different points.
- the differences noticed between the different points are not significant. Inside and outside conditions (temperature and speed) being constant, the measurements have not been disturbed. The air change rate due to stack effect has an effect in the whole of the building.

During dispressurisation cycles, investigations with an IR camera have been carried out to show the air tightness faults.

RESULTS

Energy balances, made during a heating season for every building, are shown in figure 2 : internal gain and solar gain are compared to heat losses through the envelope and to air change losses. Energy is expressed in kWh/m².week.
As we can see, gains and losses are quite the same especially in buildings 1, 2 and 4.

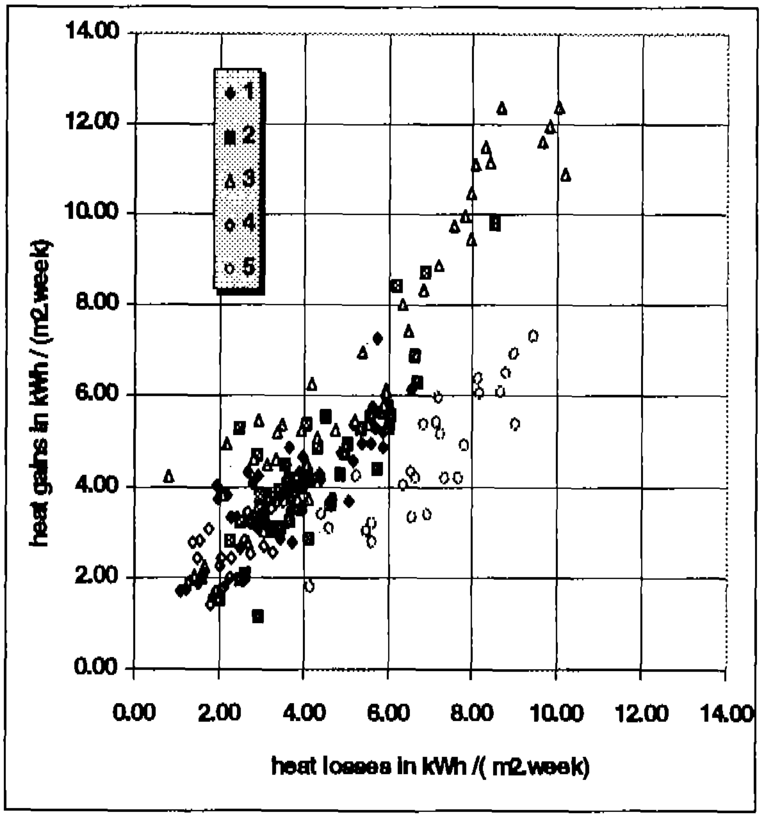

Figure 2 : weekly energy balances

Figure 3 shows weekly mean energy balances during the whole heating season and during a cold period.

During the heating season :
- solar gains are an important part in term of heating, however its calculation remains rough.
- air change losses are different, depending on the building (0.8 to 2.7 kWh/m².week), but are ever a significant part of global losses (30 to 50 %). Air leakages flow rates are larger than specific air change rate and door opening air change rate.

During a cold period :
- solar gains are obviously smaller and become negligible in relative term,
- air change losses vary from 0.8 to 3.7 kWh/m².week. Air leakage losses are important but specific air change rate is well controlled.

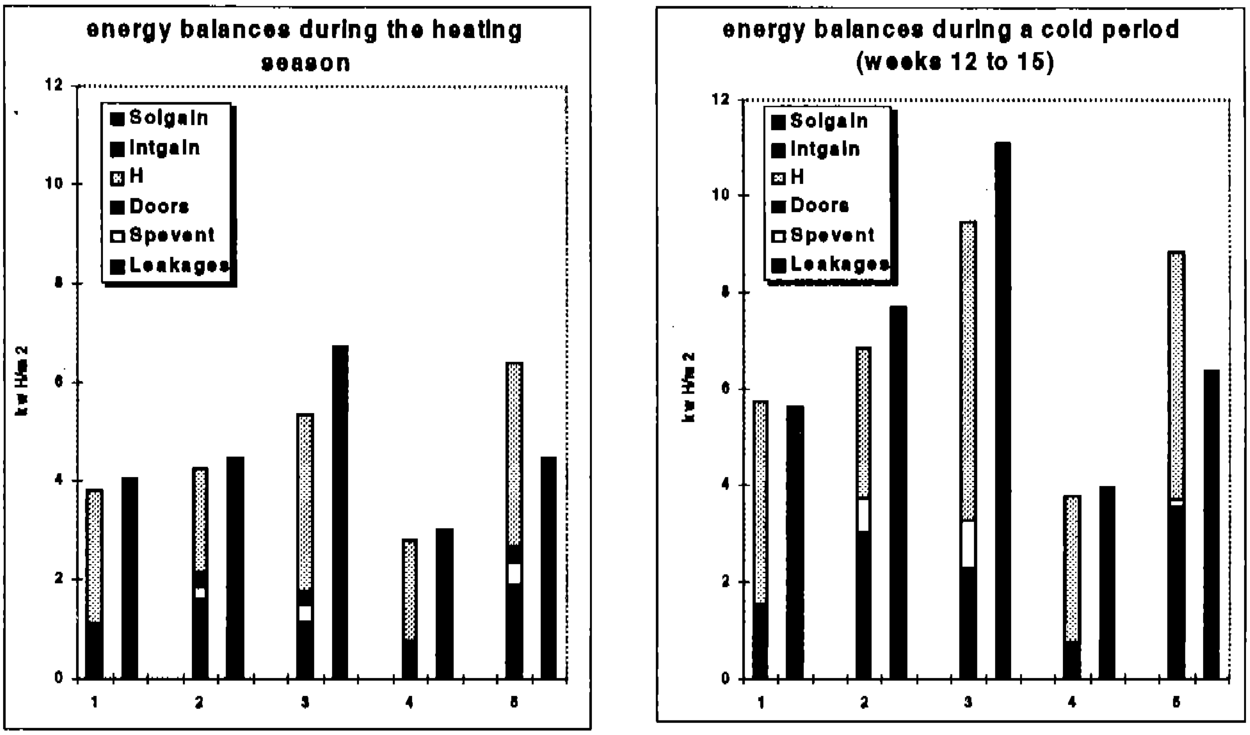

Figure 3

SIREN95 has been used to calculate the daily mean air change rate according outdoor temperatures and wind speed, without specific air change rate. See figure 4 for an example of results in building 1.

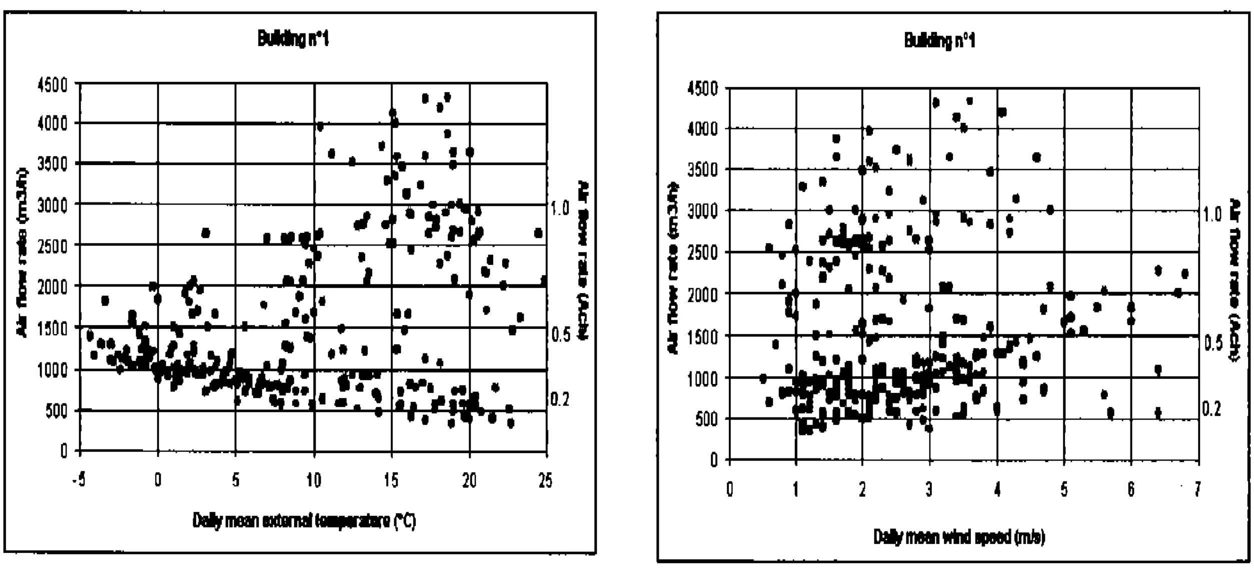

Figure 4 : daily mean air change rate according to outdoor temperature and wind speed

• according outdoor temperatures, results are divided in two parts :
- in the first one, air change rate decrease when outdoor temperature increase. It represents the stack effect air change rate through air leakages.
- in the second one, air change rate increase with outdoor temperature. It represents the behaviour, mainly the door opening.
- intermediate values can be considered due to the wind.
• according the wind speed, results are divided in two parts too :
- in the first one, air change rate increase with wind speed.
- in the second one, air change rate, due to door opening, is not related to wind speed.

It can be noticed the air change rate due to behaviour is more important than air change rate due to air leakage, and it is independent of air tightness and represents an extra air change rate of 1 to 1.5 ach.

Air change rate due to air leakage (cold weather or high wind speed) is shown table 4;

Building	(ach)
1	0.5
2	2
3	1.5
4	0.2
5	0.5

Table 4 : air change rate due to air leakage

SENSISTIVITY ANALYSIS

In order to check the impact of the different parameter, we performed a sensitivity analysis. We choose a simple building (height : 5 m , dimensions 40 m x 20 m) with an airtightness of 0.5 a.c./h under 1 Pa.

We applied 4 weather conditions : 2 outdoor temperature (0°C and 15 °C), 2 wind speeds: 0 m/s and 5 m/s.

2 indoor conditions were defined : 20 °C without stratification , 20 °C on ground and 25°C near the ceiling.

The air leakages were split in the lower and upper parts of the building with different ratios :
1/2 down ; 1/2 up
1/3 down ; 2/3 up
2/3 down ; 1/3 up

Results are shown in figure 5 and 6

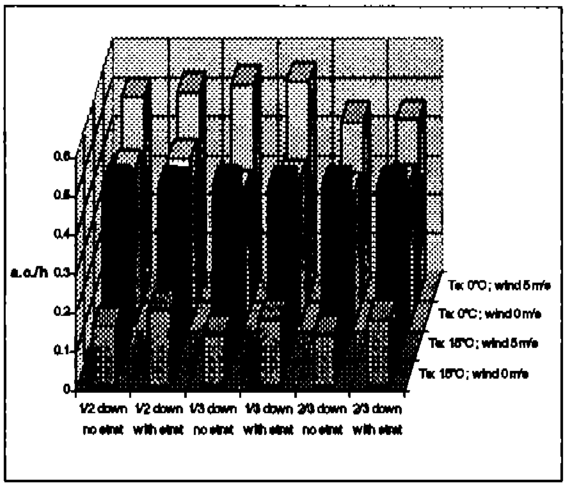

figure 5 : air change for a typical building according to outdoor conditions, position of air leakages and indoor stratification

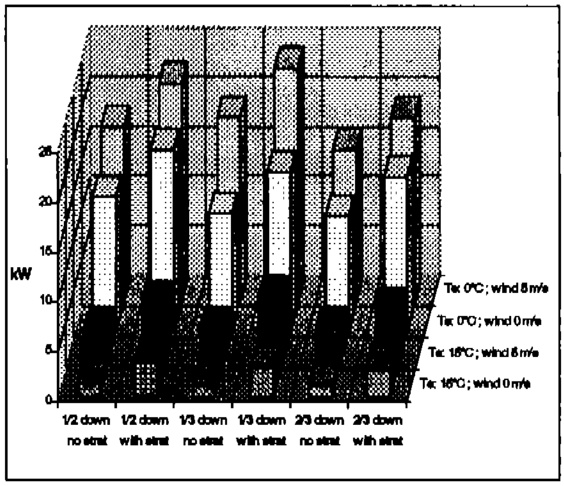

figure 6 : heat losses due to exfiltration for a typical building according to outdoor conditions, position of air leakages and indoor stratification.

When stack effect is dominant, the higher flow rates are obtained with equally split leakages. When wind effect is dominant, there is no difference due to the present assumptions that the air leakages remain equally split between the windward and leeward facades. When both effects are at the same level, a calculation has to be made, as it is not easy to predict in a simple way the flow direction for each leakage.

DISCUSSION

Further developments will be undertaken in order to obtain more applied tools.

- a design and dimensioning tool, based on a simplified predicting tool of air change rate during the heating season using a simplified set of input data and based on a data base on building techniques in term of air leakage.
- a procedure of control of air change rate due to the air leakage with a tracer gas method.

ACKNOWLEDGEMENTS

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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EXPERIMENTAL DETERMINATION OF THE PERFORMANCE OF AIR FILTERS FOR GENERAL VENTILATION

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Synopsis

Filters used for general ventilation are mass produced and tested by type at rated airflow rate in order to determine the evolution of the pressure drop and the efficiencies during an artificial and shortened clogging process. For filters of better quality it is necessary to evaluate the efficiency concerning fine dust: the traditional atmospheric dust spot efficiency method is now being substituted with an innovative method which allows one to determine the fractional efficiency versus the particle diameter within a 0.2÷3 μm range. Some examples of comparative assessment of commercial filters are here provided, taking the requirements of HVAC systems into account.

1. Introduction

Air filters used inside HVAC and for gas turbine systems are usually made up of layers of fibres [1]. These fibres are artificial, made of glass or organic material, and should have a diameter as small as possible, since the efficiency increases as their diameter decreases[2]; with fibre glass it is possible to reach diameters of about 1 μm, while, with synthetic materials, the diameters at the moment are usually larger and sometimes the efficiency is improved using adhesive substances or an electrostatic charge.

The fibres are arranged in such a way as to form a layer of variable thickness, from 1 to 10 mm, which, when it is fitted to form a flat filtering surface, provides the most suitable filter for research purposes as its geometry is uniform and simple. The data obtained in this way characterise the filter media and are used when material manufacturers and filter manufacturers are in touch; they are however usually more interesting for researchers than for the end users.

Commercial filters are seldom flat as the air velocity inside the ducts is in the order of magnitude of meters per second, while the most suitable velocity for filter media is an order of magnitude lower. For this reason, filters have a frame so that they can be placed in the ducts and also a net filtering surface which is much larger than that of the inlet (which is called frontal surface). Filters therefore do not usually have a flat shape and this fact, together with the presence of the frame, produces an air motion that is not one-dimensional. This does not help one to understand the phenomena which occur inside filters, even though it is commonly supposed that the air flow occurs in a steady state. On the other hand, a flat geometry would cause a too high resistance to the motion and a too short life of the filter.

Commercial products set their performance in order to cause a pressure drop, when the filter is clean, in the 10÷100 Pa range, and, at the moment of replacement (after 6÷12 months of life), can be in the 250÷450 Pa range. The value of the pressure drop is about the same as the other components in the air handling unit and its variation is compatible with the characteristic of the fan, in order not to cause significant airflow rate decreases or a too high energy consumption.

From the user's point of view the fractional efficiency is related to the class and, therefore, to the commercial product cost. A very small part of the cost is due to laboratory test costs which ensure the quality of the industrial products and their classification. These are type tests, that is to say, they are made on a sample which is considered to be representative of the whole mass-production.

The project engineer in charge of choosing a filter should correlate the specifications to the application he is dealing with. In theory, he should know the dimensions and concentrations of the solid particles and, at the same time, how they are filtered by the products available on the market. For this reason, it is becoming necessary to qualify filters, also using fractional efficiency, that is, efficiency in number per classes of dimensions in which it is possible to subdivide the atmospheric dust.

The filtration theory [2] does not allow one to correlate measurable characteristics of a commercial filter to its performance in function of a quantity of particles with known dimensions. The theory provides the fractional efficiency value of a single fibre for each filtration mechanism and the combined effect of two mechanisms for one fibre. The effect of many fibres can be taken into consideration in order to calculate the resistance to air motion in the case of a clean filter, but not in order to foresee the fractional efficiency value, especially when the fibres are of variable sizes and arrangements and when they are already mixed with dust.

Fractional efficiency should therefore be determined by experimental means.

At the end of their technical life, filters become waste material that can be compared to urban solid waste, and therefore have to be disposed of. They are usually sent to a waste disposal yard or burnt in an incinerator: the disposal of air filters is a problem particularly in the case of gas turbine systems because of the volumes involved, thus, the possibility of incineration is beginning to be taken into consideration and, at the same time, the behaviour of commercial products to fire. In theory, it would be better to have non-flammable filters in order to prevent fires and, at the same time, easily incinerable filters in order to dispose of them without problems.

2. Experimental results

The filters here considered were chosen in order to be representative of the availability on the Italian market and are partially described in the table where they are marked by a number: even though the manufacturer's name and the designation of the product have been indicated, this would still not be enough to clearly identify the filter.

From a scientific point of view, the information that would be missing is: the diameter and the arrangement of the fibres, the characteristics of the binder (if present) and the apparent layer density (linked to the packing fraction). These data are necessary, even though they are not sufficient, for many cases where an effort has been made to apply the theory, and the end user and the HVAC system designer are not usually interested in this information. However, it is necessary to remember that, judging from technical and commercial brochures, the uniformity of mass-produced filter characteristics is not known and for this reason it is not possible to predict which performance variations are possible within seemingly equal mass-products.

As is known [3], filters are tested while trying to reproduce conditions that are representative of those of real life inside the laboratory (fig.1). The clogging process is carried out using synthetic dust which simulates natural dust (inorganic fraction and organic fraction, subdivided into carbon black particles and cotton linters). The concentration used during the laboratory test (at most 70 mg/m^3) is much higher than that present in outdoor air (0,005÷3 mg/m^3) in order to make the clogging process faster. Arrestance measurements are performed during this process, comparing the mass held by the filter to the injected one. The arrestance (A*) measurement is not very meaningful for F class filters (fine dust) according to the European classification and used in widespread applications, as it produces arrestance values that are close to one, in other words, the injected dust is completely held by the filter at least as can be observed on the basis of the mass measurements, and which can be seen in the

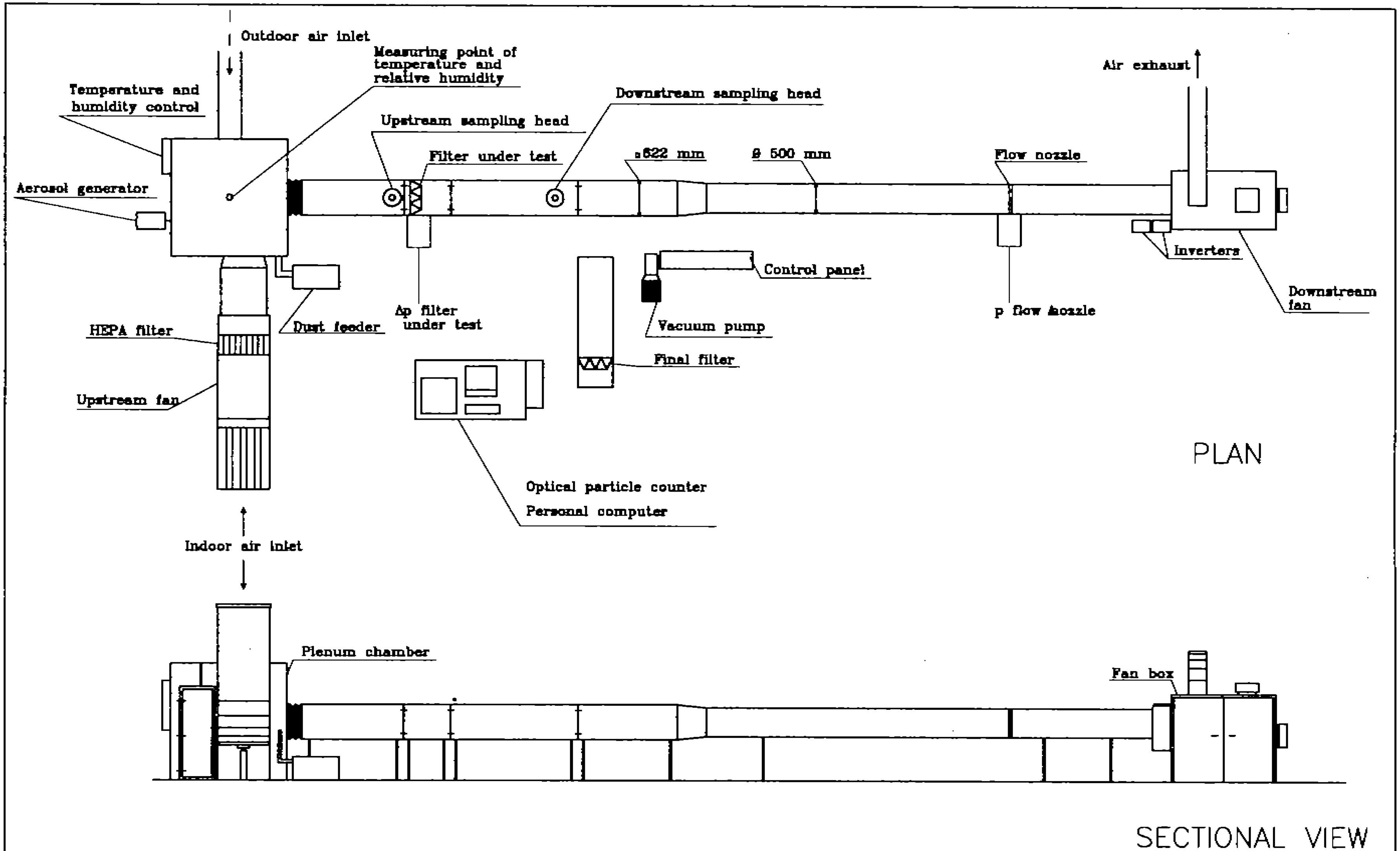


Figure 1 Test rig and set of instruments

example of fig.2a. This fact does not imply that F class filters offer uniform performances, as the mass measurements are not able to evaluate the ability to stop and hold the particles of a smaller size. It should be noted that a particle of 0.1 μm diameter has a mass equal of one millionth of a 10 μm diameter, thus smaller particles should be detected using other methods.

Up till now a method based on atmospheric dust dirtying capacity has been used and, even though this varies, depending on the place and on the time of the day, the dirtying capacity ratio between upstream and downstream of the filter being tested does not change. The so-called dust spot efficiency method consists of sampling the outdoor upstream and downstream air of the filter being tested and evaluating the opacity of the stain left on a small filter of white paper which is able to hold all the particles that are suspended in atmospheric air. It is possible to obtain a dust spot efficiency through a light transmission factor, which closely depends on the filter being tested and, at the same time, is greatly independent of the place. A dust spot efficiency value is thus obtained that is conventionally defined which seems to correspond to a superficial global efficiency, that is, based on the equivalent area of the particle surface.

The dust spot efficiency method is rather labour demanding, requires a great deal of time, above all for high efficiency filters and for places with low atmospheric dust concentrations: in practice, it determines the whole test duration of a filter that can last as long as a week. At the same time it provides a group of progressive values and an average dust spot efficiency value (see for example fig. 2a). However, this does not provide information on the size of the dust particle and on the fractional efficiency corresponding to the size classes of greater interest for F class filters, that is, between 0.1 and 5 μm.

This is the reason why the presently used test method has been updated [4] and the dust spot efficiency test has been replaced while the artificial clogging process with synthetic dust has been left unchanged. Instead of atmospheric dust, completely filtered air and an artificial aerosol, which contains a large number of particles in the 0.2÷3 μm size range have been used. By sampling the upstream and downstream air of the filter being tested, it is possible to measure the concentration, in number of particles, for the various size classes: a particle counter is required, that is, a much more expensive instrument than those used for the dust spot efficiency test. However, the measurement is performed in a few minutes and a group of data is obtained such as that shown in fig.2b.

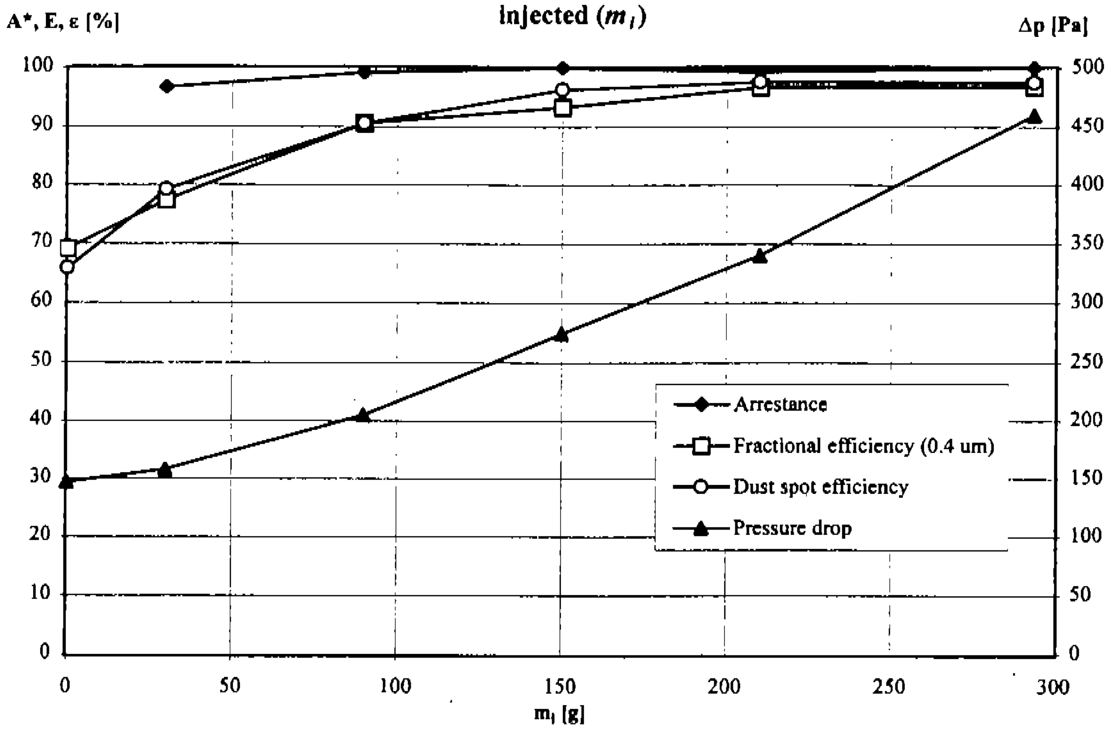
Systematic tests on commercial filters have been performed using the test rig available at the Department of Energetica at the Politecnico di Torino, adding this new method which has negligible impact on the filter being tested, so that, in fact, it has been simply been used alternating with the traditional method.

The filter media used are fibre glass and synthetic fibres. The different shape of the filters includes "pockets" of different kinds. "Pockets" is the generic name of the surfaces stretched according to the way in which the filtering material is arranged: they are called self-supporting when they are dihedrals formed by mini-pleated paper in fibre glass with a binder. The "pockets" are called non-supported when they are formed by folding and sewing a layer of fibrous material; for one of the tests here presented the initial flat layer has been folded in the laboratory and arranged on a W-form frame (W filter) which was designed and constructed for testing generic filter media (fig.3).

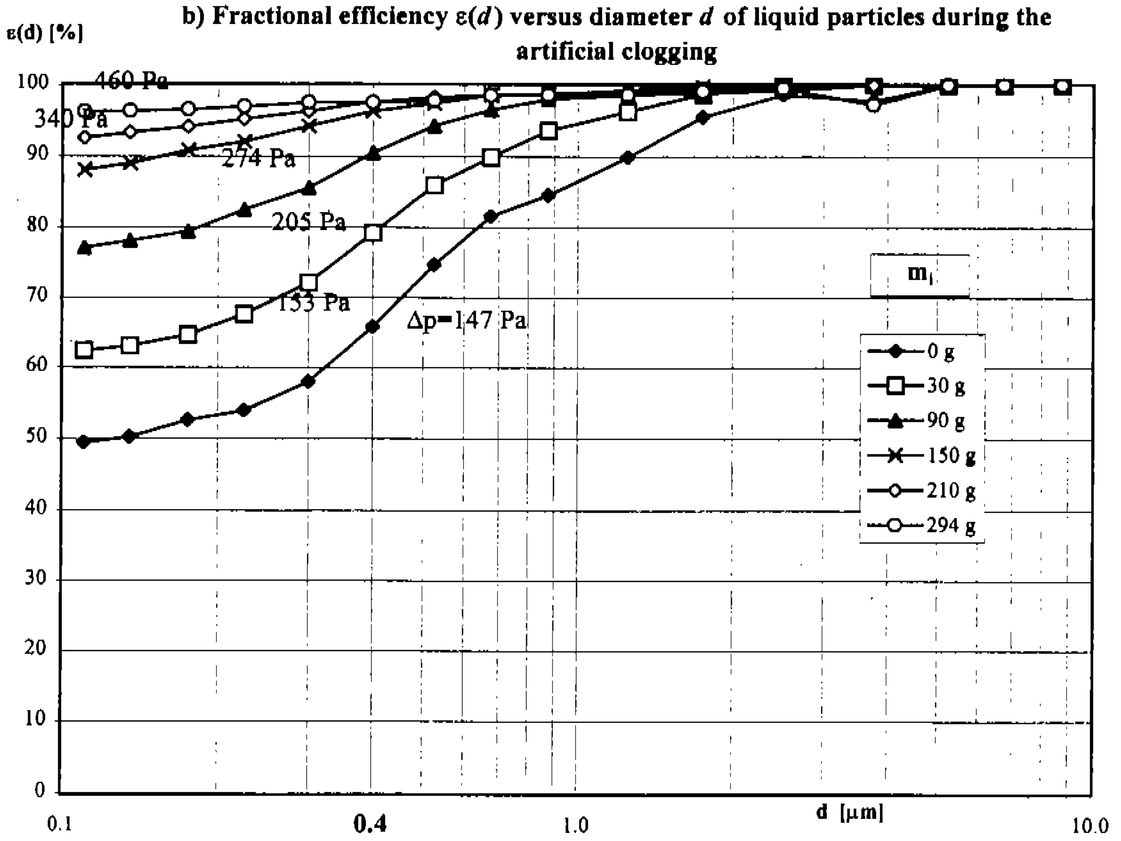
The area of the filtering surface (A) is equal to 1 m^2, in the case of the W filter, and it is much larger in the other cases; it does not have a close correlation to the filter mass (m_f), because of the presence of the frame.

Figure 2 Performance of filter n. 1 (see table)

a) Arrestance (A^*), dust spot efficiency (E), fractional efficiency (ε) for d=0.4 μm, pressure drop (Δp) versus the mass of the synthetic dust injected (m_i)



b) Fractional efficiency $\varepsilon(d)$ versus diameter d of liquid particles during the artificial clogging



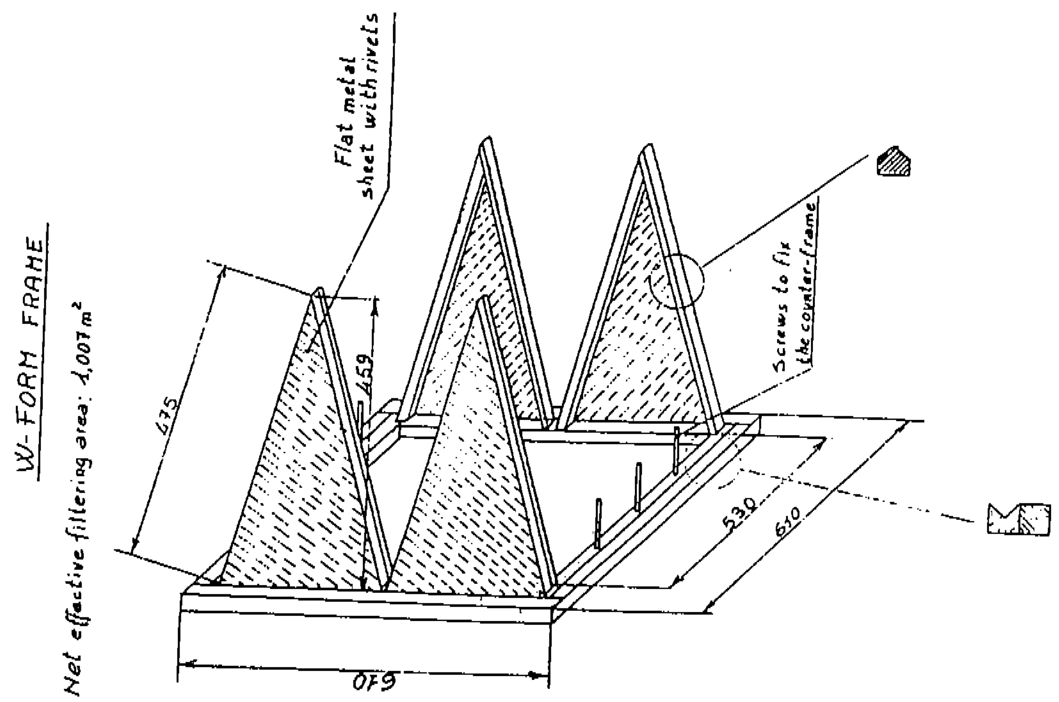


Figure 3 - W-form frame for testing filter media

The rated airflow rate of the filter ($\dot{V}_n$) is that which is indicated by the manufacturer and reflects its commercial policy: a high airflow rate corresponds to a greater filter utilisation and to a shorter life in case of the same final pressure drop or to a higher energy consumption for ventilation for the same life of the product.

The crossing air velocity (w) of the filtering material is of the order of decimeter per second while the face velocity in the filter (calculated in the frontal section with area A'=0.36 m^2) is of an order of magnitude that is greater for commercial filters; the W filter is an exception, as far as the surface and airflow rate is concerned, while it has a realistic value for w.

The initial pressure drop (Δp_i), that is, with a clean filter and at rated airflow rate, indicates the flow resistance that the air faces while crossing the filter; it is worthwhile to remember that the resistance to the flow motion is partly due to the material and partly due to the geometry of the filter, and that this resistance expresses an overall effect which does not necessarily correspond to the local situation.

The geometry being three-dimensional, it is possible to have non-uniformity of the air distribution, even when the filter is clean; this effect would of course increase during the clogging process.

The initial dust spot efficiency value (E_i) is the most important data for filter classification. According to the current European standard criterion, if E_i>20% the filter is an F class filter, that is, suitable for fine dust; in the other case it is classified as G, that is, suitable for coarse dust and therefore intended for use in less demanding applications.

The initial efficiency in number (ε_i) is obtained with the new method (see fig.2b, lower curve); the value entered in the table refers to the single size class around the diameter of the particle d = 0.4 μm.

The mass (m_a) of synthetic dust collected over the whole test, that is, when the final pressure drop was reached at the rated airflow rate, expresses the filter dust holding capacity. Also in this case, the value should be considered as an average value:

- in space, because it is not clear whether the filtering surface is clogged in a uniform way;
- in time, because the part of the injected dust which is held results from the combined effect of the dust that immediately passes through the filter and that of the dust which is first stopped and then released.

The final dust spot efficiency (E_f) is the value which is obtained at the final stage of the whole test when the final pressure drop is reached at the rated airflow rate (fig. 2a).

The final efficiency in number (ε_f) is entered only for a size class around d = 0.4 μm and is obtained with the new method. In fig. 2b, the upper curve describes the fractional efficiency trend at the end of the filter clogging process carried out with synthetic dust.

It is necessary to point out that the experimental data here presented were obtained by testing one sample for each filter; if the tests were performed again on filters with the same commercial name, the differences, if any, could be attributed to:

- the laboratory, as far as repeatability is concerned;
- the network of laboratories, as far as reproducibility is concerned;
- the manufacturer, due to the fact the sample may not be representative of his production.

The testing laboratories defend themselves by statistically treating data in order to improve the repeatability of the tests, by inter-laboratory tests in order to assess reproducibility and by formal mechanisms that limit their responsibility to the received product. However, this behaviour does not protect the end user who is therefore only protected by putting the product quality systems into effect.

3. Analysis of experimental results

The totality of the air filters here considered represents typical commercial products that are presently available in Italy: it is therefore predictable that the obtained experimental results could be representative of the rated airflow rate, pressure drop and current values of the efficiencies current values. The element of originality is represented by adding fractional efficiency values within the 0.2÷3 μm size range: the values entered in the table show the fractional efficiency in number for the 0.35÷0.45 μm size range. The question concerning the possible use that may be made of these results is still open: in order to try to clarify the answer to this question, it has been subdivided into three cases: the user's, the manufacturer's and the researcher's point of view.

The end user of the filter would like the chosen product to be able to:

- avoid dirtying the downstream surfaces and/or avoid the sensation of dusty air;
- have a long life without modifying the airflow rate;
- be non-flammable;
- be easily disposed of.

As the filter is a component that can be disposed of it is considered in a different way from the other durable elements of the HVAC systems. The user keeps the pressure drop value under control, or at least the final value that indicates the need of the replacement and which can be found in the manufacturer's technical brochure in correlation to the rated airflow rate.

When the filter is not replaced there are usually no clear negative signs. In fact, air filters can work under overload conditions and may last for a long time, but in an unpredictable way.

A second problem is connected to the growth of micro-organisms which, once again, do not lead to clear signs in the short term: the most detectable element for those close to the filter is its odour. From experience gained over the years it has however been found advisable to change the filter after one year. However when this is not done it should also be admitted that there are no immediate consequences.

As far as efficiency is concerned, the situation is different, as it is not actually possible to measure and, even if it were known, its interpretation would be difficult: there is no intuitive idea of the effects of the particles not held by the filter. The soiling of the surfaces depends on the square of the particle diameter; often surfaces are not visible and some time can pass between a noticeable soiling and perceivable consequences. The smallest particles have a negligible mass and surface so they can only be guessed at.

A clear advantage of the new test method for air filters is that it provides broader information: it is possible to know how the efficiency in number varies in function of the particle size, within the 0.2÷3 μm range. These particles are not visible, but are those that prevail (in number) in the atmospheric dust: the trend of the fractional efficiency curve versus particle diameter may be useful for the end user mainly if used in a comparative way, that is, starting from a reference situation. For example, it is known that when the air used inside the varnishing cabins is not filtered properly the effects can be seen on the varnished surface, and the users' experience would advise them of variations in the filter efficiency for the application they are interested in. The information constituted by the fractional efficiency is an input data that is useful for the designer of the ventilation systems only at present while in the future it will be necessary.

From the filter manufacturer's point of view, it is important to produce reliable products which can ensure a good margin of profit. The mechanical resistance of a filter is very important, so that it can reach the final pressure drop in a good condition, and thus there will be the possibility of characterising it in such a way as to evaluate its characteristics in an advantageous way. For this reason, the tests which allow one to compare competitors products are considered interesting, while the burden of being significant of real life conditions is left to the test method. Filter test methods in the laboratory are not representative of every real life condition, nevertheless it is necessary to resist the temptation of pointing out this fact in order to justify a commercial product which has lower performance than expected, and even forgetting it in other cases. It is important for the manufacturers to be able to use the tests that have already been carried out using the old method and for this reason it is useful to compare dust spot efficiency values with those of fractional efficiency. The data obtained during these tests and shown in the table confirm the good agreement, for F class filters, between the initial dust spot efficiency values and efficiency in number relative to 0.4 μm size values. The agreement obtained when the filter is clean also seems to remain when the filter becomes dirty, as can be seen from the values of ε_f.

For filter manufacturers it is important to demonstrate that their products can be incinerated, but not flammable: these two characteristics seem to be antithetical and deserve more attention than which is currently given. There already exists a test method for flammability [5] that requires the filter to work in the test conditions, and to be put in touch with a gas flame and then to measure the opacity of the combustion products: it is interesting to note that the possible combustion occurs in a cold environment because of the airflow through the filter. This test is performed using a clean filter and it is obvious that those who are not satisfied with the results, claim that it is not representative as the filtered material is not taken into account. The mass of this material is in fact not negligible compared to the mass of the clean filter (see table) and its contribution to flammability cannot be excluded.

Table - Filters for general ventilation characteristics

	Material	**Shape**	**A**	$\mathbf{m_f}$	V_n	**w**	Δp_i	β_i
	Fibre	P.= Pockets	[m²]	[kg]	[m³/h]	[m/s]	[Pa]	
1	glass	Rigid P.	19	5.35	4250	0.06	145	22
2	glass	Rigid P.	19	5.66	4250	0.06	215	33
3	polypropylene	Non-rigid P.	8	2.09	3400	0.12	50	13
4	polyester	W-form	1	0.66	900	0.25	30	94
5	glass	Rigid P.	19	5.08	3400	0.05	95	23
6	glass	Non-rigid P.	8	1.92	3400	0.12	85	20
	$\mathbf{E_i}$	ε_i**(04)**	$\mathbf{m_a}$	$\mathbf{E_f}$	ε_f**(0.4)**	β_f	$\mathbf{m_a/A}$	$(\beta_f-\beta_i)/(m_a/A)$
	[%]	[%]	[g]	[%]	[%]		[g/m²]	[m²/g]
1	69	66	292	90	91	70	15	3
2	89	92	192	95	97	70	10	4
3	32	6	399	46	32	110	51	2
4	15	6	280	43	31	870	280	3
5	74	72	511	-	96	110	27	3
6	66	65	633	88	90	110	81	1

The possibility of incinerating filters is a problem that is considered in countries which usually incinerate solid waste and in the case of plants that use a great deal of filters, such as gas turbine groups. In this last case, the possibility of complete burning (excluding metallic frames, for example) and the presence of substances that can generate emissions into the smoke is important: for example, plastic materials with chlorine (the manufacturer's responsibility) or heavy metallic substances in the collected dust (the user's responsibility). Filters are not incinerated alone but together with solid urban waste in a high temperature environment, which is very different from that which occurs during flammability tests. There are no common methods available at the moment to test the possibility of incinerating air filters.

From the researcher's point of view, it is important to express filter characteristics in such a way that they can be traced back to general laws and therefore to allow one to investigate the phenomena in order to give some indications on the development of knowledge and, if possible, on commercial products [6]. From this point of view, filter resistance to air motion is naturally expressed by a concentrated resistance coefficient that pertains only to the filter and which does not depend on the airflow rate, as the pressure drop does. This coefficient is indicated by β in the table and has been calculated either in the initial conditions (β_i) or in the final ones (β_f for Δp_f=450 Pa for filters n. 1,2,3,5,6 and Δp_f=250 Pa for filter n. 4), by means of the equation $\beta=2\Delta p/\rho w'^2$ where ρ is the air density and $w'=V_n/A'$ is the frontal velocity. Collected dust is responsible for the increase of the coefficient, therefore it is natural to establish a relationship with the mass divided by the area of the filtering surface (m_a/A). If this ratio were the same for different filters, it would be a particularly meaningful parameter. However, if one looks at the table, it does not seem to be so and this is generally what has happened to all attempts to generalise air filter performance. The reason for this is that filters have only been characterised by overall experimental data and they are not related, in a precise way, to the characteristics of the filtering media and the commercial product. On the basis of the filtration theory, it is possible to criticise, above all, the choice of the previously indicated β coefficient because filter resistance to motion depends on the contribution of the material (generally, proportional to the crossing air velocity w) and on the contribution of the shape, which is not flat (generally, proportional to the square of the frontal velocity w'). For example, in the case of filter 1 in the table, the pressure drop due to the material is less than half of the indicated value for the filter. Remarks of this kind make very difficult to attempt any kind of simple correlation: this fact does not prevent manufacturers from improving their products as the functionality of the systems does not require, as in many other cases, their satisfactory description from the scientific point of view.

Nowadays, the point of equilibrium between different requirements is represented by research [7] that deals with the problem of the technical life of filters and which tries to limit empirical relationships as much as possible. The overall impression is that modern means of calculation are sufficient to deal with the problem, from the quantitative point of view, even when taking the efficiency variability with the particle size into consideration. Instead, "qualitative" relationships, such as the efficiency of a group of fibres, for example, are not applicable to commercial products, even though they are very refined from the scientific point of view. There probably exists situations in which scientific interest is greater in the aerosol field and not so high in the ventilation environment as it is hard to obtain original and conclusive results, while in the industrial field these products have such small margins of profit that investments are justified only in the case of immediate commercial results.

4.Conclusions

Filtration efficiency for thin particles can be measured using the new method here described which provides fractional efficiency in number and, for this reason, it is possible to obtain much more information, in a shorter time, compared to the dust spot efficiency method which is still used. On the basis of experimental data obtained on products that are representative of the Italian market, the agreement between the old and new method is satisfactory for fine filters. With this further information, the possibility of joining HVAC system performance with the measurable characteristics of filters is closer.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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AIRTIGHTNESS PERFORMANCES IN NEW BELGIAN DWELLINGS

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SYNOPSIS

A systematic analysis of recently constructed dwellings in the Flemish Region has been undertaken within the SENVIVV-project (1995-1998) [1]. In total 200 dwellings have been examined in detail. The study involved various aspects: energy related building data (thermal insulation level, net heating demand, installed heating power, etc.), indoor climate (temperature levels in winter and summer), building airtightness, ventilation, appreciation of the occupants, etc. This paper focuses on the results of the airtightness measurements that were undertaken in 51 of the 200 investigated dwellings. These measurements revealed that the global airtightness depends strongly on the building type: on average, terraced houses are more airtight than detached houses, but less airtight than apartments. There is a wide spread on the results, especially for detached houses. The worst results are mainly caused by a poor finishing, due to the fact that a lot of owners do a part of the finishing work themselves.

LIST OF SYMBOLS

n_{50} - air change rate for a pressure difference of 50 Pa (h^{-1})
Q_{50} - leakage air flow rate for a pressure difference of 50 Pa (m^3/h)
V - volume of dwelling, room (m^3)

1. INTRODUCTION

Each year about 35 000 new dwellings are constructed in the Flemish region (i.e. the northern half of Belgium). During the nineties a standard related to ventilation and building regulations related to thermal insulation came into force. As little was known about the building practice and the compliance with the new regulations, a thorough study [1] was set up to examine the energetic performances of new dwellings. From 1995 to 1998, 200 representatively selected houses and multifamily buildings were investigated in detail. This paper discusses the building airtightness.

2. BELGIAN VENTILATION STANDARD

The Belgian standard NBN D50-001 (March 1992) [2] describes the requirements for ventilation in dwellings. In the Flemish region this standard is not compulsory (except for social housing), but every standard has to be seen as a rule of good practice, and as a consequence the performances have to be comparable with the requirements of the standard.
The philosophy of the standard is that a good ventilation consists of different aspects, represented in Figure 1.

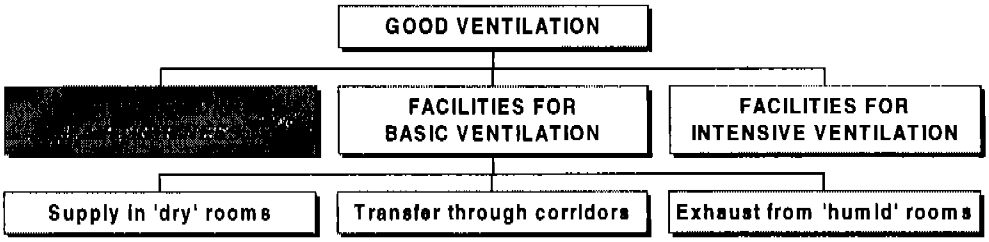

Figure 1: Elements for a good ventilation strategy, according to the Belgian ventilation standard

With respect to building airtightness, the Belgian ventilation standard only gives some guidelines for specific cases, no requirements. These guidelines are the following:

⇨ In the case of balanced mechanical ventilation: $n_{50} < 3\ h^{-1}$

⇨ In the case of balanced mechanical ventilation with heat recovery: $n_{50} < 1\ h^{-1}$

3. MEASUREMENTS: RESULTS AND DISCUSSION

3.1 Global airtightness

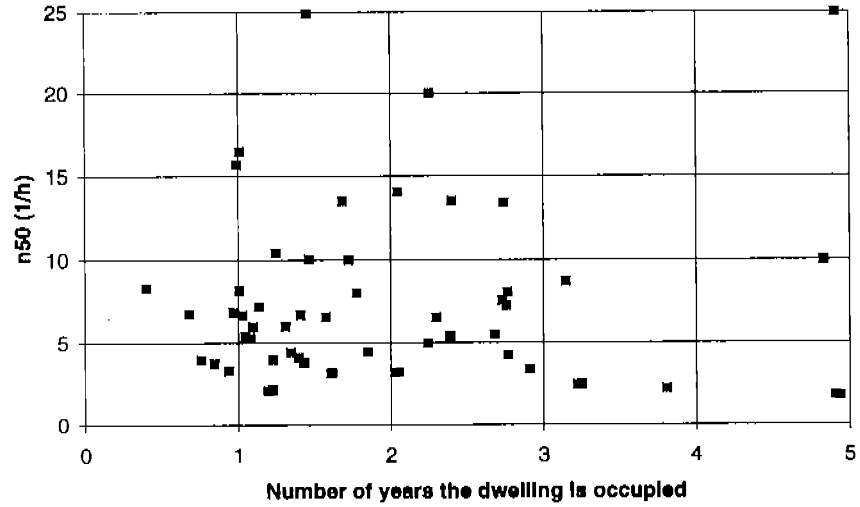

Figure 2: Global airtightness as a function of the number of years the building is occupied

The following remarks can be made:

⇨ There is a large scatter on the results of the global airtightness: the values are situated between $1.8\ h^{-1}$ and $25.0\ h^{-1}$; this means a difference by more than a factor 10. The average n_{50}-value is about $8\ h^{-1}$.

⇨ The worst results are caused by a poor finishing (at inclined roofs, connections between windows and walls, etc.), which is probably due to the fact that a lot of owners do a part of the work themselves.

⇨ It is important to mention that the measurements were performed on the insulated volume and not only on the heated volume, which means that the garage and the attic are often part of the measured volume.

⇨ On first sight there seems to be no clear relation between the airtightness and the number of years the building is occupied. One of the dwellings with a n_{50} of $25\ h^{-1}$ was already occupied for 5 years !

⇨ The airtightness seems to depend strongly on the type of building. This is shown clearly in Figure 3, where the average, minimum and maximum of the measured n_{50} values are shown for each building type. This tendency is caused by the fact that the airtightness of common walls is normally better than of external walls, due to the absence of windows, doors, etc.

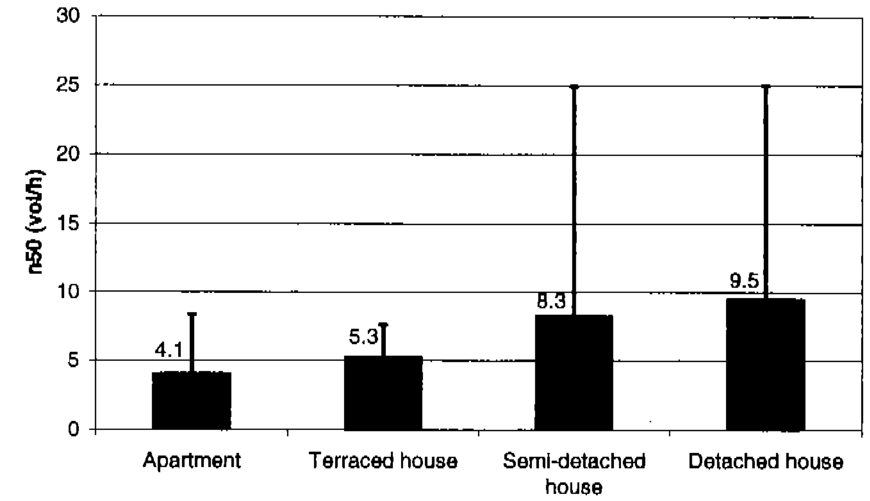

Figure 3: Airtightness as a function of the building type

3.2 Airtightness of separate rooms

To determine the distribution of the leaks over the building envelope, the leakage air flow rate at 50 Pa (Q_{50}) was determined for most of the rooms, by means of a compensating flow meter.

3.2.1 Garages and insulated attics

A first remarkable conclusion is that the most important leaks are usually found in garages and attics. The following figure (histogram) shows the contribution of garage and attic to the total leakage of the building. It can be seen that the influence of these rooms can be very high (up to more than 50% of the total leakage). On average garages and insulated attics are each responsible for 1/3 of the total leakage.

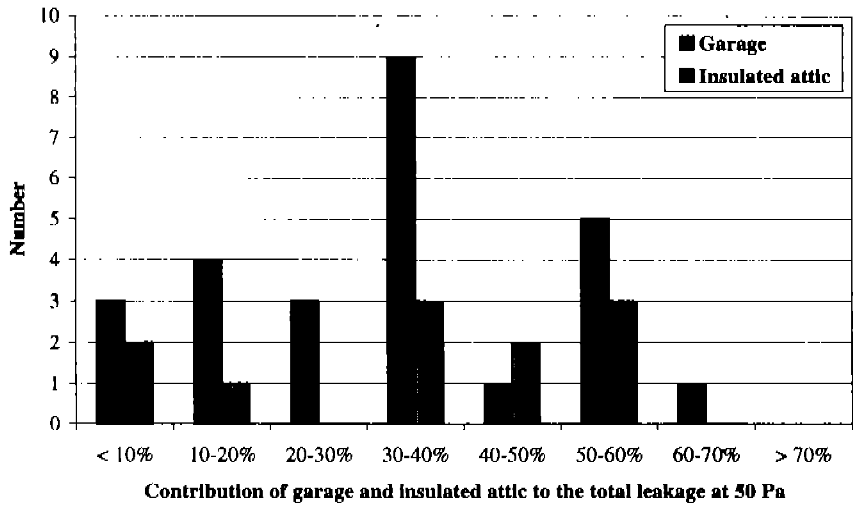

Figure 4: Contribution of garage and insulated attic to the total leakage at 50 Pa.

The inclusion of insulated, but unoccupied, garages and attics in the measurement of the global airtightness is an approach that is open to discussion:

⇨ On the one hand, these rooms are insulated with the objective to reduce the heat loss, and as a consequence the airtightness has to be very good in order to reduce heat loss by infiltration of outdoor air.

⇨ On the other hand, these rooms are not occupied, which means the comfort requirements are not so high. Moreover, the ventilation standard NBN D50-001 requires a rather important basic ventilation in garages, which is often obtained by leaving a gap at the bottom of the gate of the garage.

In order to demonstrate the importance of the chosen approach, the n_{50} was recalculated for the houses with a garage, assuming that the garage is not a part of the insulated volume of the house. This was done in the following way:

$$n_{50(without\ garage)} = \frac{\left(Q_{50(total)} - Q_{50(garage)}\right)}{\left(V_{total} - V_{garage}\right)}$$

The following figure shows the result of this calculation.

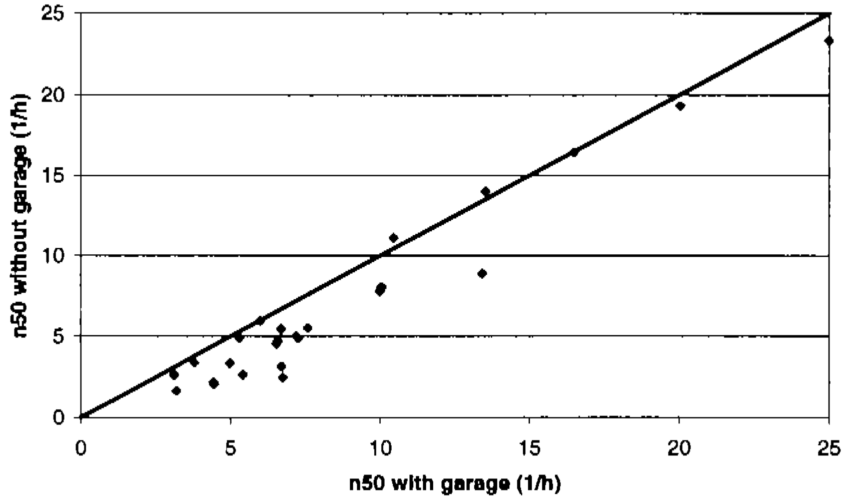

Figure 5: Influence of the garage on the n_{50}-value

One can see that mostly the n_{50}-value decreases when the garage is not taken into account. In the case of airtight garages with a high volume there can however be a slight increase.
On average the n_{50}-value decreases with 1.7 h^{-1}, but sometimes the effect is more important.

3.2.2 Other rooms

The measurement of the leakage distribution over the different rooms revealed that certain types of rooms are usually very airtight, especially bedrooms and bathrooms. Figure 6 shows the Q_{50}-values for the bedrooms of the different dwellings for which a pressurisation measurement was performed.
The following observations can be made:

- 70% of the bedrooms have a Q_{50} value that is lower than 100 m³/h. Assuming that the average pressure on the facade of a dwelling is about 2 Pa, the average infiltration air flow rate for bedrooms with a Q_{50} lower than 100 m³/h will be lower than 20 m³/h. The minimal nominal air flow rate necessary in a bedroom according to the ventilation standard is 25 m³/h. One can see immediately that in bedrooms with a high airtightness the presence of ventilation facilities is very important to avoid problems with the indoor air quality. This certainly doesn't mean that bedrooms should be made less airtight. On the contrary, a good airtightness combined with the presence of a correct ventilation device is the only guarantee for a good indoor air quality with a minimum of energy loss. Examination of the ventilation facilities in the 200 dwellings learns that most of the bedrooms don't have ventilation devices installed ! This is discussed in detail in [3].

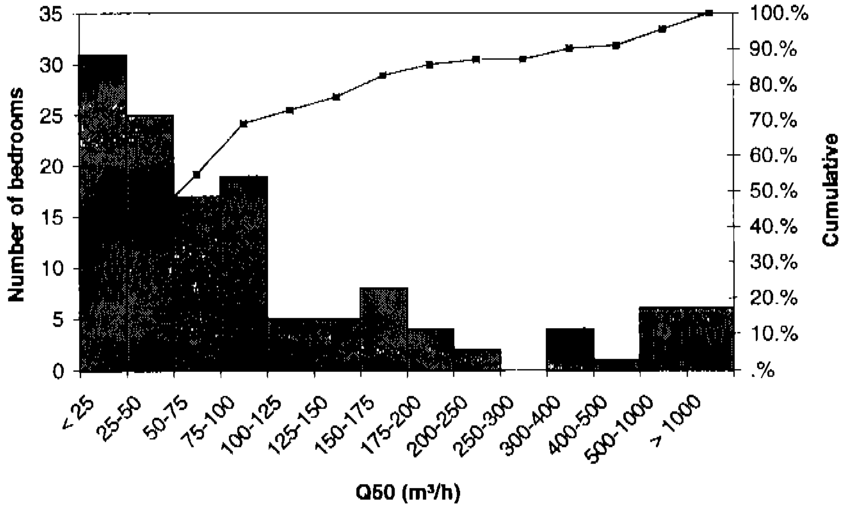

Figure 6: Q_{50}-value of the bedrooms from the SENVIVV-study

- About 10% of the bedrooms have a Q_{50} value higher than 500 m³/h. Mostly, these are bedrooms situated under an inclined roof, where the finishing was not done in an airtight way.
- Sometimes very airtight rooms can be found in dwellings with a very bad global airtightness, which demonstrates clearly that leaky dwellings are no guarantee for an acceptable indoor air quality if no ventilation devices are installed. This is shown in Figure 7.

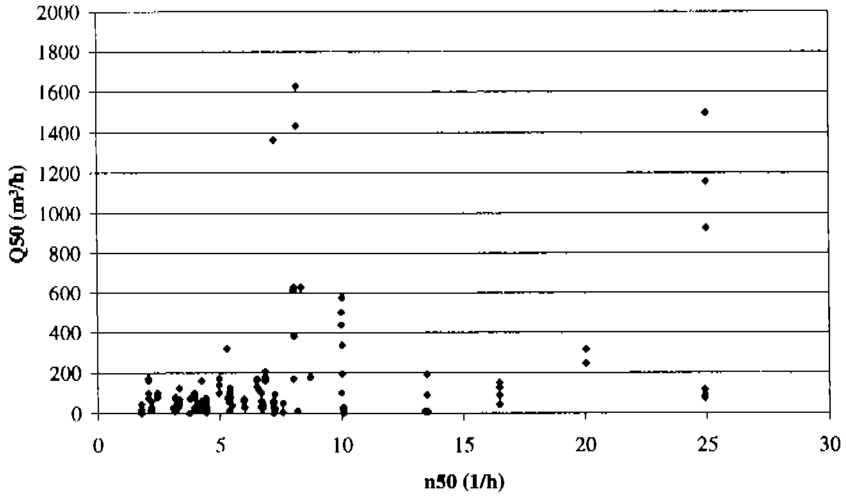

Figure 7: Relation between the airtightness of the bedrooms and the global airtightness of the corresponding dwelling.

The same remarks can be made for the bathrooms, where the situation is even more critical due to the high production of water vapour.

4 VISUAL ESTIMATION OF THE GLOBAL AIRTIGHTNESS

4.1 Goal and principle

The measurement of the airtightness of a dwelling is rather time-consuming. To investigate whether it is possible to estimate the lower limit of a dwelling's leakage in a visual way, a method was developed within the SENVIVV-study. It consists of assigning a Q_{50} value to the different building components of a dwelling and making the sum of all these values.
A distinction is made between the following groups of building components.

- Walls, floors and ceilings (Q_{50} per m²)
- Connections between wall and floor/ceiling (Q_{50} per m)
- Joints in cabinetwork (Q_{50} per m)
- Connections between cabinetwork and walls (Q_{50} per m)
- Openings and other leaks (Q_{50} per piece)

For each of these building parts Q_{50} values are given for several materials and building types. More explanation is given in [4].
The values used are based mainly on information from [5]. These values are averages of different measurement results with a rather important scatter. This means that a leak can as well be overestimated as underestimated. But, as certain leaks will be forgotten during a visual inspection, the final value will be rather a minimum value for the global leakage.

4.2 Results and discussion

The following figure compares the measured airtightness with the visually estimated value.

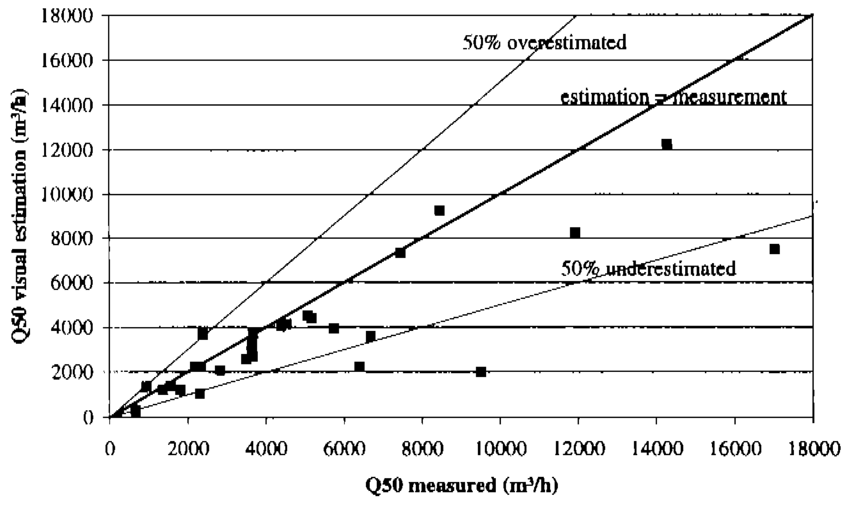

Figure 8: Comparison between the estimated and measured Q_{50} values for some dwellings of the SENVIVV-study.

The method was tried out in 28 dwellings. The following observations can be made:

- The visually estimated value seems to be smaller than (or equal to) the measured value in nearly all the cases. The method can therefore be seen as a reliable way to estimate the lower limit of a dwelling's leakage.

- ♦ In three cases the visual inspection gives an underestimation of more than 50%; in only 2 cases there is an overestimation of more than 50%.

5 CONCLUSIONS

The airtightness of Belgian dwellings is often very bad. An important reason for the worst cases is that the owners do a lot of the finishing work themselves, which is often only done after a long period of time and sometimes in a very bad way. On average, apartments have the best airtightness, while detached buildings are giving the worst results.
Garages and insulated attics seem to be responsible for an important part of the leaks in a dwelling. Even in dwellings with a bad airtightness some rooms can be very airtight, especially bedrooms and bathrooms.
It seems possible to make a quite good visual estimation of a dwelling's airtightness.

6 ACKNOWLEDGEMENTS

The SENVIVV-study was made possible by the financial support of the Minister of Economy of the Flemish Region (in the framework of the VLIET programme) and 16 companies active in Belgium (Insulation: *Isoglass, Isover, Owens-Corning, Rockwool Lapinus*; Glazing: *Glaceries de Saint-Roch, Glaverbel*; Ventilation: *Air Trade Centre, Alcoa, Aldes, Aralco, Bemal, Duco, Renson, Sobinco, Stork, Ubbink*).

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

AIRTIGHTNESS MEASUREMENTS IN THREE DWELLINGS IN ROME

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1-2-3 ENEA The Italian Committee for the New Technologies, Energy and the Environment
Department of Energy, Div. Rational Energy Uses

Synopsis

Airtightness measurements are not yet common in the Italian dwelling stock.In the framework of the MICA-ENEA contract, three dwellings were chosen to study the energy performance on the influence of natural agents. The majority of the dwellings in Italy still nowdays rely on natural ventilation and records of the fluidynamic and energy performance are not contractual documents among the parties involved. Since building airtightness is an important parameter to be investigated in case of natural ventilation, this study takes into consideration three typical examples of the Italian dwelling stock located in three different districts of the urban area and cover the range of constructions starting from the beginning of the century up to eighties before the actual heavy decrement of the new constructions .

The three dwellings are located in Rome and they have the following caracteristics:

1. dwelling located downtown in a 1915 building (Dw-1),
2. dwelling located in the south-east region of the city, in the residential areas built in the 70's (Dw-2),
3. dwelling located in the north region of the city, in the residential areas built in the 80's (Dw-3),

The instrumentation to be used includes a blower door, data logger and computer with the dedicated software to perform depressurization tests up to 50 Pa, to obtain ACH and ELA and in general all the parameters which might be of interest to foresee the behaviour of the dwellings during natural ventilation under the influence of the urban environmental conditions.

List of symbols

ACH@50 Pa	= Air changes per hour at a pressure difference ext.-int. of 50 Pa
EqLA	= Equivalent leakage area, cm^2.
ELA	= Effective leakage area, cm^2
Pa	= Pressure, N/m^2

Methods

A ***Minneapolis Blower Door mod-3*** was installed in two cases in the external perimeter of the dwelling, while in the newest one the door was installed in the entrance door.

The blower door consists of four components :

- Blower Door Fan
- Door Frame (aluminium model)
- DAB (Data Acquisition Box) 8 channels data logger plus portable computer in which APT (Automated Performance Testing System) ver 1.0 software was installed.
- Accessory case.

Blower Door Fan

The Blower Door Fan consists of a precision molded fan housing with a 550 W AC motor capable of moving up to 10 900 m^3/h of air. Air flow through the fan is determined by measuring the slight vacuum created by the air flowing over the fan inlet when the fan is operating. It is used to blow air into or out of a house. When air is blown out of the house, it causes a slight negative pressure or vacuum in the house relative to outside. This negative pressure induces outside air to enter the house (infiltration) through cracks or holes found in

any exterior house surface.The Blower door fan meets the flow calibration specifications of both CGSB Standard 149.10-M86 and ASTM Standard E779-87. To accurately measure fan flows less than 4 100 m^3/h of air, calibrated low flow rings are provided and are attached to the fan inlet. The standard Minneapolis Blower Door system comes with 2 low-flow rings capable of measuring as low as 510 m^3/h.
The blower door can be also used to pressurize the house by blowing air into the house and creating a slight positive house pressure relative to outside (exfiltration).

Door Frame
The door frame (and nylon panel) is used to seal the fan into an exterior doorway. It is adjustable to fit any size opening. Final adjustment and sealing are achieved by means of cam levers on the side of the assembler.

DAB (Data Acquisition Box)
The DAB contains 8 pressure channels, along with 8 analog voltage input channels.
Each pressure channel is comprised of a calibrated differential pressure tranducer connected to a pair of 1/8 " OD taps. The pressure channels have a swichable resolution of 0.1 to 0.5 Pascals, with a corresponding range of approximately +/- 400 or +/- 1 000 Pa. Each pressure channel has built-in auto zero capability.
When operating in the automated airtightness testing mode, channel P1 is used to measure the pressure in the building, while channel P2 is used to measure flow through the airtightness testing fan.

Since the confidence with depressurization tests was greater than pressurization or cruise tests, it was decided to start with these first and then the others in a further stage.
Sunny days having negligible wind velocities were chosen, to minimize pressure fluctuations and readings uncertainties, opening for hoods and fireplaces ducts were sealed.
Multi point depressurization tests were undertaken, as shown in the results section.[1]

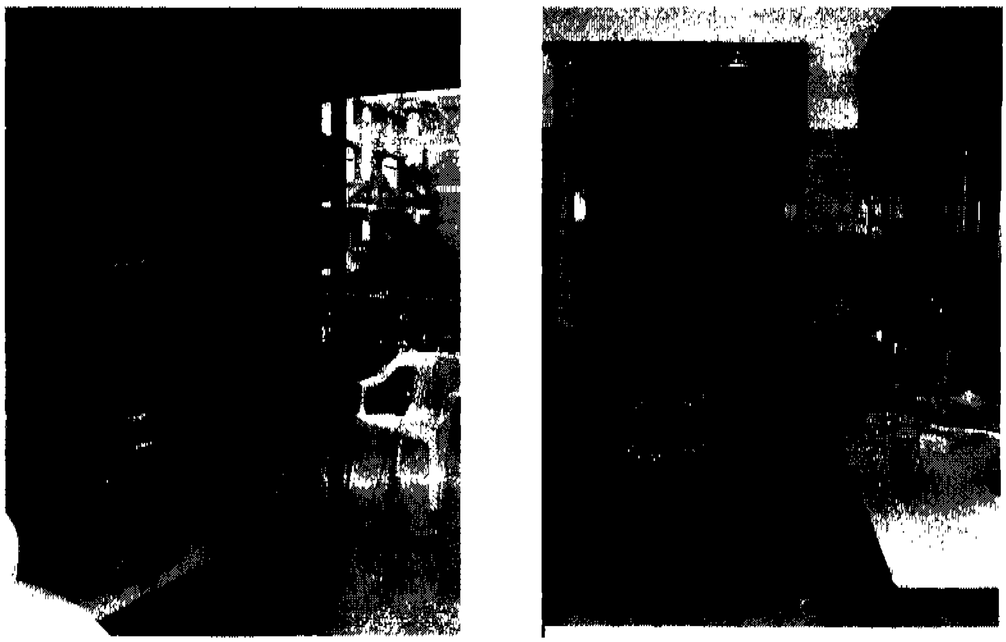


Photo 1- 2 Arrangement of the blower door in Dw-2

Results

Tab.1 Test results

	Dw.Vol. (m^3)	EqLA @ 10 Pa (cm^2)	LBL @ 4 Pa (cm^2)	ACH @ 50 Pa
Dw-1	341	806	458	5.4
Dw-2	322	1665	942	11.9
Dw-3	246	723	406	6.9

Equivalent Leakage Area (**EqLA**), is defined by Canadian Researchers at the Canadian National Research Council as the area of a sharp edged orifice (a sharp round hole cut in a thin plate) that would leak the same amount of air as the building does at a pressure diff. of 10 Pa.
Effective Leakage Area (**ELA**) was developed by Lawrence Berkeley Laboratory (LBL) and is used in their infiltration model. The effective leakage area is defined as the area of a special nozzle-shaped hole (similar to the inlet of the blower door fan) that would leak the same amount of air as the building does at a pressure of 4 Pa.[2] [3]

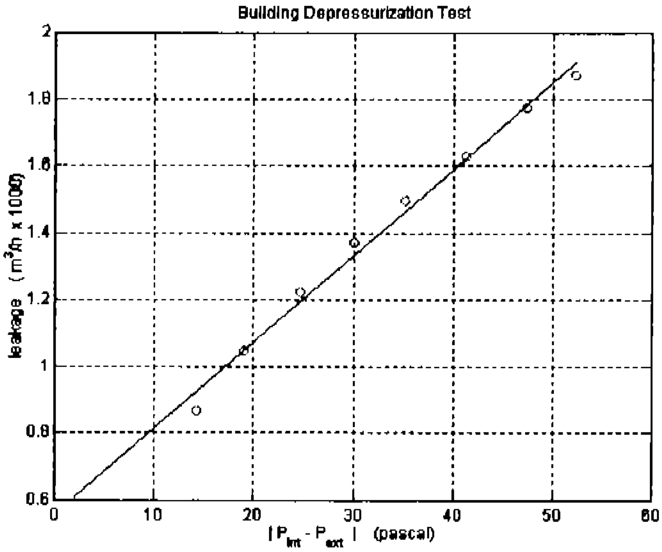
Once each airtightness test sequence was completed, a "best fit" line (called the Building Leakage Curve) was drawn authomatically through the collected blower door data.
The Building Leakage Curve can be used to estimate the leakage rate of the building at any pressure.
The Building Leakage Curve is defined by the variables Coefficient (C) and Exponent (n) in the following equation :

$$Q = C * P^n$$

where :
Q is airflow into the building (m^3/s).
C is the coefficient.
P is the pressure difference between the inside and outside of the building.
n is the exponent.
We had the following figures :

Tab.2 Building leakage curve

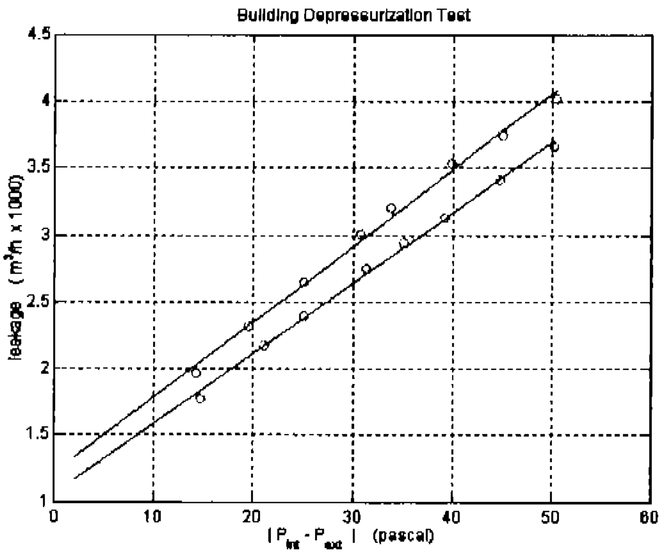
	Flow Coeff. (C)	Exponent (n)	Correlat. Coeff.
Dw-1	111.9	0.579	0.99954
Dw-2	229.6	0.583	0.96919
Dw-3	96.6	0.597	0.99993



DW–1

Date: 22-APR-1998
Address: Porta Pia
City: Rome - ITALY
Volume: 341.0 m^3
Floor: 111.9 m^2
Surface: 57.4 m^2

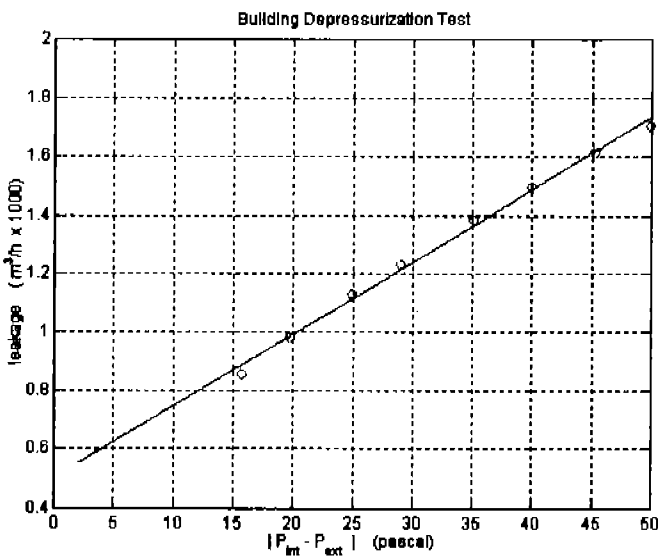
	ΔP (Pascal)	FLOW (m3/h)	FLOW (l/sec)
1	52	1871	520
2	47	1773	492
3	41	1631	453
4	35	1501	417
5	30	1370	381
6	25	1225	340
7	19	1047	291
8	14	867	241



DW–2

Date: 23-APR-1998
Address: Via Rocciatori 21
City: Rome - ITALY
Volume: 322.2 m^3
Floor: 107.4 m^2
Surface: 106.0 m^2

	ΔP (Pascal)	FLOW (m3/h)	FLOW (l/sec)
1	50	4014	1115
2	45	3738	1038
3	40	3531	981
4	34	3206	890
5	31	3007	835
6	25	2656	738
7	20	2310	642
8	14	1963	545
1'	50	3655	1015
2'	45	3407	947
3'	39	3139	872
4'	35	2948	819
5'	31	2758	766
6'	25	2399	666
7'	21	2171	603
8'	15	1771	492



DW–3

Date: 22 – APR - 98
Address: Serpentara - Lotto 12
City: Rome - ITALY
Volume: 245.9 m^3
Floor: 90.8 m^2
Surface: 101.7 m^2

	ΔP (Pascal)	FLOW (m3/h)	F LOW (l/sec)
1	50	1703	473
2	45	1617	449
3	40	1500	417
4	35	1387	385
5	29	1236	343
6	25	1132	315
7	20	982	273
8	16	855	238

Dwellings 1 and 3 are in the European continental average and range (3-10 ACH @ 50 Pa), while the 2nd is too leaky [4].
The equivalent leakage areas which the blower door software outputs, roughly correspond to the combined area of all the house's leaks. These data could be used in the codes to simulate different situations and conditions.
We have to consider that these results are not limited to the groups of the experts dealing with the matter. They have to be used and easily interpretated by the weatherization firms and by those field technicians who have no confidence with the theory.
We report many attempts and simple rules of the thumb to convert the blower door measurements into an average infiltration rate, to simplify the estimations of permeability, energy consumption, since the rate of infiltration is constantly varying while a single pressure test is available.
Experiments carried out in Sw and in Usa gave the result that assuming

$$\textit{average infiltration rate}\ \ ACH = \frac{ACH_{50}}{20}$$

the value was surprisingly reasonable. In this formula ACH_{50} denotes the hourly change rate at a pressure difference of 50 Pa between inside and outside.
Further to this, on the basis of the use of climatic data for North America a climate factor to reflect the influence of outside temperature (affects the stack effect) and windiness was developed [5]. Since this factor reflects both temperature and seasonal windiness, a cold, calm location might have the same climate factor as a warm, windy location.

$$\textit{Correlation factor},\ N = C * H * S * L,$$

where,
C = climate factor, a function of annual temperatures and wind (ranges from 14 to 26 for North America).

H = height correction factor

Tab.3

N° of stories	1	1.5	2	3
Corr.factor H	1.0	0.9	0.8	0.7

S = wind shielding correction factor

Tab.4

Extent of shield.	Well Shield.	Normal	Exposed
Corr.factor S	1.2	1.0	0.9

L = leakiness correction factor

Tab.5

Type of holes	Small cracks (tight)	Normal	Large holes (loose)
Corr.factor L	1.4	1.0	0.9

An estimate of the average annual infiltration rate is then given by :

$$\textit{average air changes per hour } ACH = \frac{ACH_{50}}{N}$$

This formula provides a more flexible value according to the different cases to be assessed. We have to point out that the field measurements might not be so abundant as required. Furthermore, multiple simulations can enlarge the investigation cases, different climatic conditions, different occupancy can be studied and evaluated.
During these last years our division achieved some capabilities in the use and diffusion of models to predict the trend of the concentrations of the main indoor pollutants in dwellings, offices and hospitals.
To calculate air flows and contaminant dispersal in multizone buildings CONTAM 96 by NIST (National Institute of Standard and Technology) was used[6].This code uses the multizone network approach to airflow analysis.The building is treated as a collection of zones connected by airflow paths.These zones may represent groups of rooms,individual rooms,or even portion of rooms,as well as shafts and portion of the building air handling system.Within each zone the temperature and contaminant concentration is considered to be uniform.The airflow paths include doorways, small cracks in the building envelope, and a simple model of the air handling system (AHU).

Procedures and local standards have been developed in North America and North Europe to find out where is the minimum tightness to be achieved [7].
Larger amounts of infiltration are tolerated in milder climates ;if weatherization works have to be performed is not completely assured from field measures, other factors must be accounted such climate, exposition and the operation of furnaces etc.
In the blower door handbook quick formulas easily give the economical result of weatherization works.

Conclusions

- The blower door resulted in a flexible and adjustable tool, easy to transport from a dwelling to another.
- Tests are brief, interventions and adaptations on the apparatus are minimal, the available software reduces manual calculations and therefore the tests might be undertaken by technicians having normal training.
- Tenants generally wellcome actions which may result in an immediate benefit for their dwelling, since they realize the possible savings in the expences for climatization.

- The empirical indexes are easy and impressive for the definition of infiltrations and exfiltrations. Studies to assess their full correspondence in our mild climates should be undertaken.
- There is not a defined minimum value below which weatherization and sealing should be stopped. Knoweledge of the local climate is necessary.
- Tenant of Dw-2 was surprised to find his apartment so leaky. He realized why in his opinion gas bills were high with respect to similar constructions.
- Tenant of Dw-2 immediately made a remedial work on the frame of the door of his kitchen ; this resulted in an improving of airtightness when the test was repeated (cfr. graph of Dw-2 which is given in two trends).
- Even if windows and doors are shop tested [8] and show in the modern frames values about 7 $m^3/h/m^2$ @ 100 Pa, site erection may be poor and result in high permeability.
- Confidence in computer simulations for indoor air quality will be greatly increased entering the real leakage data, since until now we had to rely on estimated values only.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

SIMULATION STUDIES ON A KITCHEN VENTILATION SYSTEM

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Simulation Studies on a Kitchen Ventilation System

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Synopsis

The efficiency of a kitchen ventilation system is usually determined by its ability in heat and effluent removal. The main part of a ventilation system is the hood, with its face (or capture) velocity. Heat generation associated with the cooking process is the main factor that affects the thermal comfort. The heat removal capability is studied under different capture velocities so as to determine the minimum requirement for efficient removal of heat and effluent. Four arrangements of make-up air are simulated, with air coming from the front of hood, from the ceiling, from the underneath burner and from the wall. Various angles are also attempted as the direction of the make-up air coming from the wall. Finally, the interaction between the kitchen and the refreshment area will be studied. All simulation works were performed using the CFD package, FLUENT (V4.3).

1.0 Introduction

During the cooking process, the chemical and physical characteristics of food are changed, with impurities such as grease, smoke and small solid and liquid particles being produced and dispersed into the ambient air. The purpose of a well-designed ventilation system is to remove the impurities and part of the heat created by the cooking equipment, so as to provide a comfortable and hygienic environment.

Many types of food centres exist in Singapore; two main groups are those without air conditioning, such as hawker center, and those air-conditioned food centers in shopping centres and hotels. Both groups show the same problems associated with their kitchen ventilating system. The indoor climate of both kitchens and refreshment area of a hawker center is often unsatisfactory which have significant effects on the workers' productivity and customers' comfort.

Ventilating techniques have been provided in the hawkers' centers in Singapore for many years. However, many of them are not functioning properly due to bad design, incorrect usage and lack of maintenance. Many factors affect the effectiveness of ventilation; supply and exhaust airflow are two main ones.

Chinese and Indian food enjoy great popularity in Singapore. The preparation of both kinds of food often involves deep-frying and therefore generates more grease, vapor and heat than that of western preparation. Since there has been a lack of studies on the food centers in Singapore, the present project considers the air motion in the Singapore kitchen, in particular, the contaminant-removing capability of oily cooking processes in the hawker centres.

2.0 Acceptable Thermal Comfort

Thermal comfort is that condition of mind that expresses satisfaction with the thermal environment [1]. Though regional climate conditions, living conditions, and cultures differ widely throughout the world, the preferred temperature that people choose for comfort has been found to be similar.

ASHRAE Standard 55 [2] provides the guideline for the environmental parameters for human occupancy, in which acceptable temperature is up to 29°C with humidity at the 60% upper limit. It is also noted that in this standard, air speed may offset increased temperature up to a maximum at 34°C. The health requirements in kitchen are: room temperature of kitchen at 28°C, the upper limit of the air humidity at 16.5g/kg dry air, and a relative humidity (R.H.) of 70% [3].

3.0 The Boundary Conditions for Computational Analysis

As the refreshment area is an "open concept" type, the thermal comfort is affected by the temperature and humidity of the ambient air, the heat and R.H. from kitchen and various activities in this area. One study[4] shows that the average daytime temperature in Singapore is 28°C, but the lowest R.H. of ambient air is well above 60%. So it is possible to keep the temperature to achieve ASHRAE acceptable operative temperature, but it will be difficult to keep its R.H. upper limit at 60%.

The model size of kitchens is based on the typical size of cells of the hawker centres in Singapore. In a 2-D model, the room is 2.8 m × 3 m, the hood is 0.6m×0.6m, and the hood is 1.1 m over the cooking surface. The grid used in the simulation was 30 (horizontal) × 32 (vertical). The cook's position is assumed to be at 0.4m in the front of the support (see Fig. 1).

The contaminants are produced at a high level of concentration over a short period of time. Both solid and liquid particles are generated during cooking. The solid particles are usually a result of the food burning and the generated carbonaceous particles. Large liquid particles will quickly fall back onto the adjacent surfaces due to gravitational forces. Most liquid droplets are very small, and they remain airborne and drift about before settling on a surface. Annis and Annis [6] sampled grease aerosol produced by five natural foods fried in an electric skillet. The median particle diameter of these particles is lower than 1 μm and its average diameter is about 0.5 μm. Average concentration is less than 1.2 mg/m^3. However, their study is rather basic because of the complexity of the particle size distribution existing at various stages of frying for a variety of cooking processes have not been taken into account.

The normal skin temperature of human being is 28°C [1]. Area B represents the burner head. The 400°C is chosen as its typical burner head temperature for common use [3]. Area A simulates the heat source, and the heating value is 4.5 kW. Although the acceptable R.H of standard 60% is lower than the normal condition in Singapore, it is still taken as the humidity of the make-up air. The 0.6mg/m^3 and 0.5 μm are selected as the average size of the particles. The initial conditions in the kitchen are: the temperature of fresh air 28°C, humility 80% and the initial temperature in the kitchen 33°C.

4.0 Simulation Results

4.1 Temperature and capture velocity

The capture velocity of hood is varied from 0~0.5m/s, with step increase of 0.1 m/s. From the temperature contours in Figs. 2 and 3, it is found that the capture velocity affects the temperature greatly. When the capture velocity increases to 0.3m/s and above, the temperature around the cook's working place is acceptable. As the velocity increases continuously, the temperature around the cooking surface could be controlled better, but there is no obvious temperature improvement observed around the cook. The results show that the comfortable thermal environment can be achieved by keeping the hood facing velocity at 0.3m/s or above.

4.2 Radiation temperature and capture velocity

The radiation temperature relates to the type of heat source employed. According to the radiation theory and equations used in FLUENT, the radiant flux is related only to the temperature and the location of the heat-emitting source. If the heat source is treated as the heat flux, specifying the capture velocity should have little effects on the radiation heat transfer. The simulation results support the theory. With the increase of capture velocity, there is no obvious change in radiation heat transfer and no obvious effects on the radiation temperature distribution (see Figs. 4 and 5). On the contrary, if the heat source is temperature specified, with the increase of capture velocity, the thermal condition of the kitchen could be improved. But the suitable capture velocity for the controllable radiation heat depends on the temperature of the heat source. Therefore, it could be concluded that the performance of hood has little effect on radiation temperature due to the heat flux, regardless of whether the make-up air is available or not, but the thermal condition could be improved if the heat source is temperature specified.

4.3 Humidity and capture velocity

The humidity removal ability of a hood is studied by generating a high humidity source from the cooking surface and observing the effect of the capture velocity on the subsequent distribution. The simulation results show that when there is no exhaust or when the exhaust velocity is lower than the escape velocity of vapor, the water vapor will disperse into the room rapidly and affects the comfort of the cook. Once the capture velocity is above the escape velocity of vapor, even at low speed, the vapor can be controlled well and humidity varies little around the cook. (See Figs. 6 and 7). In other words, once the capture velocity is already above the escape velocity of vapor, the increase of capture velocity does not have obvious effect on the vapor removal.

4.4 The particles and capture velocity

The kitchen contaminations are produced at a high level of concentration over a short period of time. The contaminations involved include particulate, moisture, heat, odors, and gases, and these contaminants can be produced in a variety of combinations. Solid particles are usually a result of an error in cooking that causes the food to burn, generating carbonaceous particles. Vegetables in particular are of a cellulosic nature, and they readily form these solid particles when burned.

The large liquid particles are formed through minor explosions within the cooking vessel and they are visible as they move through the air and spatter on the surrounding surfaces. The size of the particles formed this way ensures that most will quickly fall back onto the adjacent surfaces due to gravitational forces. The more common liquid particles are small and they remain airborne and drift about before setting on a surface. When oil are heated to elevated temperatures, there is evaporation. A mixture of warm air and warm evaporated oil molecules is carried upward by the thermal currents. As the mixture enters a cooler region, the oil vapor condenses into a liquid and is converted into very small particles.

Annis and Annis [5] studied five kinds of natural food, and the following are their results of the size distribution and concentration of naturally generated cooking aerosols in the region of a range hood. The average diameter of particles generated from the cooking surface is about 0.5 μm, and the average concentration produced is about 0.6 mg/m^3. As shown in Figs. 8 and 9, only when the capture velocity is equal or higher than the particle escape velocity can they be removed; otherwise, the particles will disperse into the room.

4.5 The location of the make-up air system

Most ventilation systems in a hawker centre are not operated in their full capacity. The performance of the hood when installed in different places is studied. At a capture velocity of 0.2 m/s, the make-up air can come from the front of hood, down of heavy duty, ceiling, and from the wall, as shown in Fig. 10. The simulation results show that when the capture velocity is low, location in front of hood has better performance than others. (The down location is the worst. The air current's effect on the heat removal could be understood if the velocity vector is referred. But the situation will change when the capture velocity increases.)

4.6 The direction of make-up air coming into the room from the wall

When the make-up air system is installed in the wall, by keeping the capture velocity at a constant speed of 0.2 m/s and varying the in-coming direction of fresh air, the performance of the hood is studied. If the angle varies from 30° to 45°, the temperature can be controlled better than other angles. Once the angle for fresh air coming in exceeds 60° or below 30°, the heat can not be removed in time. As a result, the cook will feel uncomfortable. The velocity vector profile display the velocity field which will affect the temperature field and therefore explain how air flow affects the performance of the hood (see Figs. 11 and 12).

4.7 The comfort in the refreshment area and capture velocity

The thermal comfort in the refreshment area is affected by many factors, such as the temperature and R.H. of the ambient air and by the heat and R.H. from the kitchen. But the main effect comes from the various heating activities in the area. When the capture velocity is large enough, both the R.H. and temperature can be controlled in the kitchen. Hence, both temperature and R.H. will not affect the thermal comfort in the refreshment area. However, the temperature contour (see Fig. 13) shows that there is no obvious thermal improvement in this area when the hood capture velocity increases. The main reason is that when the ratio of refreshment area to the kitchen area is too large, it is difficult to improve the thermal condition in the refreshment area, which depends mainly on the hood velocity. Moreover, there are many heating activities and heat sources in the refreshment area. For example, each food dish could have a temperature higher than 50°C. The radiation transfer from the cooking equipment in the kitchen is another heat source, which will affect the temperature in the area.

5. Conclusions

The operational performance of a hood has significant effects on the temperature distribution of the ambient air in the kitchen, and therefore affect the thermal comfort of the cook. But it has little effect on the radiation temperature heat flux specified. Once the capture velocity is equal to or large than the velocity of vapour escape velocity, the vapour could not disperse into the kitchen and the R.H. of the ambient air would not increase. The hoods installed at different places and the direction for make-up air to enter have different performance. The performance of hoods in the kitchens has little contribution to the thermal comfort in the refreshment area. The main factors that affect the thermal condition are the various activities in the refreshment area.

6. Acknowledgements

The authors acknowledge the financial assistance provided by the Ministry of the Environment, Singapore.

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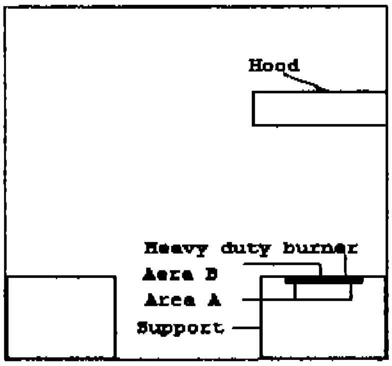


Fig. 1 Kitchen model for simulation

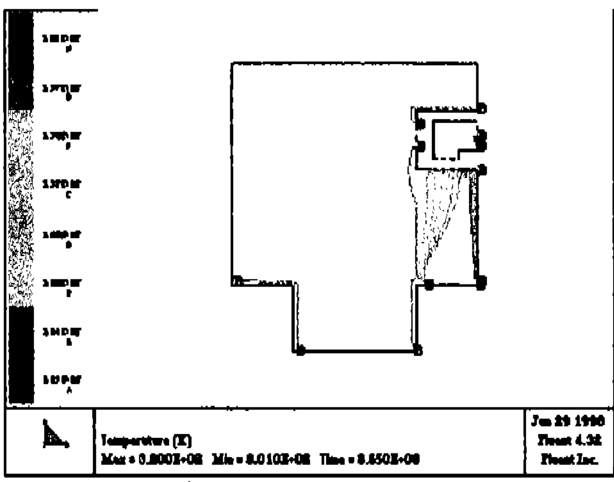


Fig 2. Temperature contour
(capture velocity 0.2 m/s)

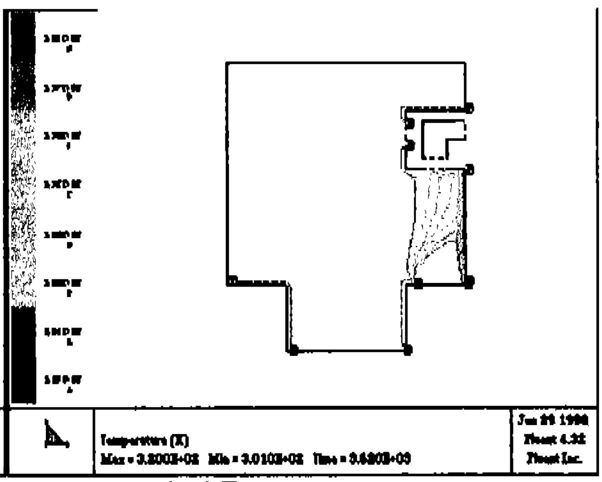
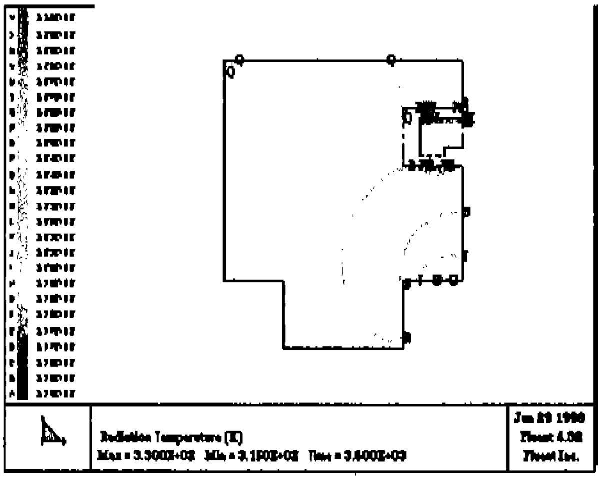


Fig. 3 Temperature contour
(capture velocity 0.4 m/s)



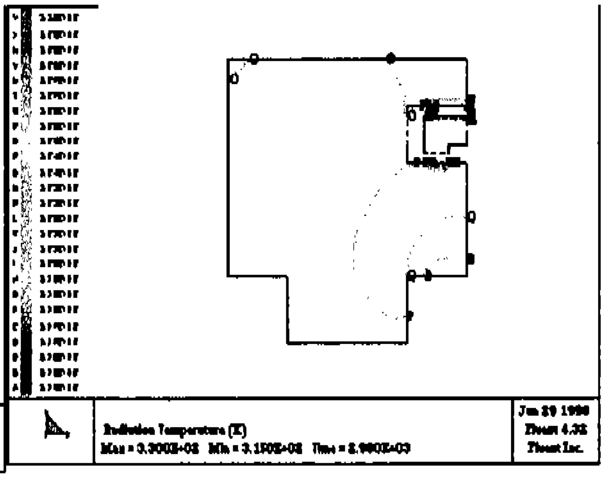


Fig. 4 Radiation temperature contour (capture velocity 0.2 m/s, only heat flux 4.5 kW available)

Fig. 5 Radiation temperature contour (capture velocity 0.5 m/s, only heat flux 4.5 kW available)

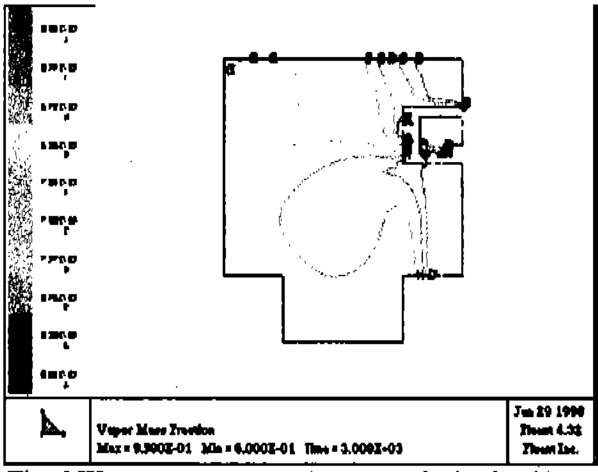


Fig. 6 Water vapor contour (capture velocity 0 m/s)

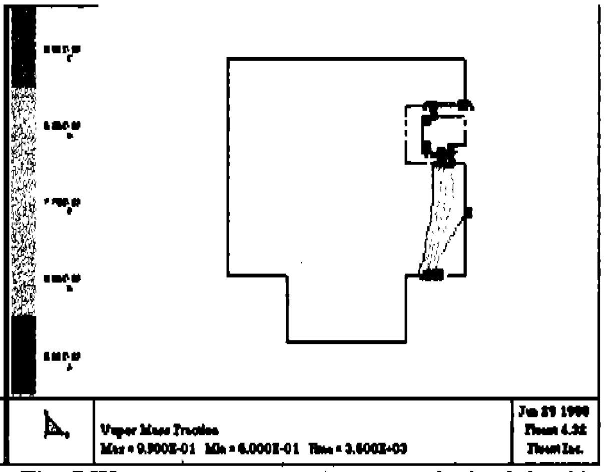


Fig. 7 Water vapor contour (capture velocity 0.1 m/s)

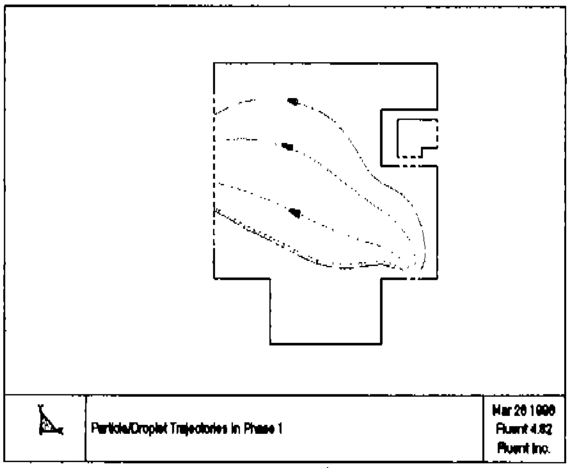


Fig. 8 Effluent trajectory (capture velocity 0 m/s)

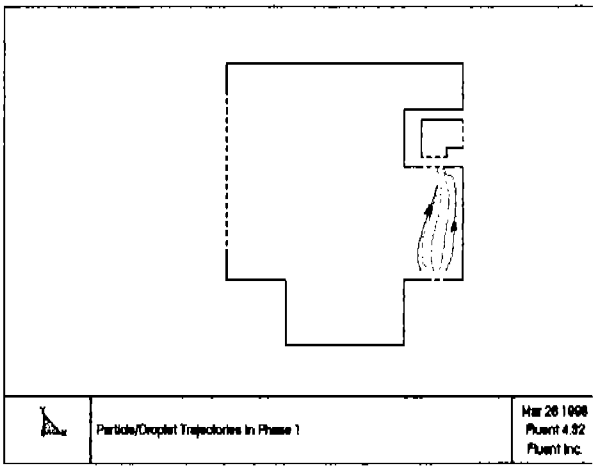


Fig. 9 Effluent trajectory (capture velocity 0.1 m/s)

(The lines in Figs. 8 and 9 are the track of effluent disperse)

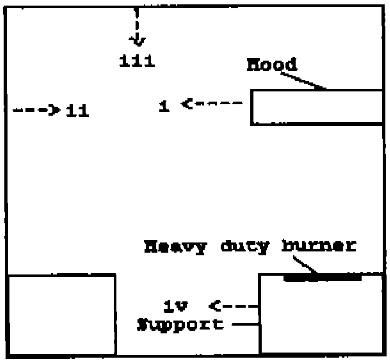


Fig. 10
i-- Make-up air coming from the front of the hood
ii--Make-up air coming from the wall
iii--Make-up air coming from the ceiling
iv--Make-up air coming from below heavy duty burner

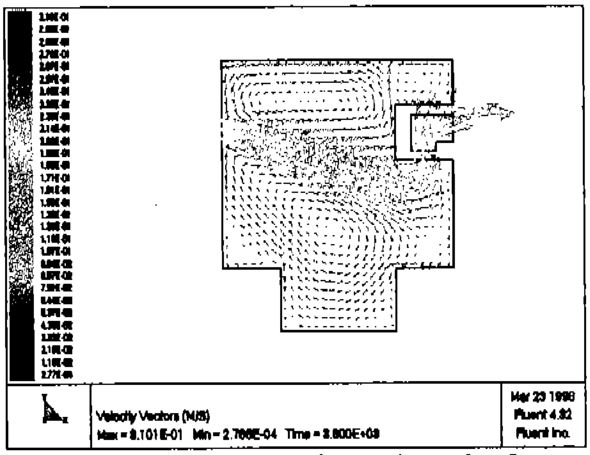


Fig. 11 Velocity vector (make-up air coming from the wall with an angle of 15°)

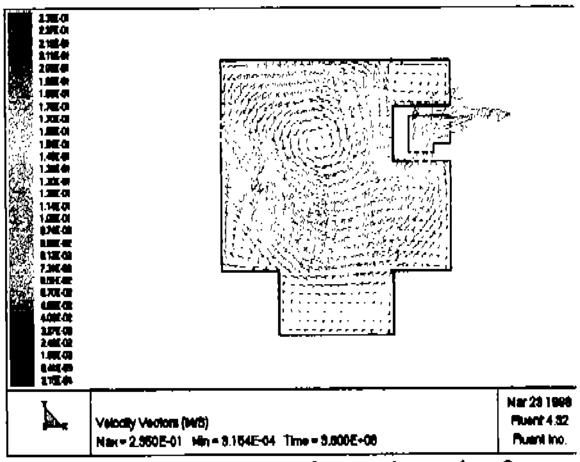


Fig. 12 Velocity vector (make-up air coming from the wall with an angle of 45°)

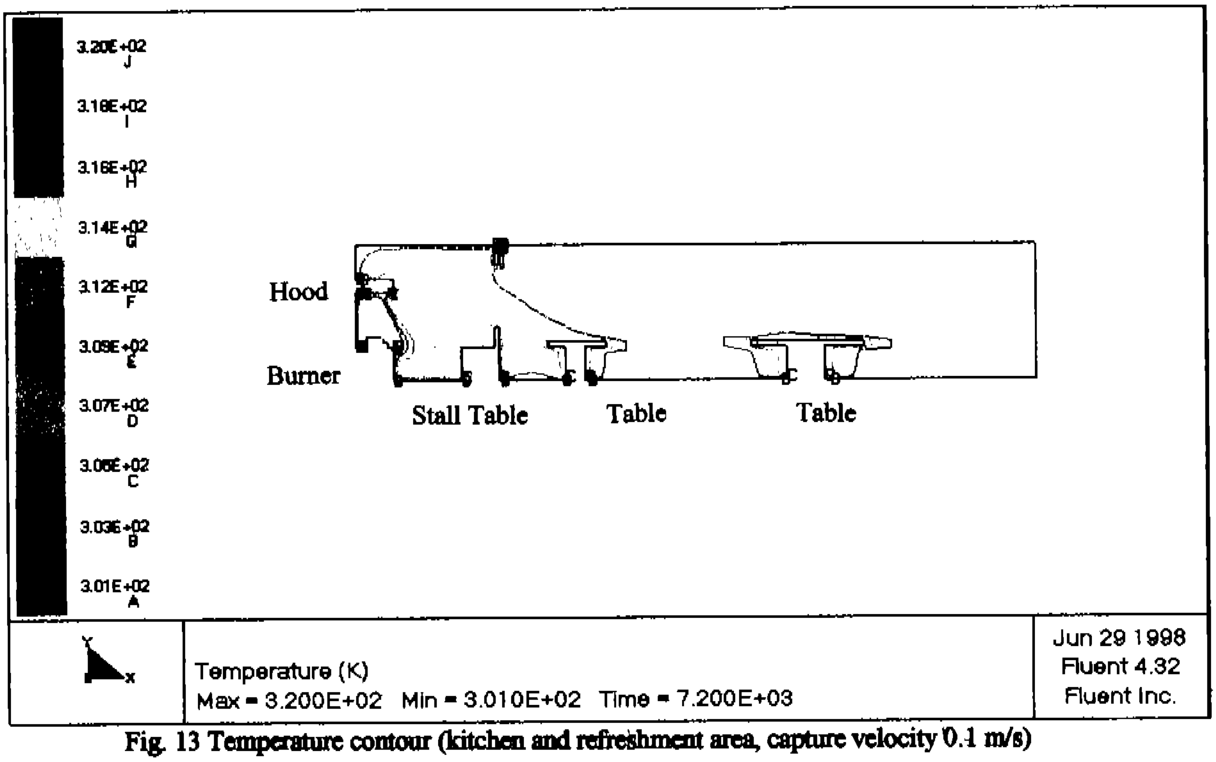


Fig. 13 Temperature contour (kitchen and refreshment area, capture velocity 0.1 m/s)

Besides that, a staircase step of dimension 0.3 m x 0.3 m is also included in the area of the bus. The domain of numerical calculation will cover up to 10 m to the right, left and top side of the bus, which will represent outside hot environment.

The domain inside the bus is given finer grid than the outside environment, being mapped with square cells 0.05 m x 0.05 m. The density of grid is reduced gradually to the left, right and top side from the domain of interest up to the extreme end of outside environment as shown in Figure 3.

2.2 Physical System and Boundary Conditions

The k-ε turbulence model is applied in this model with gravitational force activated because buoyancy plays an important role in the simulation. For the boundary conditions, the two air-conditioned jets blowing 20^0C cool air at an angle of 45^0 into the area of the bus are specified at 1 ms^{-1} while the air curtain is specified with constant temperature 20^0C, blowing cool air vertically downwards at the hot-cold air interface at various velocities. As for the boundary conditions of extreme end of outside environment and wall of the bus, they are specified as adiabatic boundary wall. The outlet is located at the centre top of the bus, and can be considered as return air for the air-conditioning system.

3.0 Results and Discussions

3.1 Without air curtain

When the door is opened, hot air from outside would flow in at the top, and cold air inside the bus will flow out from the bottom, as shown in Figure 4. This is a transient effect and the air exchange through the door is increased until it reaches a maximum at 7s.

The transient effect would cause the temperature inside the bus to increase exponentially until it reaches a maximum throughout the area beside the door of the bus at 20s. This is shown in Figure 5. As can be seen, the temperature at the high position is increasing faster than those at the low position, showing that hot air from the outside is coming into bus at the high position.

At that moment, the air-con jet near the door is not effective in cooling the bus area. Figure 6 shows that the area beside the door is filled with hot ambient air (30^0C), after the door is opened for 15s.The pressure difference across the door when it is opened is shown in Figure 7.

The air exchange follows the transient trend, and reduces the pressure difference to zero at the steady state, about 20 to 30s later. During the transient state, the pressure inside is higher than outside and this causes the airflow at the bottom greater than at the top. The situation lasting for about 20 s. Figure 8 shows the components of air exchange when door is opened.

When the door is opened, the air inside is getting hotter until it reaches the outside ambient temperature. This hot air is brought back to the bus evaporator through the return air duct. Assuming that the jet temperature is kept at 20^0C all the time, this will give an additional heat load to the cooling coil ($Q = m \times c_p \times \Delta T$). The mass flow rate m is equal to the mass flow rate from the jet, which is also equal to the mass flow rate through the outlet. The specific heat capacity C_p is constant at 1004 J/kg.K. and ΔT represents the temperature difference (T_{outlet} - 20^0C). Figure 9 shows the extra energy added to the cooling coil attributed to hot and cold air exchange.

After the door of the bus is opened for 60 s, it is closed back. At that moment, the inside area of the bus is at 30^0C. The temperature beside the door of the bus is investigated in order to find out how much time is needed to cool the bus area back to initial temperature. Figure 10 shows 240 s is required for the jet to cool down the bus back to initial temperature of 20^0C.

3.2 With air curtain

The performance of air curtain depends on many variables, including number of jets, thickness of air door, width of the doorway, height of the doorway, jet velocity, initial jet turbulence intensity and the pressure and temperature difference across the jet[5].

In this simulation, the jet velocity is taken into account in order to find out the optimum speed to seal the hot air from flowing into the bus. From the plot of the velocity vector in Figure 11, the flow of air curtain 2 ms^{-1} is not symmetrical at the hot-cold air interface. The pressure created by the difference in air density causes the curtain to break contact at the bottom part. In this case, the air curtain can only prevent the hot air flowing into the bus at the top. Some of the cool air which is flowing out from the bottom will be replaced by hot air and cause the temperature to increase. However, if the velocity of the air curtain were high, the rate of heat transfer through air curtain streamline would be increased. This can be shown in Figure 12, in which air curtain of 8 ms^{-1} will cause the average temperature to rise faster than 6ms^{-1}. Therefore, an optimum velocity of the air curtain, between 2 to 8 ms^{-1} need to be developed for the minimisation of air flow while still keeping the warm out.

Basically, 3 positions along the centre points (LOW[0.3m], MEDIUM[1m], and HIGH[1.8m]) are taken for temperature comparison for speeds 2 , 4 , 6 , and 8 ms^{-1}. Generally, the most suitable temperature for occupant comfort is 24^0C. Therefore, the air curtain can be considered as effective if it can keep the average temperature inside the bus less than 24^0C. Figures 13 shows the variation of the average temperature with different air curtain speed. From this figure, it is found that air curtain 6 ms^{-1} is capable of sealing the controlled environment within 24^0C for ambience 30^0C. Besides that, it can generally maintain a lower temperature as compared to 2,6 and 8 ms^{-1}. So, it can be concluded that an air curtain of 6 ms^{-1} is able to keep the warm air out. Again, the temperature contour in Figure 14 shows that 6 ms^{-1} can maintain temperature in bus cabin below 24^0C, even when the door had been opened for 60s.

As can be seen from the velocity vector in figure 15, there will be a full circulation of air from the air curtain source to the return air after the door is opened for 10s. Before 10s, there is a transient state. There is two small air circulation air at both sides of the curtain streamline, and hot air from outside will be entrained into the bus at the bottom. However, after 10s, the mass exchange through the door will be approaching steady-state and the rate of mass exchange is zero. As compared to the case without air curtain in Figure 16, the significant result of air curtain is that to reduce the transient state time, besides reducing the average temperature at 3 positions by 7^0C, as shown in Figure 17.

The additional heat load to the cooling coil can be calculated by $Q = M \times c_p \times \Delta T$. Although the temperature difference ΔT of (T_{outlet} - 20^0C) is reduced by 3^0C, the total mass flow rate through the return air M is increased by 0.5 kg/s (from 0.06 kg/s to 0.56 kg/s). This is because the air curtain is drawing cool air from the return air. Therefore, the airflow through cooling coil is increased. The product of $m \times c_p \times \Delta T$ for the case with air curtain 6ms^{-1} shows that an additional energy of 1.3 kW is incurred when compared to the case without air curtain, as shown in Figure 18. Therefore, for ambient 30^{0}C there will be an extra energy wasted if we need to maintain the temperature inside the bus less than 24^{0}C by using air curtain with constant flow rate and temperature.

Delaying door opening time before the air curtain start operating will provide a chance for air curtain to fully develop its streamline and enhance the sealing effect. Figure 19 shows the velocity vector by delaying door opening for 5 s. From this figure, we can observe that the longer the delay time, the more established the velocity vector in order to form a full circulation from the air curtain source to the return air. This will form a strong blocking air stream in order to prevent the hot air penetrating into the bus. Moreover, the pressure inside the bus will be uniformly distributed and this will reduce the air exchange through the door when it is opened.

Figure 20 shows a comparison of average temperature of 3 positions of air curtain 6 ms^{-1} with and without door delaying time. Basically, it can reduce the temperature inside the bus by at least 1^{0}C within 20 s. After that, heat transfer through the air curtain will cause the temperature to increase gradually. Nevertheless, delaying door opening time after air curtain operate is still an important factor in order to prevent hot and cold air mixing.

4.0 Conclusions

Simulation shows that without air curtain, the air exchange through opening door is tremendous and all the cool air will be replaced by hot air after 20 s. The process of air exchange is a transient state and it follows the quadratic trend from 0 to 20 s, with the maximum point occur at 10 s. By installing the "constant temperature (20^{0}C) air curtain", it is effective in preventing hot air going into the bus when the door is opened. However, using the 20^{0}C air curtain will increase the heat load to the cooling coil.

5.0 Acknowledgement

The authors gratefully acknowledge assistance given by Carrier Transicold Singapore Pte Ltd for the present studies.

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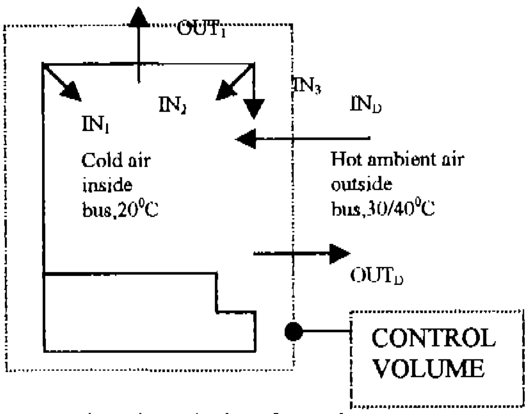

IN_1 : Air flow into the bus from air-con. Jet 20°C blowing 45° into the bus.
IN_2 :Air flow into the bus from air-con jet 20°C blowing 45° into the bus.
IN_3 : Air flow at the hot-cold air interface from air curtain, 20°C into the bus.
OUT_1 : Air flow out from the bus through the outlet, considered as return air.
IN_D : Air flow into the bus from ambient (30/40°C) through opening door.
OUT_D : Air flow out from the bus through opened door

Figure 1: Air flow when the door is opened with air curtain

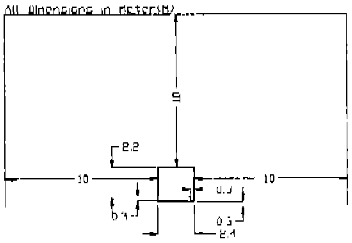

Figure 2: Dimension in 2d geometry set-up

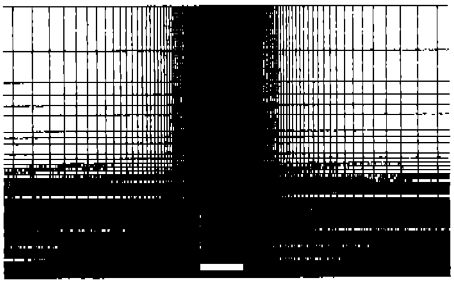

Figure 3 : Grid generation

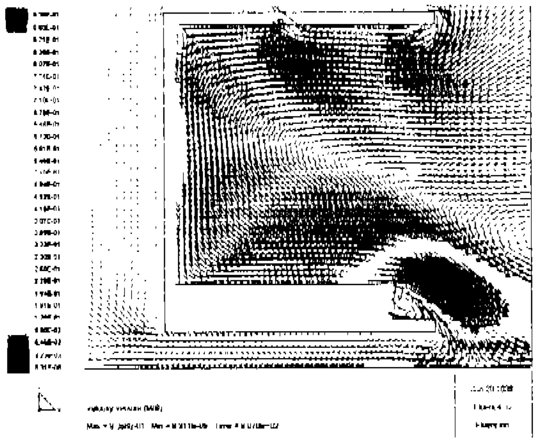

Figure 4: Air flow across the door after 7s opening, without air curtain

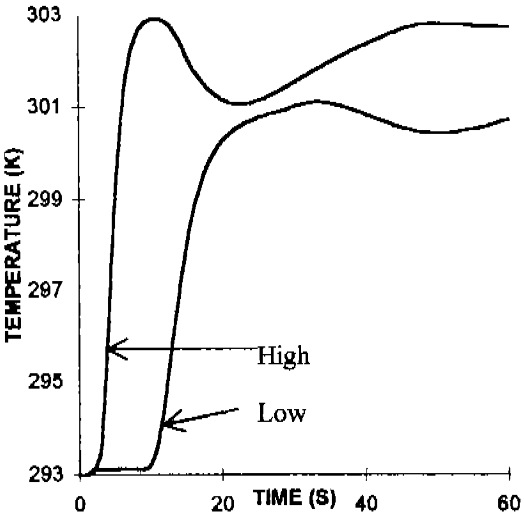

Figure 5:Temperature beside door of bus when door is opened without air curtain

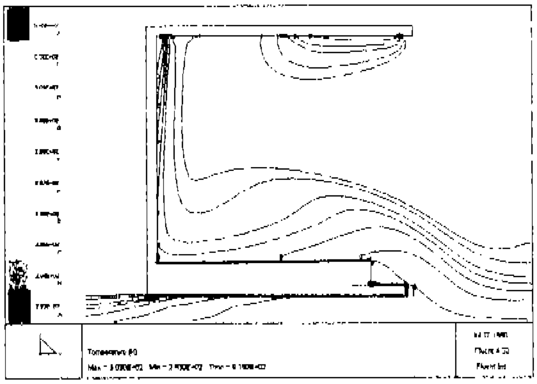

Figure 6: Temperature distribution for ambient 30°C, without air curtain, door opening time 15s

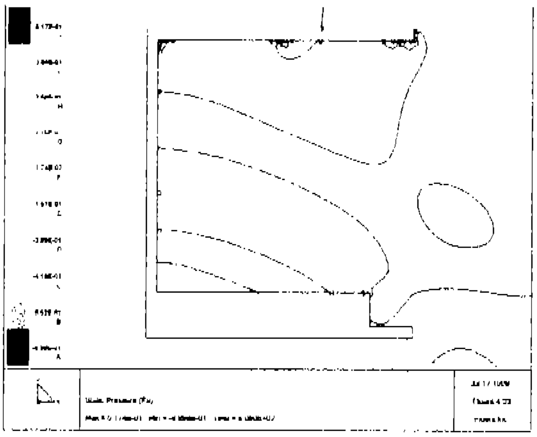

Figure 7: Pressure distribution for ambient 30^0C, without air curtain, door opening time 5s

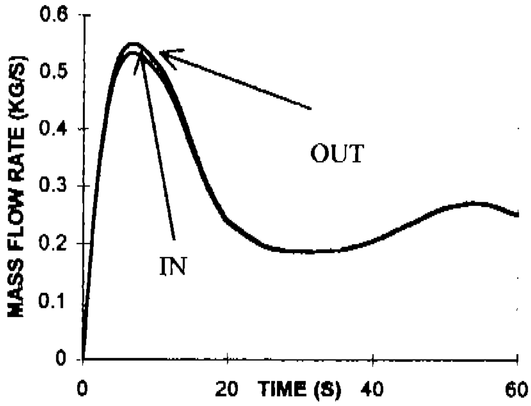

Figure 8: Variation of mass exchange through door for ambient 30^0C, without air curtain

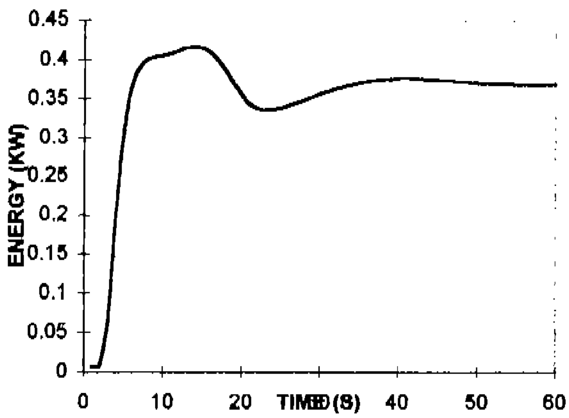

Figure 9: Additional heat load to cooling coil without air curtain

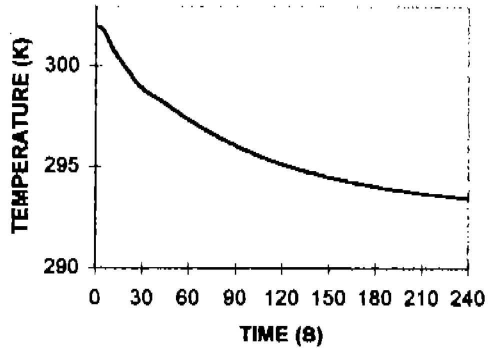

Figure 10: Temperature beside door of bus when door is closed back, without air curtain

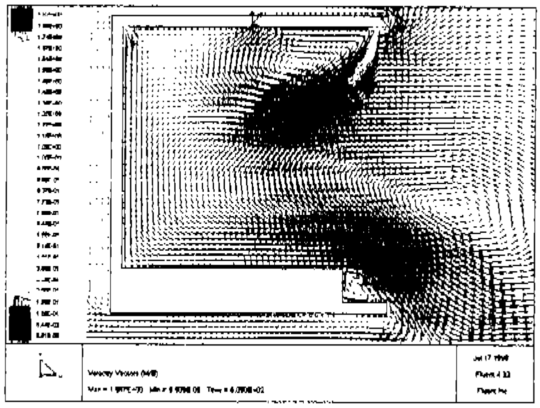

Figure 11: Velocity vector of air curtain 2 ms^{-1}, ambient 30^0C, 5s after door is opened

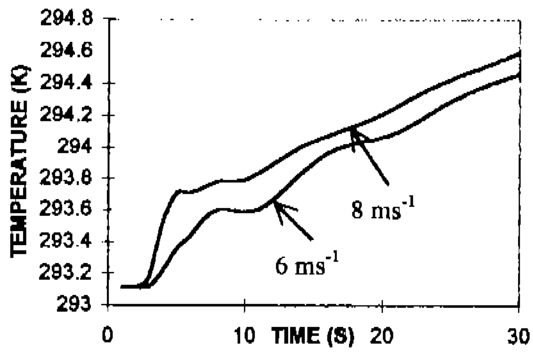

Figure 12: Average temperature beside door of bus with air curtain 6 & 8 ms^{-1} for the first 30s

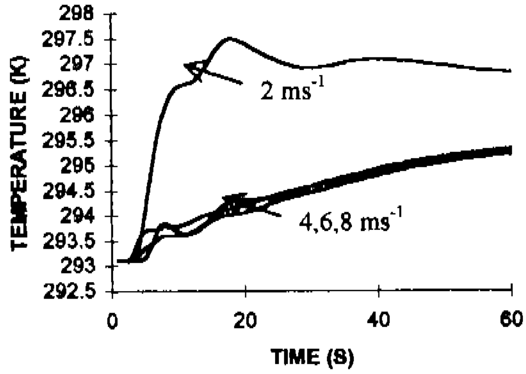

Figure 13: Average temperature for different air curtain velocity, ambient 30^{0}C

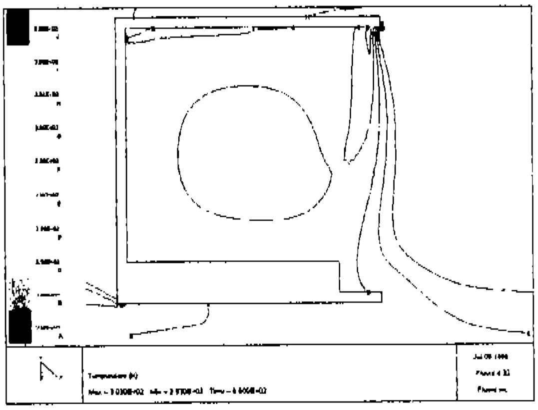

Figure 14: Temperature contour of air curtain 6 ms^{-1} for ambient 30^{0}C, 60s after door is opened

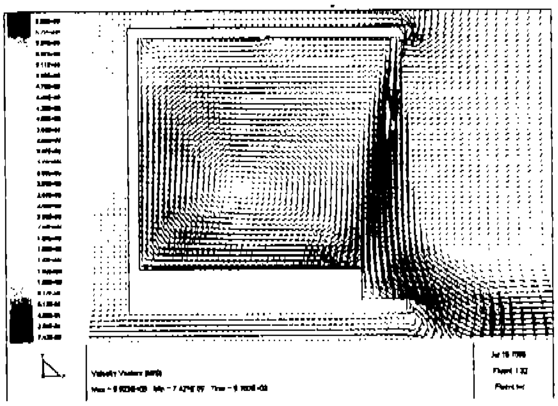

Figure 15: Velocity vector of air curtain 6 ms^{-1}, ambient 30^{0}C, 10 s after door is opened

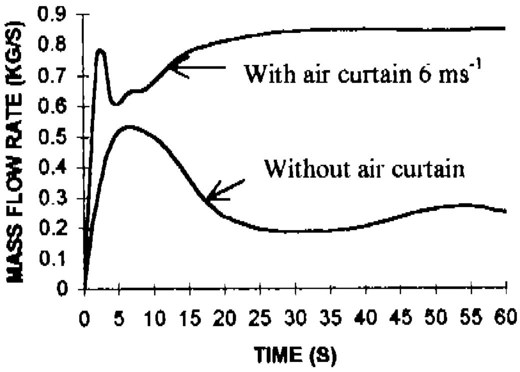

Figure 16: Mass exchange with and without air curtain

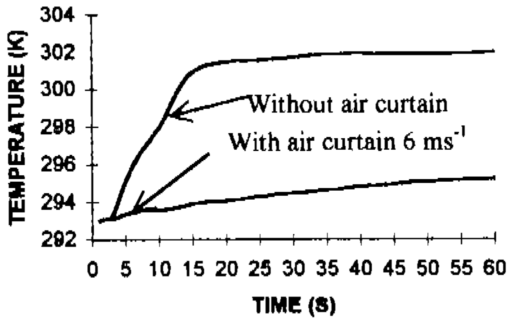

Figure 17: Temperature variation with and without air curtain

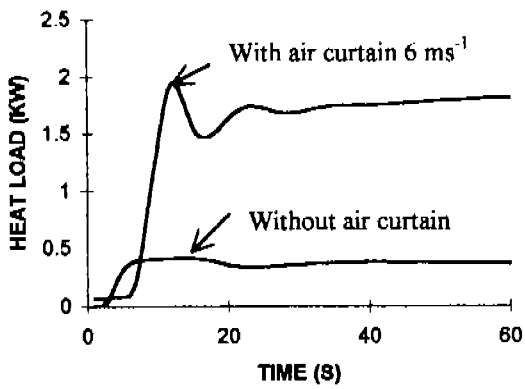

Figure 18: Additional energy for cooling coil with and without air curtain

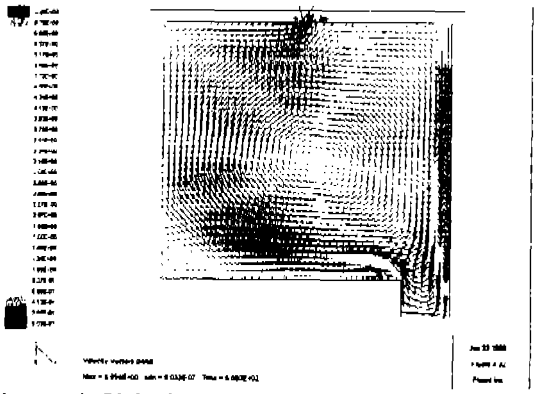

Figure 19: Velocity vector of air curtain by delaying door opening time for 5s

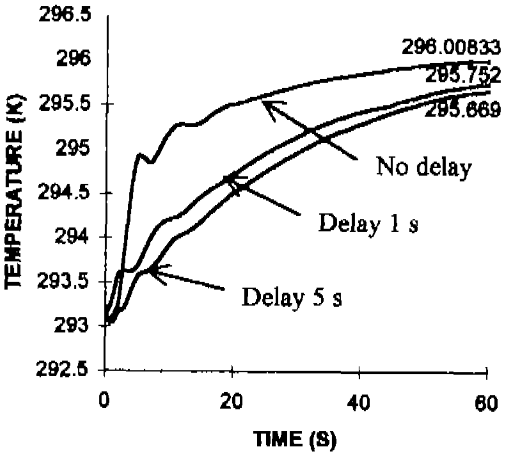

Figure 20: Average temperature of 3 positions of air curtain 6 ms^{-1}, with and without door delaying time

VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

A NOVEL VENTILATION / HEAT RECOVERY HEAT PUMP

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A NOVEL VENTILATION / HEAT RECOVERY HEAT PUMP

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SYNOPSIS

The trend towards improving building air-tightness to save energy has increased the incidence of poor indoor air quality and associated problems, such as condensation on windows, mould, rot and fungus on window frames. Mechanical ventilation / heat recovery systems combined with heat pumps offer a means of significantly improving indoor air quality as well as providing heating and cooling required in buildings.

This paper is concerned with the development of a novel ventilation / heat recovery system for the domestic market[1]. The new system has a high theoretical coefficient of performance and uses an "environmentally-friendly" refrigerant. In addition, the heat pump is compact and requires little maintenance. Computer modelling of the system has been carried out using different refrigerants. Several prototype systems have been designed, constructed and tested in the laboratory. These include ventilation heat recovery systems comprised of heat pipes with wire fins.

INTRODUCTION

Over recent years one of the main concerns in housing has been to build homes which are more energy efficient, thereby reducing heating costs, increasing occupant comfort and reducing the amount of pollutants released into the atmosphere by heating systems and power generation processes. This has mainly been achieved by increasing insulation levels, improved window technology, making the building shell more airtight and using efficient heating systems.

The trend towards improving building air-tightness to save energy has increased the incidence of poor indoor air quality and associated problems, such as condensation on windows, mould, rot and fungus on window frames, damp patches on walls and house mites in mattresses and carpets. Mechanical ventilation/heat recovery systems combined with heat pumps offer a means to significantly improve indoor air quality as well as provide heating and cooling required in buildings.

Recent studies have found that ventilating homes can reduce the number of house dust mites[2]. Dust mites aggravate asthma in over a third of asthmatic patients.

More than two million people in the UK are asthmatic and the numbers are showing a steady increase. In 1989 the National Health Service spent approximately £217 million on drugs for asthma, this is about 8% of the total NHS budget, 670,000 people in the UK could be suffering from asthma because they are allergic to dust mites[3].

In Denmark some studies showed that the health of people with asthma improved considerably when they moved to well ventilated homes which had few dust mites[4].

The heat pumps high performance means it consumes much less fuel than conventional heating boilers and so would emit a much lower quantity of CO_2, the principal contributor to the greenhouse effect. The importance of this was highlighted by the commitment made by the U.K. government at the Rio Earth Summit to return CO_2 emissions to 1990 levels by the year 2000. Although heat pumps are frequently employed for industrial and commercial applications, the domestic market in the U.K. for these systems has not been large and widespread use of heat pumps in domestic buildings has been limited due to their high capital cost and maintenance requirements.

A novel ventilation / heat recovery system has been developed that is compact, has a low capital cost and requires little maintenance. The unit will supply fresh air at 200 m^3/hr with effective heat transfer. This will provide 0.35 air changes per hour for a typical four bedroom detached house. This value complies with the ASHRAE Standard to maintain general indoor air quality.

DESCRIPTION OF THE SYSTEM

The ventilation/heat recovery heat pump is based on the integration of a rotary heat pipe/metal fibre impeller with a compressor and/or ejector unit, which allows air movement, heat recovery and heat pumping to be carried out in a single unit. The unit consists of a rotating heat pipe array which has metal fibre extended surfaces and incorporates a compressor or/and an ejector. The metal fibres act as impellers and efficient heat exchangers by presenting a large surface area to the air they move. The heat pipe array is contained within conventional centrifugal fan casings, arranged to allow warm extract air to pass over the evaporator section of the heat pump and cool fresh air to pass over the condenser section. Heat recovery is effected via a small charge of refrigerant which is vaporized in the evaporator of the heat pipe array then compressed to allow heat rejection to the cool fresh air in the condenser. In operation, the domestic ventilation/heat recovery system will extract stale air from the kitchen and bathroom, recovering heat from the air in the process. The recovered heat is upgraded by the compressor and/or ejector and is transferred via the heat pipe array to the incoming fresh air.

HEAT PUMP PERFORMANCE

The basic measure of performance of a heat pump is its coefficient of performance (COP), which is defined as the ratio of useful heating effect to the rate of energy input to the system. A BASIC language computer program has been written to model the steady state coefficient of performance of the ventilation/heat recovery heat pump. Equations of state were used to predict the thermodynamic condition of various refrigerants around the vapour compression cycle. The refrigerants investigated were water, methanol, pentane, R32, R407a, CARE 10, CARE 30, CARE 40 and CARE 50.

As this system is intended for domestic applications, the temperature of the extract air stream from which heat is recovered is taken as constant at 20°C. With the heat source temperature fixed, the evaporator temperature is determined by allowing a terminal temperature difference of 10°C between the refrigerant and the extract air, fixing the evaporator temperature at 10°C. The condenser temperature is determined by selecting a typical warm air supply temperature of 20°C and again allowing a refrigerant to air temperature difference of 10°C, fixing the condenser temperature at 30°C. Having determined the evaporator and condenser temperatures the COP of the ventilation/heat recovery system can be calculated and the results are presented in Table 1.

Refrigerant	Evaporator pressure (bar abs.)	Condenser pressure (bar abs.)	Compressor power (W)	Suction volume (m^3/hr)	C.O.P.
Water	0.012	0.042	64.7	149.7	15.44
Methanol	0.074	0.218	119.6	29.6	8.37
Pentane	0.50	1.14	78.6	6.63	12.69
R32	11.06	19.28	81.0	0.42	12.35
R407a	7.51	13.48	77.7	0.61	12.87
CARE 10	2.21	4.04	82.2	1.92	12.17
CARE 30	3.8	6.70	68.5	1.17	14.61
CARE 40	6.36	10.85	81.3	0.80	12.29
CARE 50	7.29	12.00	80.7	0.71	12.40

Table 1. Coefficients of performance for various refrigerants in the Ventilation / heat recovery heat pump.

As Table 1 shows, water is the refrigerant with the highest COP but it also has the highest compressor suction volume, meaning a compressor with a large physical size is required. As a compact and low cost system is desirable water is therefore considered to be unsuitable for use in this system. Methanol is also rejected due to its relatively low COP and high suction volume. The remaining refrigerants all merit further consideration: pentane for its high COP and operating pressures close to atmospheric, the remainder for their high COPs and low suction volumes.

ROTATING HEAT PIPES WITH METAL FIBRE EXTENDED SURFACES

The extended surfaces of the rotating heat exchangers must provide an efficient means of transferring heat from the extract air and to the supply air. A theoretical fin efficiency analysis of various extended surface profiles has been carried out for the above operating temperatures and optimum fin heights and profiles were obtained. Testing has been carried out on a rotating heat pipe array without a compressor to determine its effectiveness as a heat exchanger.

The Rotating Heat Pipe Heat Recovery System was tested to measure the efficiency of the heat pipe array as a fan and as a heat exchanger. Figure 1 illustrates the experimental set-up of the test rig.

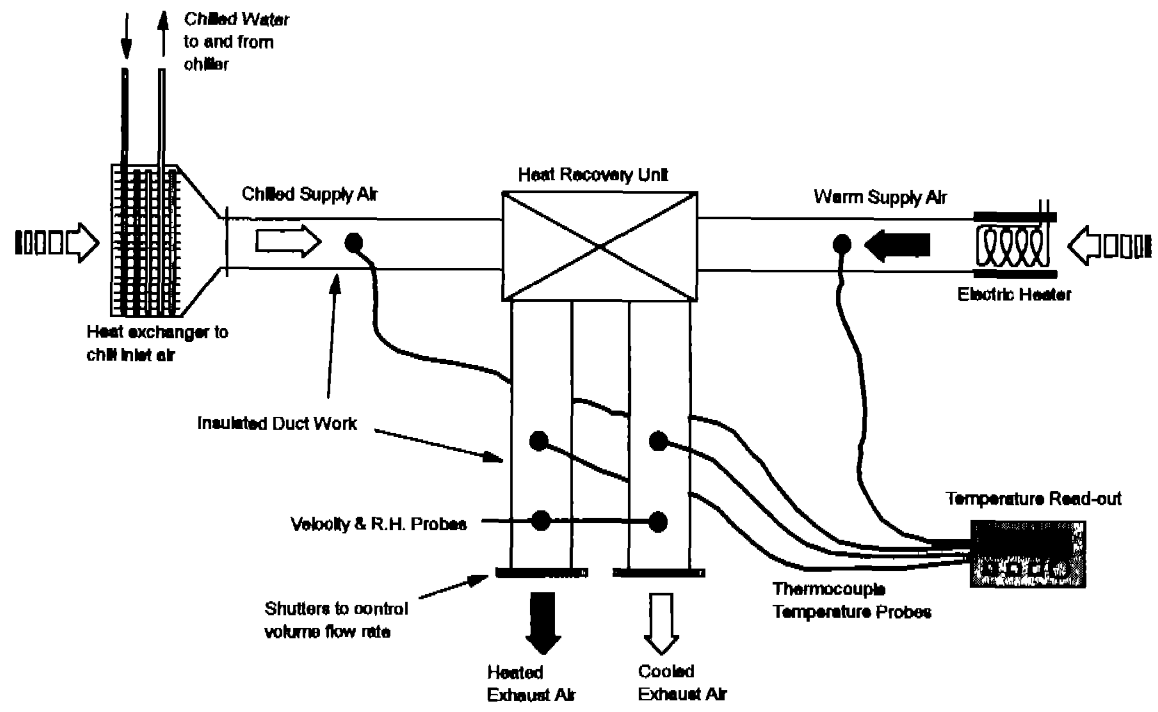
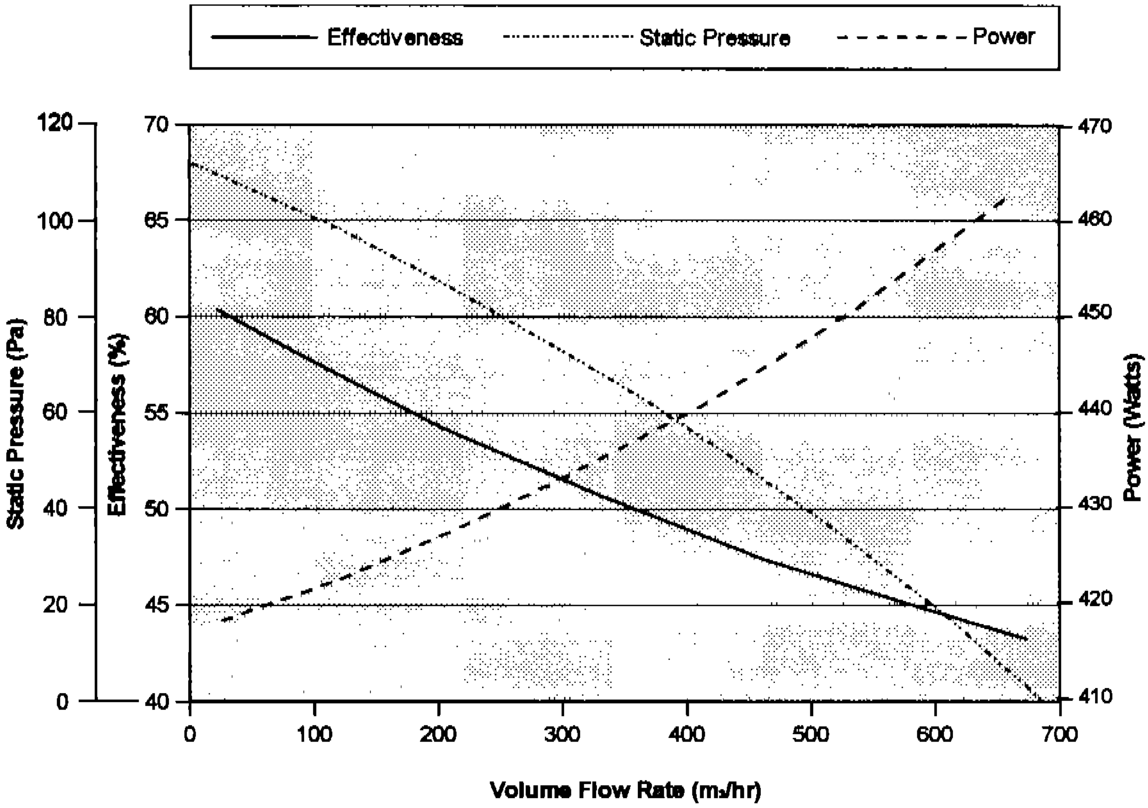


Figure 1: Experimental Set-up of Test Rig

The test rig was tested for a range of operating temperatures, rotational speeds, refrigerant volume fills, heat pipe numbers and geometry, fin heights and profiles. The range of results obtained were used to design the optimum heat pipe heat exchanger configuration.

Figure 2 shows a typical set of results for the Rotating Heat Pipe Heat Exchanger. The results shown are for a heat pipe array rotated at 900 rpm. Each copper tube was evacuated and contained a refrigerant fill of water. Screen mesh wicks were used to aid refrigerant return from the condensers to the evaporators. The evaporator and condenser sections of the pipes were covered with dense copper wire loop fins.



Speed of Rotation - 900 rpm,

Exhaust Air Inlet Temperature = 20°C, Supply Air Inlet Temperature = 0°C

Figure 2: Characteristic Curves for Heat Pipe Heat Recovery System

DISSCUSSION AND CONCLUSIONS

The rotating pipes with wire fins have been shown to provide adequate rates of ventilation. At 1480 rpm the rotating heat pipe impeller can supply air at $1200m^3/hr$. The static pressure produced by the impeller is more than adequate to overcome any duct losses in a whole house ventilation system.

Use of heat pipes alone without a compressor provides good levels of heat recovery. The Rotating Heat Pipe Heat Recovery System has 55% effective heat recovery at $170m^3/hr$. This compares with 65% effective heat recovery for commercial static plate heat exchangers handling the same air volumes.

Heat transfer rates could be improved by further optimising the design of the rotating heat pipes. Fin and pipe geometry could be enhanced to achieve this.

The levels of power consumption could be dramatically lowered by reducing the friction present in the fan casing seals and by reducing the overall weight of the rotating pipes and clamps.

Currently tests are being carried out on the heat pump system. Results to date compare well with the theoretical values illustrated and will be presented in a future paper.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

WIND DRIVEN VENTILATION IN COURTYARD AND ATRIUM BUILDINGS IN URBAN AREAS

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SYNOPSIS

A wind tunnel study was carried out to investigate the airflow through courtyard and atrium building models. Ventilation strategies resulting from the use of different atrium roof pressure regimes (positive pressure and suction) were examined and compared with the performance of the open courtyard. The model buildings were monitored both in isolation and in idealised urban environments of varying group layout densities. The effect of wind direction was also observed. The results from the study suggest that the open courtyard in an urban environment had a poor ventilation performance whilst an atrium roof with many openings operating under a negative (suction) pressure regime was the most effective. Changing the wind direction from perpendicular to the building façades to a 45° incidence angle had the effect of making the differences in the observed flows between all the models much smaller.

INTRODUCTION

The use of natural ventilation in non-domestic buildings is now seen as a sustainable approach to providing acceptable internal environments for building users. There has been much recent work in the UK [1] and Europe [2] to develop a better understanding of natural ventilation and to produce relevant design tools for engineers and architects. One building type that has been considered to offer great potential for natural ventilation is the atrium. It is has proved such a popular form since its 'reinvention' in 1967 with the construction of the Hyatt Regency Hotel in Atlanta that it is now unusual to find a new commercial building that does *not* contain some form of atrium.

The atrium building utilises both stack and wind forces to generate air flows through the room spaces adjoining the atrium well. In summer the stack force can be dominant and the atrium well acts as a chimney to vent warm air out of rooftop openings. For other times of the year wind forces may create pressure gradients between the outer façades of the building and the inner façades facing the atrium well. A complex mix of parameters will determine the magnitude of the wind-induced airflows. These include the magnitude and distribution of the pressure coefficients around and inside the atrium building, the leakage characteristics of the façades, the shape of the roof over the atrium well and the sheltering effects of surrounding buildings. For atrium buildings sited within a congested urban area the sheltering effect may be so great that external wind-induced pressures on the building's walls will be very small. Under these circumstances the atrium roof can be the key element for generating sufficiently strong positive or negative pressure gradients to induce satisfactory natural ventilation flows. The influence that atrium roof form, wind direction and surrounding buildings have on air flow through atria is the subject of this study, which was conducted using building models in a wind tunnel.

BACKGROUND

The use of roof elements to enhance the ventilation of buildings in built-up areas is not new. Wind towers and wind catchers have been prevalent in the Middle East and North Africa for hundreds of years, and have formed a key component of the indigenous architecture of those regions. Figure 1 shows traditional wind towers from Iran [3].

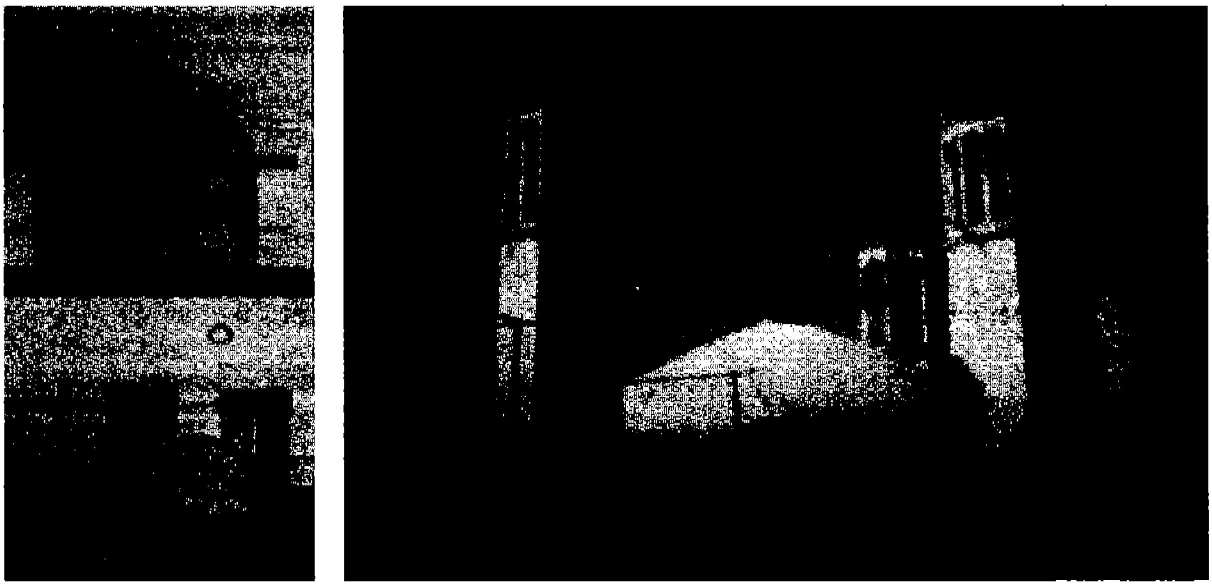

Fig.1 Wind towers of Iran (from [3])

Vents incorporated into the roof ridge and eaves provide a more integrated solution. Baumann et al [4] used a wind tunnel to investigate a 'jack roof' configuration for densely packed housing in hot humid climates, with vents placed in the sides of an elevated roof ridge. The jack roof was found to be effective in inducing internal air movement. A study by Riskowski et al [5] of the airflow performance of a range of commercial and fabricated ridge vents provided quantitative data for a range of wind speeds and directions.

The shape of the roof can also have a major role in inducing internal air movement. Kindangen et al [6] performed a CFD analysis of ten roof configurations to study their impact on airflow velocities and distributions. For the isolated, cross-ventilated dwelling modelled the shape of the roof did have an effect on the airflow patterns, and, in particular, air velocities. Wind direction, roof overhangs and roof heights were also important influences on airflow. Some studies of ventilation in atrium buildings have been carried out, mainly as either wind tunnel or CFD [7]. Most such studies have tended to investigate either isolated building models or have been trying to apply the results to a particular actual full-scale building. Little work has been done on a parametric analysis of the interactions between several factors, such as roof shape, atrium ventilation mode, wind direction and surrounding buildings. Such a parametric analysis would help identify the best combination of design parameters to maximise the benefits of wind-driven natural ventilation in atria. The experimental details of just such a parametric study are given below.

EXPERIMENTAL PROCEDURE

In order to evaluate the wind-driven natural ventilation in courtyards and atria in an urban setting a range of model buildings were constructed, instrumented and then positioned in a boundary layer wind tunnel.

Building models

The models represented four storey courtyard and atrium buildings at a scale of 1:100. The models measured externally 339 x 339 x 130mm high. The central courtyard / atrium was square in plan with the sides being equal to the height of the building (i.e. 130 x 130 x 130mm). The model walls representing room depth were 104mm deep. Monopitch roofs were placed over the courtyard opening to produce a range of ventilation strategies. Each roof was 52mm high, giving a roof pitch of just under 22°. The models were constructed from Perspex and consisted of rectangular building block modules that could be fixed together to create a range of model types. Together with the courtyard (model A0), four atrium roof ventilation strategies were to be investigated:

a)	closed roof	[model A1]
b)	suction	[models A4 and A5]
c)	positive pressure	[model A6]
d)	near atmospheric	[model A7].

These strategies, which were all achievable with the monopitch roofs, are shown schematically in Fig. 2.

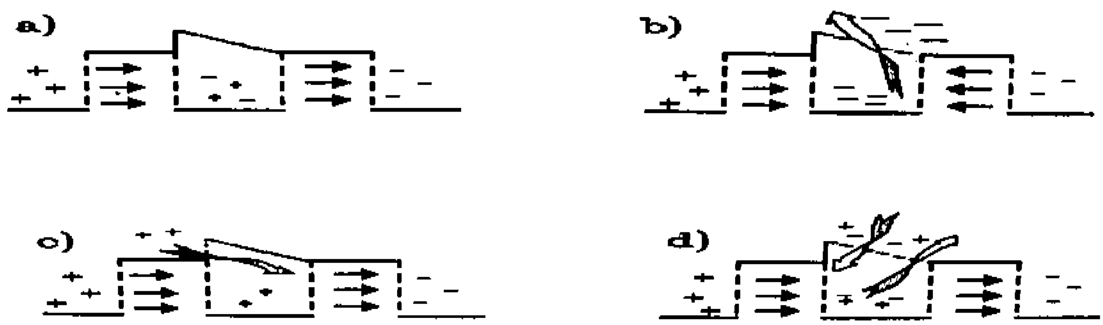

Figure 2 Atrium roof strategies for wind-driven ventilation

The models' walls and roofs were perforated with 10mm diameter holes to simulate building leakiness. The porosity of the walls (hole area to total façade area) was 11.4%. The porosities of the monopitch roofs (relative to the total façade area) were 0% for the closed roof (model A1), 11.4% and 30.4% for models A4 and A5 respectively, 11.4% for model A6 and 30.4% for model A7. Airflow rates through the models were measured directly with a specially made orifice plate device that was incorporated into one of the modular Perspex building block modules. The device was a square edge plate of 17mm diameter inserted between two short brass pipes of 25mm diameter, and fitted with two corner pressure tappings. The pressure drop across the tappings was measured using a digital manometer. The orifice plate was calibrated, in its Perspex container, against a precision commercially available flowmeter with an accuracy traceable to national standards. The dynamic pressure in the tunnel at gradient boundary height, together with the internal pressure in the atrium well at mid height, were also recorded. The flows through each model were monitored on each floor and on each façade of the model at two locations positioned centrally. The error in flow observations was estimated at approximately ±10%, reflecting the fluctuations of the manometer signals. The measurement arrangement is shown schematically in Figure 3 and the actual orifice plate located in an atrium model is shown in Figure 4.

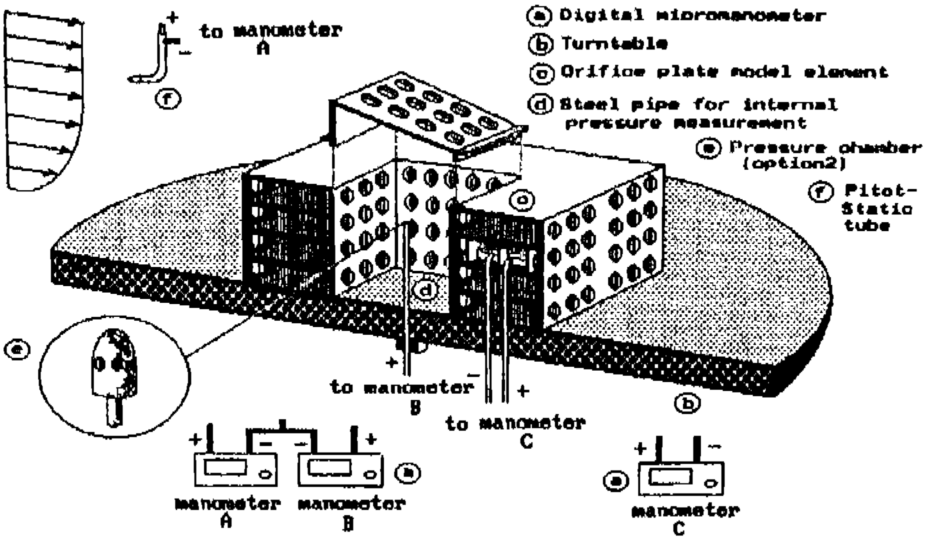

Figure 3 Schematic of measurement arrangement

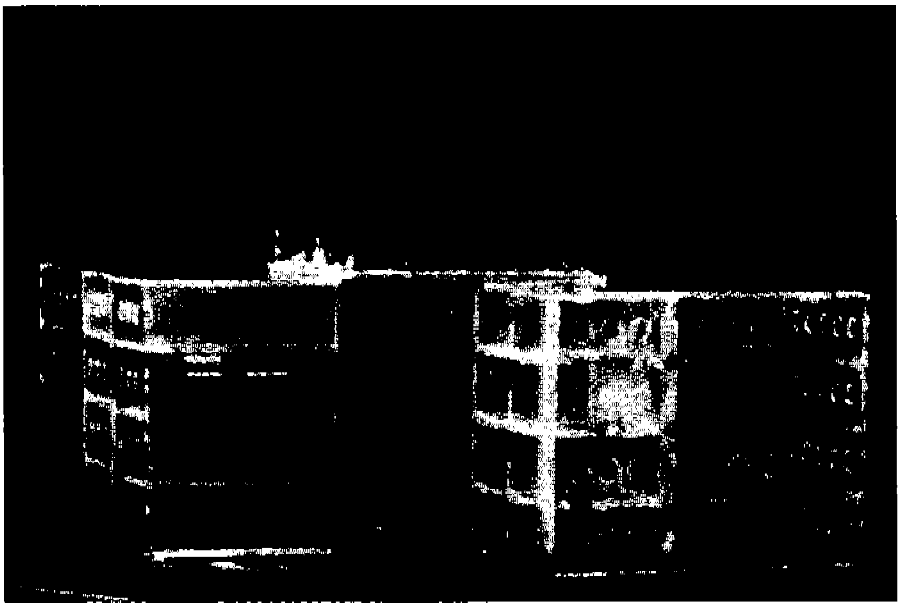

Figure 4 Orifice plate located in building model

The wind tunnel

The instrumented model was placed at the centre of a 1.1m diameter turntable in an atmospheric boundary layer wind tunnel. The turntable allowed two wind directions (0° and 45°) to be investigated. The tunnel had a working length of 7.2m with a cross-section measuring 1.2 x 1.2m, and a maximum gradient speed of 25 ms^{-1}. A series of spires, castellated fence and roughness elements in the wind tunnel generated a suburban type velocity profile at the turntable with a gradient height of 800mm and a power-law exponent of 0.245. The urban environment around the model was simulated by surrounding the model with rectangular wooden blocks of the same dimensions as the model, where the height of each block, H, was the same as the eaves height of the atrium and courtyard models. The blocks were arranged in either a uniform or staggered (checkerboard) arrangement. The wall-to-wall spacing between the blocks, Sc, was set at 1.5 and 2.3 times the building height H. The lateral spacing, L, was set to 0.5Sc for the staggered arrangement and to Sc for the in-line layout. These arrangements gave group layout densities of 0.28 and 0.40 for the uniform layout and 0.48 and 0.60 for the staggered layout. Group layout density is defined as the ratio of building plan area to building site area. A set of measurements was also made on all the models in isolation with no surrounding buildings.

The blocks were laid out to a fetch radius of 15H (three rows of blocks upstream and three rows downstream) as test results showed no change in measured airflows above this fetch value. The blockage in the tunnel was up to 8% at normal wind incidence (0°) and up to 11% for the 45° wind incidence direction. Although these values are a little high it was decided to apply no corrections to the results.

Each model was secured to the wind tunnel turntable and the required layout put in position. The tunnel was run for one hour to allow flow and temperature conditions to stabilise. The wind tunnel was run at its maximum gradient speed of 25 ms^{-1}, which corresponded to a speed at model eaves height of 16.4 ms^{-1}. These high speeds were used to ensure the highest Reynolds numbers possible. Each measurement consisted of logging the pressure drop across the orifice plate whilst simultaneously recording the dynamic pressure in the tunnel at the gradient height of 800mm with a pitot-static tube. The results were expressed as a non-dimensional flow coefficient CQl:

$$CQl = Q / (A \times V_{800}) \quad (1)$$

where Q is the flow through the orifice plate, A is the area of the openings in the model room block and V_{800} is the reference gradient wind speed. In other words, CQl represents the ratio of the velocity at an opening in the model to the gradient velocity. CQt was used to represent the average of all the CQl values measured on each model.

RESULTS AND DISCUSSION

The results of the experiments will be discussed in terms of the average model flow coefficient CQt and the minimum value of CQl observed in each model for the two wind directions.

At 0° wind angle

Among the structures tested, the courtyard model A0 had the poorest ventilation performance. For any of the urban layouts the CQt values were typically between 0.065 and 0.071, compared to a value of 0.126 for the courtyard in isolation. The minimum CQl was very low, being measured at one location at 0.02.

The closed roof atrium model A1 showed a slightly improved performance over the courtyard, with CQt values from 0.062 to 0.093 in the urban layouts (compared to 0.147 for the isolated case), and a minimum CQl of 0.04. However, the distribution of the flows within the closed model was uneven, with weak ventilation flows identified on the leeward side of the atrium. This suggested that improved ventilation could be achieved by encouraging flows to enter via the roof.

Atrium models A6 (with a porosity of 11.4%) and A7 (with a porosity of 30.4%) operated under positive and near atmospheric roof pressure regimes respectively and demonstrated better air flow characteristics than the previous models. Model A6 had CQt values from 0.086 to 0.119 (0.179 in isolation), whilst model A7 had CQt scores from 0.070 to 0.126 (0.162 in isolation). Minimum flow values for both models were raised to around 0.06.

Atrium models A4 and A5 both operated under a roof suction regime, with A4 having a porosity of 11.4% and A5 a porosity of 30.4%. This roof suction mode was in conflict with the negative pressure forces that were generated on the leeward walls of the model. As a consequence the CQt values for the low porosity roof A4 displayed little or no improvement over the other models, with a range of 0.083 to 0.088. The much greater roof porosity of model A5 created larger roof suction flows and removed the problem of the negative leeward wall pressures. CQt values were now found to be from 0.131 to 0.139, with a minimum CQl never falling below 0.10. It was observed that some of the CQt values were slightly higher for model A5 in an urban layout than when in isolation. This is thought to indicate that the negative leeward wall pressures may have been reduced in magnitude as the group layout closed up.

At 45° wind angle

Two major changes were observed when the wind direction was altered to 45°. Firstly, most of the CQt values for the models increased –that is, the ventilation performance of the models improved. Secondly, the range of CQt values between the different atrium and courtyard models for a given group density became much narrower, being typically within ±10% of the mean value for all models. The average CQt values were approximately 0.15 at a group density of 0.28, 0.12 at 0.40 and 0.48 densities and 0.09 at 0.60. The high porosity suction roof of model A5 still performed slightly better than the other arrangements, but the magnitude of the improvement was only significant for the highest group density value. The detailed results from the experiments are presented in Table 1, where the change in the CQt values produced by the sheltering effects of the urban layout are quantified. Figure 5 also includes the minimum values of CQl observed on each model.

Table 1 Average flow coefficients CQt for uniform (U) and staggered (S) group layout densities (D) and wind directions 0° and 45°

	CQt		CQt		CQt		CQt		CQt	
	Isolation		**U, D = 0.28**		**U, D = 0.40**		**S, D = 0.48**		**S, D = 0.60**	
Model	**0°**	**45°**	**0°**	**45°**	**0°**	**45°**	**0°**	**45°**	**0°**	**45°**
Courtyard A0	0.126	0.179	0.074	0.144	0.071	0.118	0.065	0.126	0.057	0.084
Closed roof A1, 0%	0.147	0.188	0.093	0.153	0.086	0.124	0.080	0.119	0.062	0.091
Suction roof A4, 11.4%	0.140	0.166	0.085	0.145	0.087	0.114	0.088	0.122	0.083	0.095
Suction roof A5, 30.4%	0.135	0.159	0.134	0.137	0.136	0.127	0.139	0.135	0.131	0.124
Positive roof A6, 11.4%	0.179	0.189	0.108	0.159	0.119	0.127	0.086	0.117	0.098	0.094
At. pres. roof A7, 30.4%	0.162	0.162	0.126	0.126	0.100	0.100	0.104	0.104	0.070	0.070

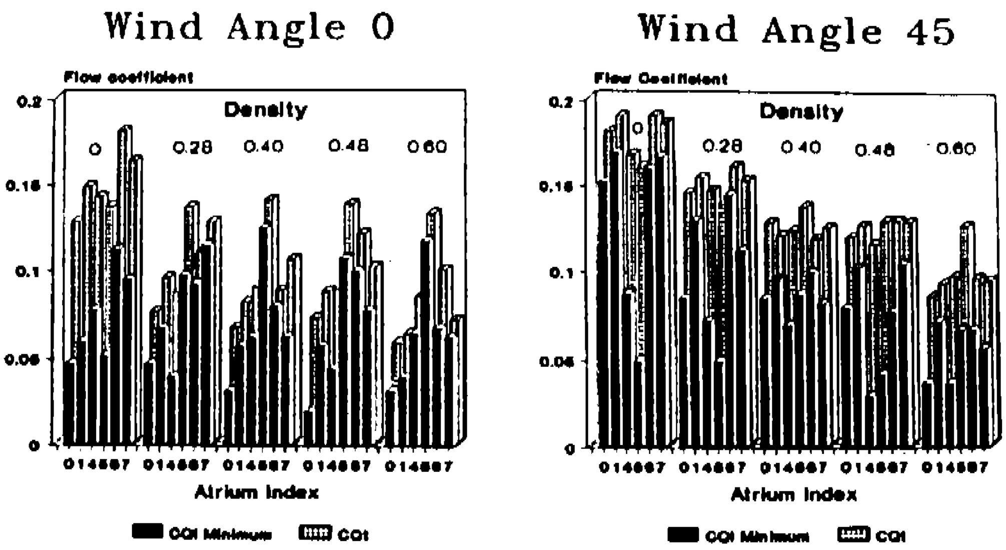

Figure 5 CQt and minimum CQl values (0 density is models in isolation)

CONCLUSION

This study has investigated the ventilation performance of courtyards and atrium buildings in isolation and in urban group layouts. An open courtyard in urban areas was found to have a weak ventilation performance, particularly when the courtyard was perpendicular to the oncoming wind. Covering the courtyard with a porous roof to form an atrium enables the large pressure fields on the roof to provide stronger ventilation pressure differentials. Roofs positioned to experience positive or near atmospheric pressure conditions performed less well than roofs exposed to negative pressure forces (suction) when winds are perpendicular to the buildings. At an oblique wind direction (45°) most of the atrium roofs performed to a similar standard. There are two major problems with using atrium roofs as ventilation devices in urban areas. Firstly, to use the weaker positive pressures requires large surface areas of roof and / or a great number of openings. Secondly, for the suction roofs the negative pressures on the leeward side of the building counteract the negative pressures on the roof. More efficient ventilation roof design may involve exploiting Venturi effects or vortex generation at roof leading edges where accelerated flows could be utilised.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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FILTERING AND HUMIDITY MEASUREMENT IN THE EXHAUST AIR OF BATHROOMS AND TOILETS WITHOUT WINDOWS

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Summary

The inadequate dissipation of humidity from living spaces and bathrooms has become a significant problem area in recent years. This can be attributed both to the replacement of old, poorly sealed windows by new windows with better seals, and to the increasing use of tiles and other building materials which hinder an adequate absorption of water vapour.

The residents tend to reject repeated opening of the windows for ventilation purposes on grounds of the ensuing energy costs. The result is the formation of mould both in the living rooms and in the bathrooms.

The installation of humidity controlled ventilation in bathrooms without windows involves the danger, that a drop below the dew point at the humidity sensor may lead to unwanted continuous operation. This, in turn, will result in a very short service life for the filters and excessive energy consumption.

Possible solutions are to be sought both in the construction of the building and in a controlled process of ventilation.

Introduction

The currently applicable Regulations on Thermal Insulation WSVO [5] are aimed at cutting the heat losses in and from buildings, in order to reduce the heating energy input necessary for their compensation (reduction of energy-related CO_2 emissions). Since the Regulations on Thermal Insulation have been in force, it has been possible to reduce the transmission heat losses to 1/4 of their original value, thanks to effective heat insulation of the outer shell of the building and methods of building with sealed joints. The heating consumption for ventilation has not changed significantly over the same period in absolute terms, though this does mean, that the proportion of heating energy consumption attributable to ventilation has increased from 20 % to 50 %.

The release of humidity within the rooms, on the other hand, has remained practically constant. Around 60 to 70 g of water are released per person per hour. Whereas persons and plants represent a practically constant humidity load over the period of the day, showers and cooking facilities are of importance above all as peak loads.

The following table summarises the humidity loads to be expected for the individual rooms of an apartment, together with the resultant minimum outside air flows.

Table 1 illustrates clearly both the heavy humidity loads of the bathrooms in use and a far from negligible humidity emanating from plants. The developments in building construction (water vapour absorption capability tending towards zero) mean that condensation on building elements can only be prevented by way of sufficient ventilation.

Table 1.: Humidity loads in apartments

Room	Typical range or mean value	Air temperature °C	Humidity produced g/h	Outside air flow m^3/h	Room volume m^3	Minimum ventilation h^{-1}
Living rooms	Range Mean value	20	100-300 200	25-70 45	40-80 60	0.3-1.8 0.8
Bedrooms	Range Mean value	16	20-100 60	5-30 20	20-40 30	0.1-1.5 0.8
Children's room	Range Mean value	20	90-200 150	20-45 35	20-60 40	0.3-2.3 0.8
Bathroom - in use - daily average	Range	24	700-2600 50-150	135-500 10-30	20-30	4.5-2.5 0.3-1.5
- in use - daily average	Mean value		1000 100	190 20	25	8 0.8
Kitchen - in use - daily average	Range	20	600-1500 20-180	150-350 5-40	20-40	3.8-1.8 0.1-2.0
- in use - daily average	Mean value		1000 100	230 25	30	8 0.8

Necessity of ventilation

The humidity loads occurring within the apartments can only be removed through ventilation. Two forms must be distinguished in this connection:

- Open-window ventilation
- Controlled ventilation

Open-window ventilation

Fig. 1 shows the distribution of the current ventilation situation in the eastern states of Germany (Saxony, Saxony-Anhalt, Thuringia, Brandenburg, Mecklenburg-West Pomerania). It is revealed clearly, that open-window ventilation dominates. A frequently found variation alongside open-window ventilation is natural air-shaft ventilation.

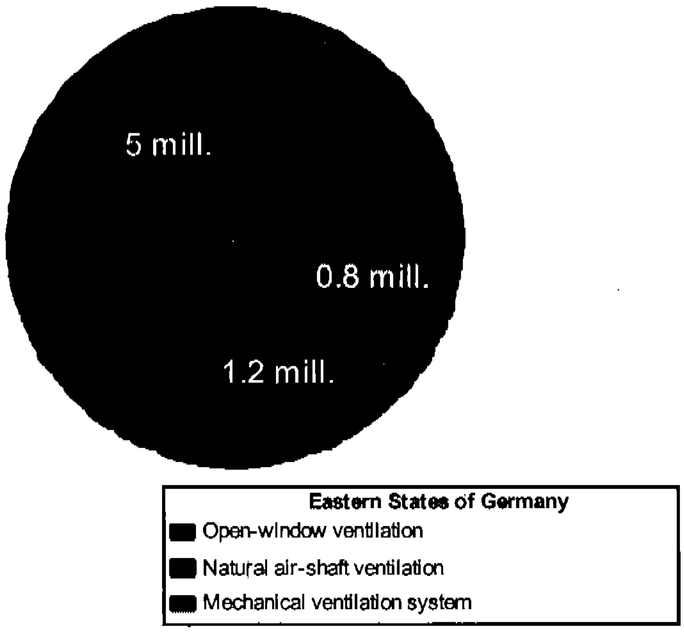

Fig. 1. Distribution of types of ventilation in the eastern states of Germany

In order to guarantee sufficient exchange of air with "non-continuous open-window ventilation", the authors have defined a minimum value, whereby the window should be opened for ventilation for 10 minutes every 2 hours. This value is still considerably below that laid down by Petzold [7] from a building climate point of view.

Open-window ventilation brings the disadvantages, that a scarcely quantifiable volume of outside air flows into the rooms, and that air-borne pollution, noise and possible draughts cannot be excluded. Open-window ventilation is furthermore dependent on the ventilation characteristics of the window.

Open-window ventilation leads to unwanted losses of heating energy, the amounts of which may reach considerable proportions. Fig. 2 shows the ratio of annual heating energy consumption for ventilation, documented in various periods of opening of the windows.

Even when applying the above-mentioned ventilation rule of "10 minutes ventilation every 2 hours", the heating energy consumption for ventilation is already some 30 to 40 kWh/m^2 a.

In other words, as Fig. 3 shows, almost half the annual total heating energy consumption in a modernised pre-fabricated apartment block (the typical housing solution in the eastern states of Germany) is used solely for ventilation.

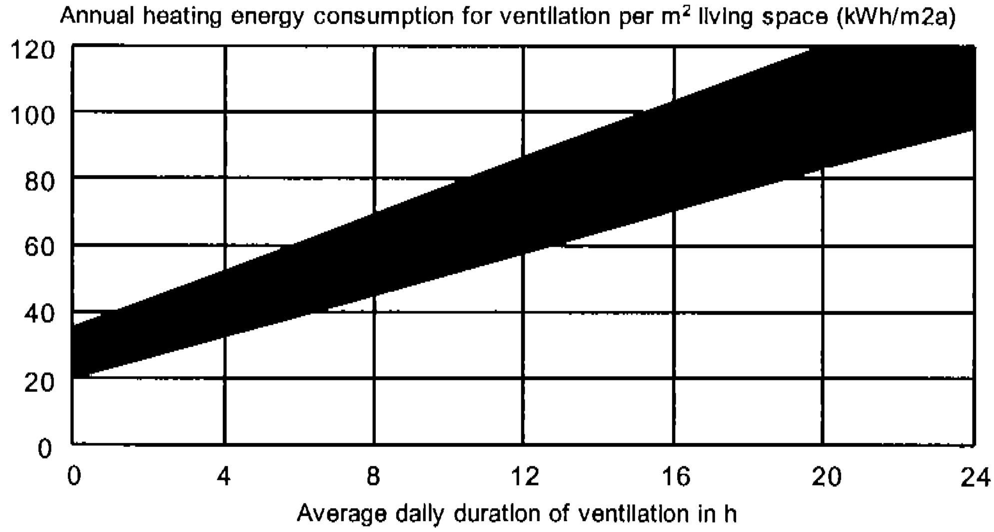

Fig. 2: Heating consumption for ventilation with tilted windows

In the case of permanent ventilation of the rooms, the share of ventilation heat rises significantly above that of transmission heat (compare Fig. 3), which in the final analysis constitutes a waste of energy.

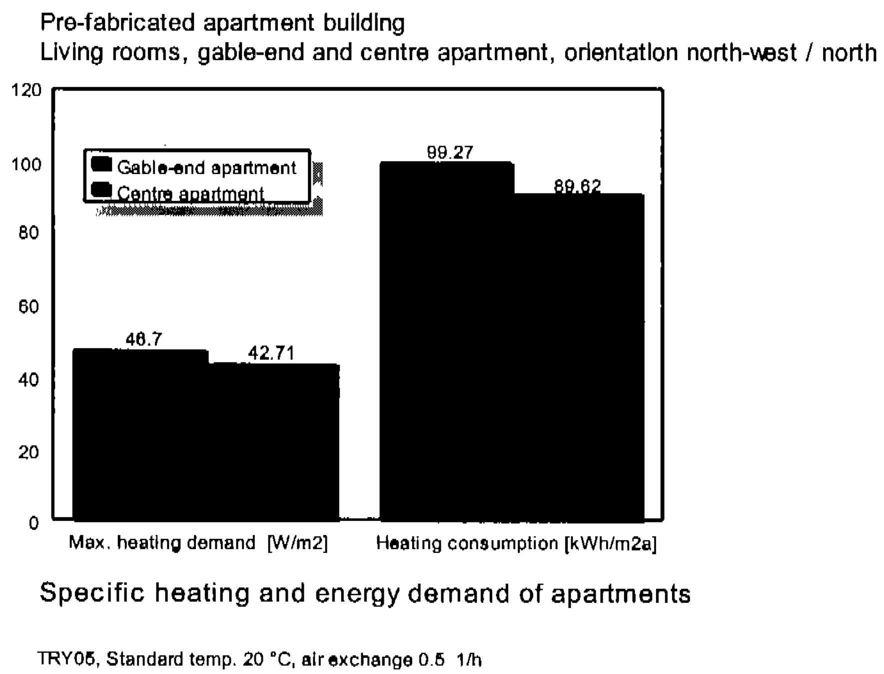

Fig. 3: Specific values of the heating and energy demand of the apartments

The ventilation habits of the residents have not changed in line with the new tighter sealing designs of both the window constructions and the building itself. As a result, negative effects arise, such as a deterioration of the air quality through emissions of pollutants or odours, and the formation of mould on the walls due to the inadequate dissipation of humidity, which are basically attributable to the objective of saving energy costs.

Occurrence of damage through housing improvement measures according to

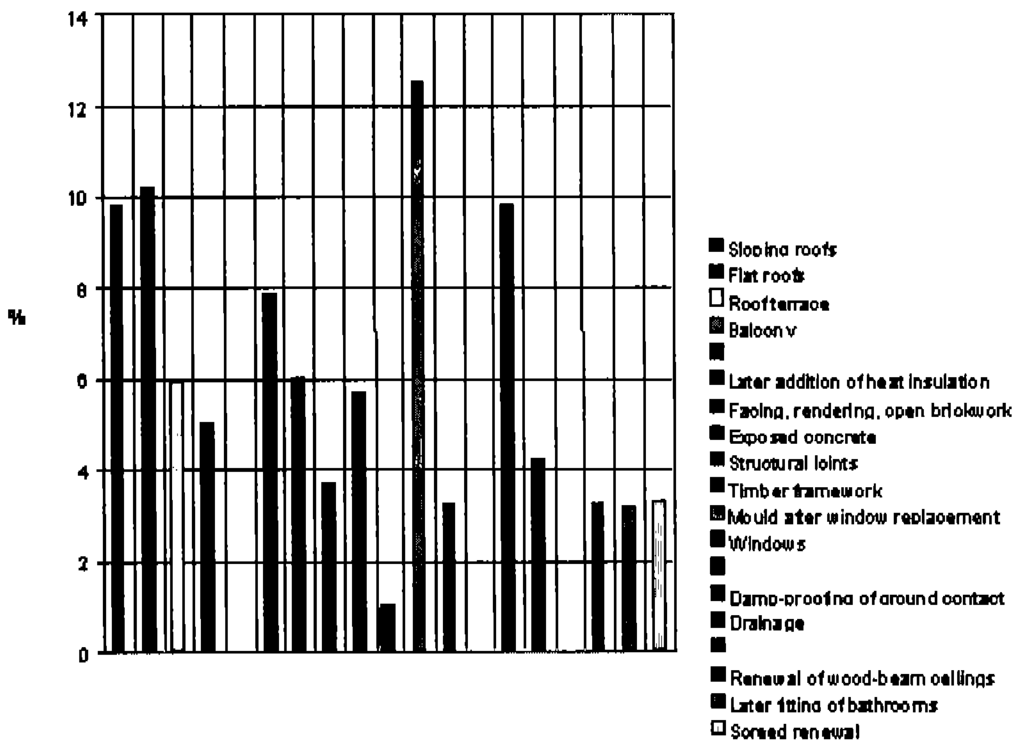

Fig. 4: Occurrence of damage through housing improvement measures from /1/

The share of damage following the replacement of old, poorly sealed windows by new windows with better seals is, according to /1/, over 12 %. This damage is manifested in the form of mould, which can most often be documented by the formation of black spots in inaccessible places. This means, that the necessity of ventilation in the form of open-window ventilation is generally not viewed by the residents under the aspect of humidity. A transition to some form of controlled room ventilation is becoming indispensable.

Controlled ventilation

In order to ensure a minimised, but at the same time adequate exchange of air, taking into consideration the factors hygiene, building physics and energy demand, it is necessary to install a mechanically controlled ventilation system. Such systems permit exact observance of a necessary minimum outside air exchange.

The housing ventilation systems currently available on the market can be divided into four main categories:

- decentralised single-room ventilation units, whereby a distinction is made between systems with and without heat recovery
- central ventilation units, with which a centrally located unit provides ventilation for all the rooms of an apartment or home via a system of air ducts. In this case, a distinction can be made between various technical solutions for heat recovery
- central ventilation units for apartment blocks, which must be viewed separately on account of their special characteristics, such as fire safety, hygiene, and acoustic aspects.
- ventilation units with integrated air heating

While the above systems were developed predominantly for normal living areas, inside bathrooms without windows must be ventilated mechanically, as laid down in DIN 18017 [6]. There is in this respect no direct correlation between DIN 18017 and DIN 1946 part 6. The outside air enters through the incomplete seals of the neighbouring rooms and flows through the bathroom via special ventilation openings. DIN 18017 specifies the following flow rates:

Table 2: Ventilation requirements according to DIN 18017

Room	Planned ventilation flow rate in m^3/h	
	duration of use ≥ 12 h/d	unlimited use
Bathroom also with WC	40	60

Mechanical bathroom ventilation in accordance with DIN 18017 introduces further aspects. Humidity controlled ventilators can and should permit good adjustment to the requirements of the user. The design principle is shown in Fig. 5.

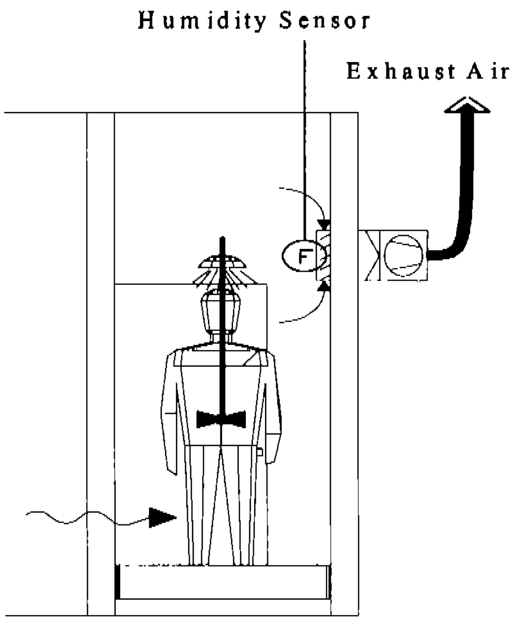

Fig. 5: Design principle of a humidity controlled bathroom ventilation system

The operation of such a system gives rise to the following problems:

- On account of the periodic operation, the piping and installed components (filter, humidity sensor) cool to below the room air temperature
- Water vapour is released suddenly and in large quantities when taking a shower or bath
- The electrical impulse causes the ventilator to start up with a time delay; in many cases there will be a risk of draughts due to inadequate heating
- The exhaust air laden with humidity is passed over the cooler elements of the system and condenses, e.g. on filters and humidity sensors
- The condensation of water at the humidity sensor produces a longer control signal and in turn longer operation of the ventilator

- Moisture saturation in the filter material causes dust contamination in the exhaust air to clog the filter, see Fig. 7
- Longer operation of the system leads to more rapid contamination of the filter
- Mould is formed on the filter material

Fig. 6 shows a new filter, Fig. 7 a contaminated humidity sensor and a contaminated filter.

These individual phenomena lead to an increased power consumption of the ventilators and to acoustic problems on account of the shifting of the working point of the ventilator.

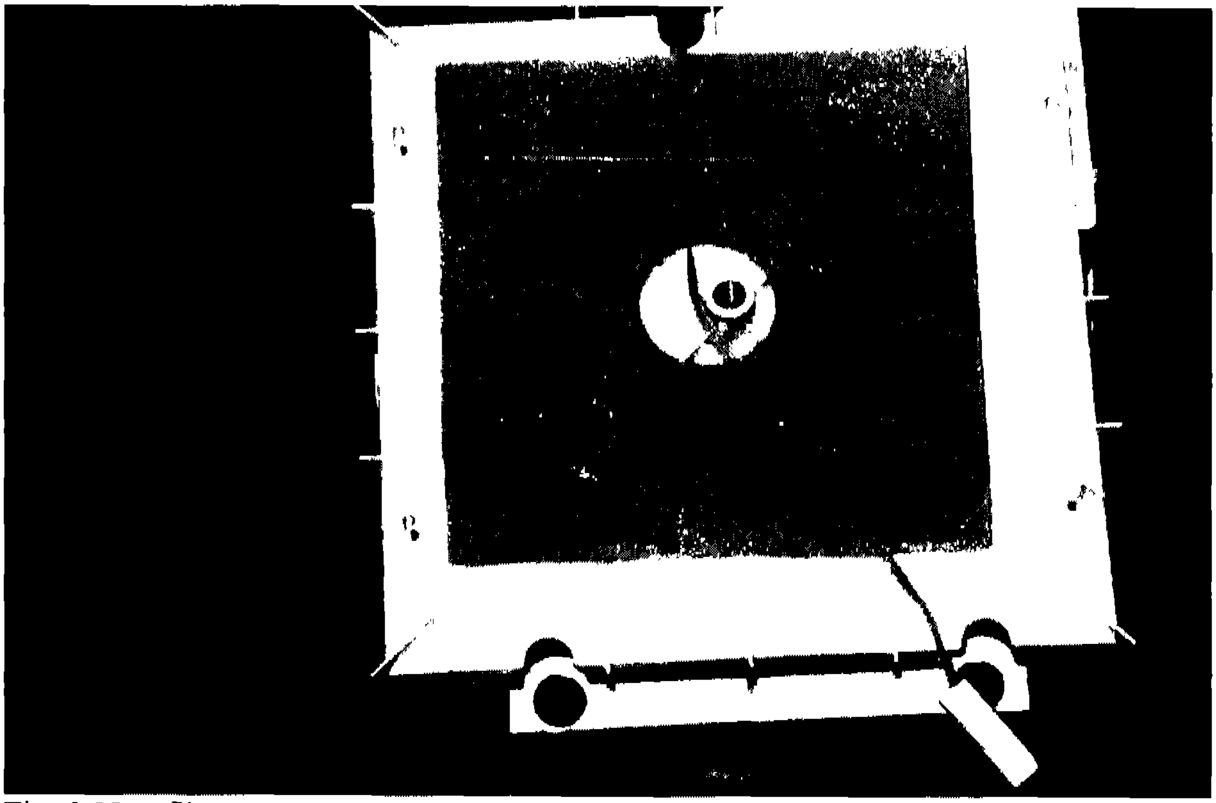

Fig. 6: New filter

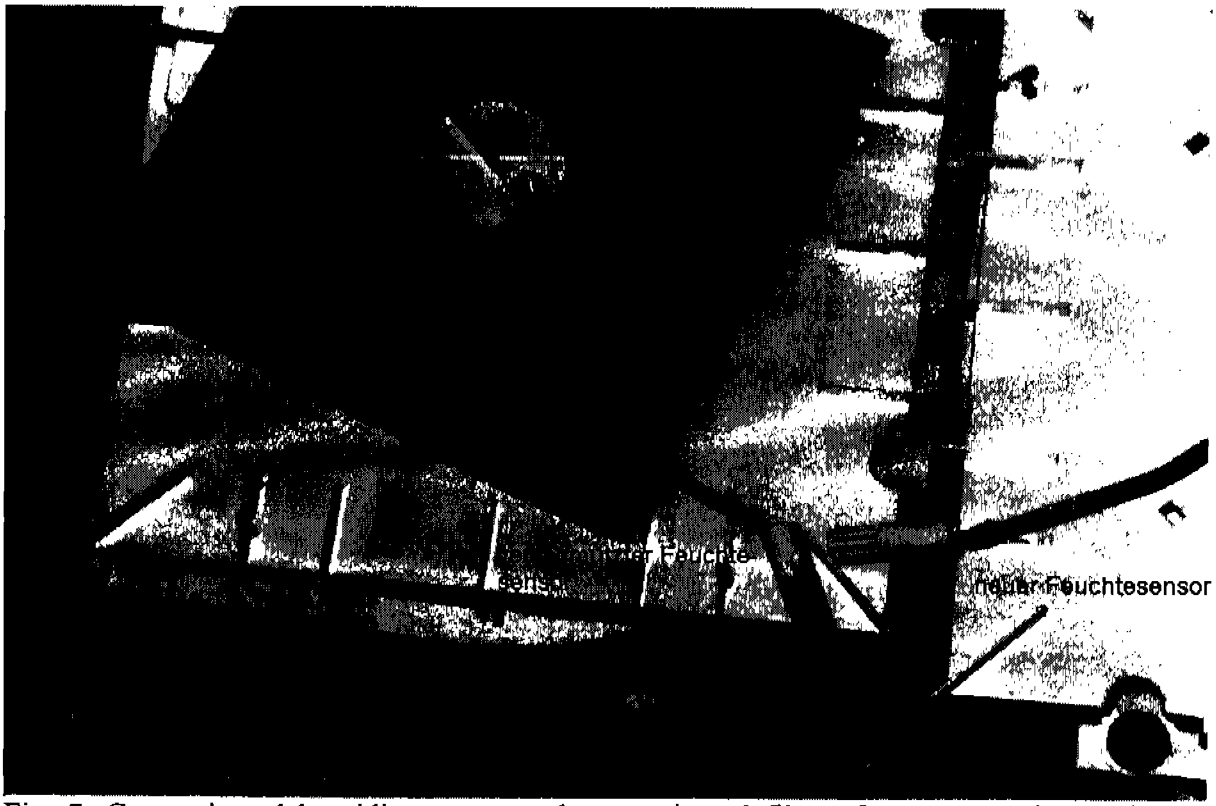

Fig. 7: Contaminated humidity sensor and contaminated filter after an operating period of approx. 1 month

The problem for humidity controlled bathroom ventilation is illustrated in Fig. 8. This diagram shows the development of the measuring signal of a capacitive humidity sensor over time. High-quality humidity sensors are generally characterised by efficient splash-water protection.

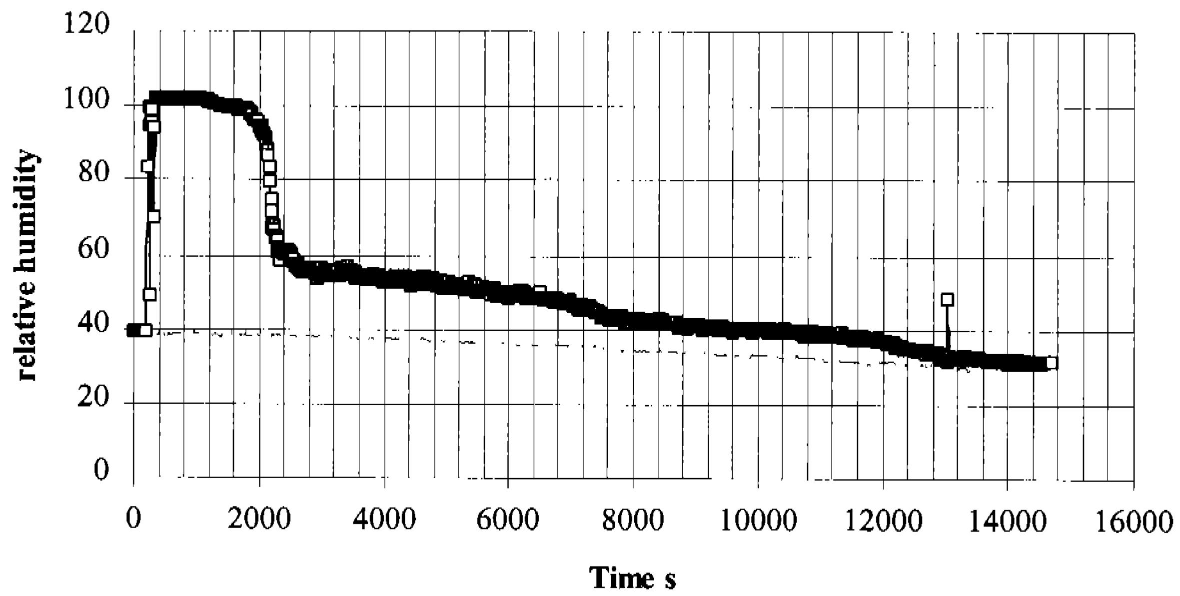

Fig. 8: Humidity measurement with capacitive humidity sensors over time

As Fig. 8 reveals, even direct splashing of the protective cover led only to a temporary signal increase, which was then followed by an immediate drop in the recorded relative humidity. When the protective cover was removed, however, the response was quite different. The water droplets in the sensor led to an exaggerated humidity signal over a period of approx. 30 minutes. Even after this time, the measured value differed noticeably from that of the properly protected humidity sensor for a further period of more than 3 hours. Any significant contamination would prolong this period even further.

Outlook and possible solution

Inadequate ventilation can lead to considerable condensation effects on the surrounding building structures, both in living rooms and in the bathroom. The use of mechanical bathroom ventilation may transfer this problem to the exhaust air filter - which is subject to mould growth due to continual dew-point condensation - or the subsequent duct system.

Solutions to the problem may be found in the following variations:

- suitable storage of humidity in the building structures
- temporary raising of the surface temperature of the surrounding building structures
- active sorption systems (condensation traps), which could be integrated into the bathroom

The humidity sensors should be arranged such that they are not in the permanent condensation area (utilisation of the waste heat of the ventilator). The regulation concept must also be revised, in order to prevent continuous operation.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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ATTENUATION OF CYLINDRICAL SILENCERS IN HVAC SYSTEMS

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Attenuation of cylindrical silencers in HVAC systems

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Synopsis

In this study, we have tested more than 80 silencers of different sizes (from Ø 250 to 1250), length, insulating thickness and with or without central pod.

The attenuation, measured at several velocities between 0 - 8 m/s, was compared to some literature estimations (Sabine, ASHRAE, ...) and has shown strong differences. They induce that literature estimations should be used only in the same conditions they were made and, not too widely, as it is currently done because real knowledge of the influence of parameters is lacking.

The influence of construction parameters such as insulation material, thickness and density, has been shown and correlations between attenuation and free area were found on the full range of silencers.

List of symbols

ΔL	global attenuation (dB)	
ΔLlin	global attenuation per meter (dB/m) equal to ΔL/	
P	perimeter of the cylindrical silencer (m)	
A	cross area of the cylindrical silencer (m^2)	
Anom	free cross area of the silencer with central pod (m^2)	
L	length of silencer (m)	
Φ	diameter (mm)	
F	pod diameter (mm)	
c	sound velocity (m/s)	in air $c = 20 \sqrt{T} = 342$ m/s (T = 293 K)
λ	wave length (m)	$\lambda = c / f$
fc	cut off frequency (Hz)	
α	Sabine absorption coefficient	

1. INTRODUCTION

In this study we have tested more than 80 silencers of different sizes :

- 42 silencers without pod (Ø 250 up to 1250 mm)
- 39 silencers with pod (Ø 315 up to 1250 mm)

according to EN ISO 7235 "Acoustics Measurement procedures for ducted silencers".

The lining was of different thickness (70 to 100 mm) depending of the silencer diameter.
The length of each silencer varied from one to twice the diameter.
The attenuation was tested for three air velocities in the silencer (0, 4 m/s, 8 m/s) but no real regenerated noise has been observed.

2. BIBLIOGRAPHIC STUDY

We did not find a lot of information about cylindrical silencers except the following ones :

2.1 Sabine formula

This formula is widely used for circular ducts :

$$\Delta L = 1{,}05 \,.\, \alpha^{1{,}4} \,.\, P / S \,.\, L = 4{,}2 \,.\, \alpha^{1{,}4} \,.\, L / \Phi$$

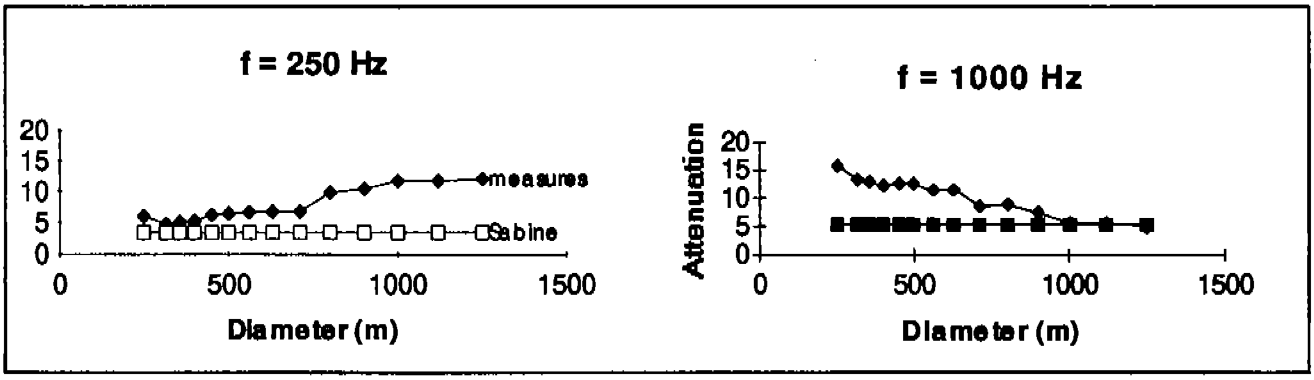

Figure 1

Figure 1 shows the comparison between measurements and Sabine estimation for 2 frequencies (250 and 1000 Hz).

In fact, some authors like Harris [1] restrict the application of Sabine formula for rectangular ducts to :

1- width of duct between one and twice the height

2- $0.2 < \alpha < 0.4$

3- $250 < f < 2000$ Hz

The absorption coefficient α of the lining of the tested silencers was over 0.65 for all frequencies from 250 Hz up to 4000 Hz. Therefore, our lining was not compatible to these limits.

2.2 ASHRAE formula

In ASHRAE ([2] and [3]), we can find a polynomial formula :

$$\Delta L = (A + B.e + C.e^2 + D.\Phi + E.\Phi^2 + F.\Phi^3). L$$

The formula was established for diameter of duct between Ø 150 mm and Ø 1500 mm, in fiberglass lining of density 12 kg/m^3 and thickness 25 to 75 mm.

Figure 2 shows that a strong difference can be observed mainly due to our lining density.

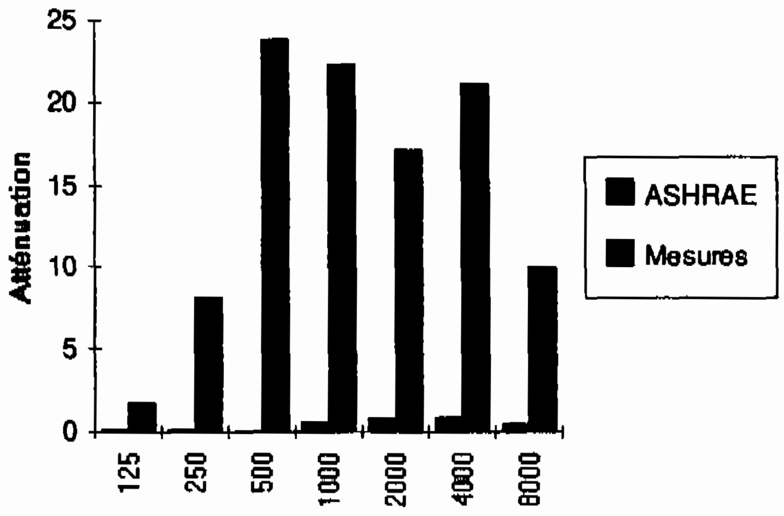

Figure 2: comparison between measurements and ASHRAE formula (D=500mm)

Actually, coefficients B and C of ASHRAE formula are equal to zero for frequencies over 1000 Hz. Therefore, the attenuation does not depend on the lining thickness for high frequencies ($f \geq 1000$ Hz). This has been verified in our tests.

2.3 Silencers with pods

Hischorm [4] indicates that the maximum of attenuation may vary of 2 octave-bands between Ø 305 and Ø 1524 mm.

Figures 3 and 4 show the comparison of his values with our measurements.

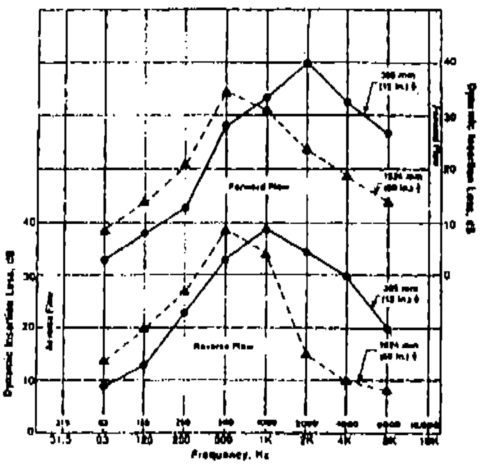

Figure 3: difference of insertion loss between ∅ 305 and ∅ 1524 mm (front velocity : 30 m/s) [4]

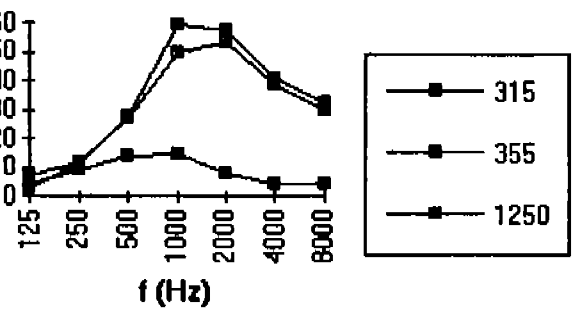

Figure 4: difference of insertion loss between ∅315, 355 and ∅ 1250 mm. Measurements CETIAT

For the measurements performed in CETIAT, this change of one octave band was only noticed between ∅ 315 and ∅ 355 mm. Then, the maximum stayed in the same octave band for all diameters.

3. TEST RESULTS

3.1 General comments

Attenuation appears mostly to be linear, except for smallest diameters (∅ 250 and ∅ 315 mm) which have stabilised around 25 dB. Therefore, in the following, we will consider :

$$\Delta Llin = \Delta L/L$$

when the lining thickness changes, we note discontinuities for low frequencies up to 500 Hz. From 1000 Hz and up, our tests confirm that the attenuation does not depend on the lining thickness.

The measurements did not show any change for frequencies over the cut-off frequency.

3.2 Regression

For each frequency, we have noted a regression :

$$\mathbf{Log\ (\Delta Llin) = k_1 - k_2\ Log\ (Anom)}$$

This regression is mainly valid in high frequencies but can be applied to the full range of frequency with a maximum error of 3 dB for silencer without pod and 5 dB with pod.

Coefficients k_1 and k_2 depend on :
- silencer geometry,
- lining thickness,
- frequency.

For silencers with pod, these coefficients change for 2 ranges of diameters :
- ∅ 355 - ∅ 900 mm
- ∅ 1000 - ∅ 1250 mm.

We couldn't explain the origin of this change since geometry and construction of these silencers did not appear to change between ∅ 900 mm and ∅ 1000 mm. This fact and the change of octave band, in which the maximum attenuation is measured, show that this regression must be used with caution for silencers with pod.

3.3 Influence of lining thickness and density

As the range of silencers we studied only contain 3 different lining thickness with 2 different density, we cannot make any regression.
Nevertheless, it appears that the attenuation is more sensible to density than to lining thickness.

4. CONCLUSION

We note that estimations of attenuation, given by the literature, currently used without any limit of application, should be taken with care and restricted to the exact conditions they were established for.
The attenuation of a cylindrical silencer is linked to the free cross-area, the density and thickness of its lining and its geometry.
The density appears to be much more sensible on attenuation than the thickness of lining. For silencers without pod, the attenuation does not depend on this thickness for high frequencies ($f \geq 1000$ Hz).

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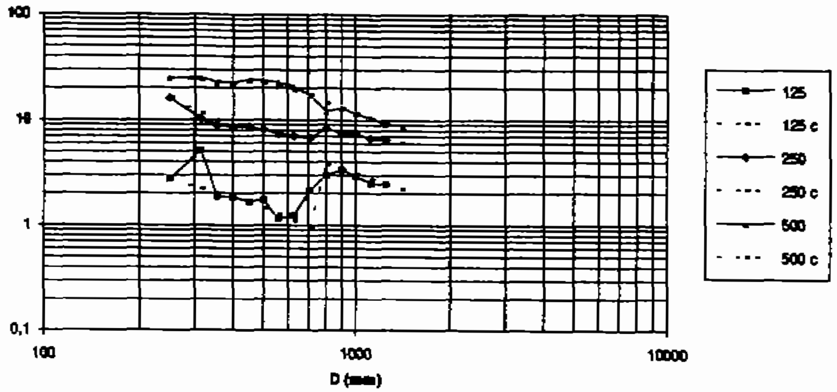
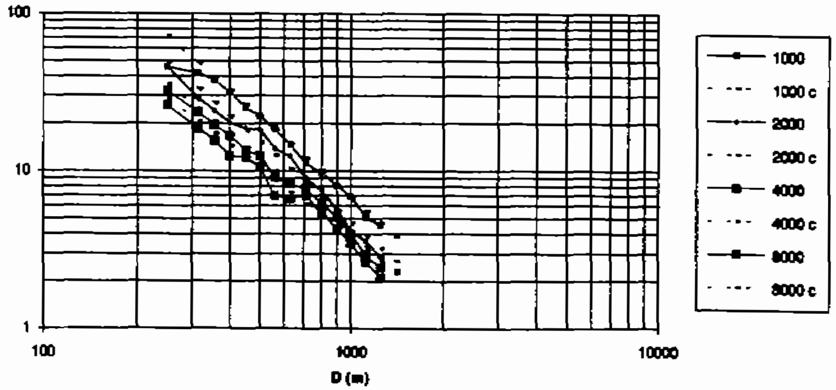
APPENDIX 1: Comparison regressions-measurements for silencers without pod

(125 - 500 Hz)



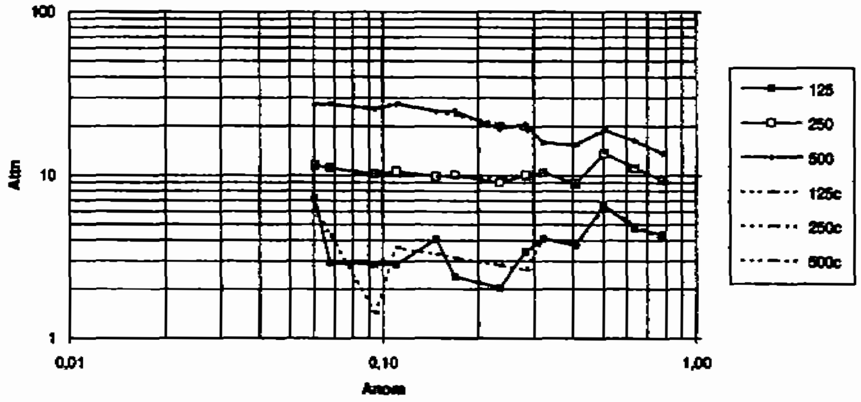
(1000 - 8000 Hz)



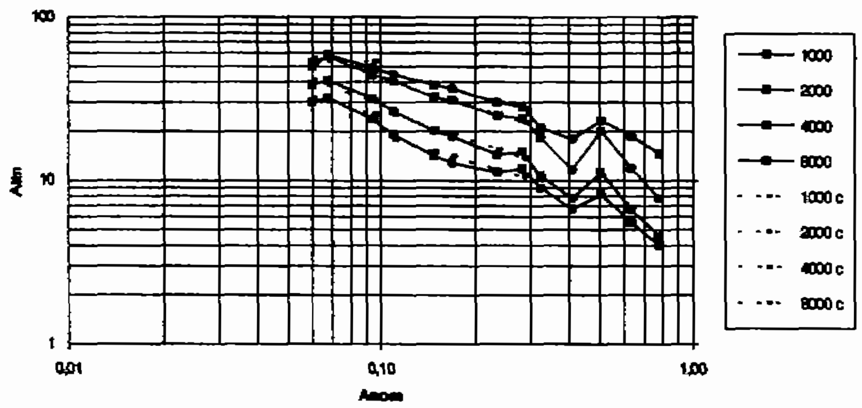
(*) c = calculated with regression

APPENDIX 2: comparison regressions-measurements for silencers with pod

(125 Hz - 500 Hz)



(1 000 Hz - 8 000 Hz)



VENTILATION TECHNOLOGIES IN URBAN AREAS

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EXPERIENCES FROM WALL EXHAUST SYSTEMS IN BLOCKS OF FLATS

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Synopsis

A self administrated questionnaire was mailed to over 300 dwellings in blocks of flats using the wall exhaust. In almost all the dwellings there was a controllable ventilation unit. The units were either a mechanical exhaust ventilation system type with outdoor air inlets or mechanical supply and exhaust ventilation system with heat recovery and outdoor air intake on the wall. In the questionnaire, the daily use of ventilation unit, noise levels as well as odors and their sources in the dwellings were asked. The prevalence of odors in the dwellings using wall exhaust was in the same level as in dwellings with traditional way to conduct exhaust air from dwellings to roof reported in earlier studies. Habitants in the dwellings with mechanical supply and exhaust ventilation system reported less draught, slightly less odor problems and judged the air quality more fresh even in the bedrooms in the mornings than habitants in dwellings with mechanical exhaust ventilation system only. Habitants in dwellings with mechanical supply and exhaust ventilation system were much more satisfied with the ventilation system and, then regarded their dwellings more comfortable than habitants in dwellings with mechanical exhaust ventilation only.

1.0 Introduction

The usual way of extracting the waste air in a multistorey residential building is to conduct the air above the roof. In older buildings exhaust ducts are made from concrete or brick. They are very untight and often in poor condition. A new exhaust air duct system to the roof should be built which is very expensive in an old building. There are also some specially constructed new buildings as terraced multistorey buildings with several penthouses. In such buildings it is difficult to find place for exhaust air fans without causing noise or odor in neighbourhoods. Furthermore, in a normal multistorey residential building a wall exhaust solution would facilitate the use of an apartment-specific mechanical ventilation system in multistorey buildings as the duct-work to the roof could be left out.

Wall exhaust is not allowed according to the Finnish building code (1). Local building authorities can permit it in single cases. At present there are more than 15 multistorey residential buildings with more than 500 dwellings with wall exhaust solution.

Design guidelines for wall exhaust has been developed in Finland (2). There are no minimum distances between air intake and exhaust openings if the exhaust air velocity is 8 m/s or higher. Otherwise the minimum distance should be at least 40 times the square root of the free area of exhaust air opening. For example, an exhaust duct with a diameter of 0.125 m requires a minimum distance of 4.5 m from the nearest outdoor air intake.

2.0 Research

2.1 Buildings

The purpose of this study was to examine whether exhaust air from each apartment could be discharged outdoors via exhaust vents mounted on the outer wall in such a way that no odors or other harmful effects are caused.

In this study, at first all dwellings in blocks of flats with wall exhaust system were charted. In summer 1996 there were 12 buildings with 360 flats, out of which 335 flats had wall exhaust and the rest an ordinary roof exhaust system. Half of the dwellings had a mechanical exhaust system and the other half mechanical supply and exhaust system. Almost all the ventilation systems could be controlled by the occupants.

Half of the dwellings were built in the 1960's or 1970's and have been renovated during the years 1995-1996. The rest of the dwellings were new, built between 1993 and 1995. The time of occupancy of the dwellings varied from two months to three years. One third of the dwellings were condominium apartments and the rest were rental apartments.

The typical exhaust discharge velocity with maximum air flow was 3-4 m/s. One third of the dwellings had a maximum exhaust discharge velocity more than 8 m/s. The placement of the wall exhaust had three different main solutions.

1) There were about 100 dwellings where the exhaust air was ducted when needed from the roof of the stairwell to the outer wall of the stairwell. This solution introduces the most largest distance between the exhaust air opening and the fresh air opening.

2) The most common solution was to conduct the exhaust air straight from the kitchen to the nearest outer wall. This solution was used in dwellings which mechanical exhaust ventilation only. If there are only two or three dwellings per floor, it is possible to place the exhaust air opening and the fresh air inlets with sufficient distance between each others. The more dwellings in the same floors the more difficult it is to place exhaust air openings and fresh air inlets without being placed too close to each others. This problem occurs especially with small dwellings. In Figure 1 is an example of such building but with mechanical supply and exhaust ventilation system.

3) In two buildings most of the exhaust air openings of dwellings were placed straight over the fresh air inlet. Also part of the dwellings had exhaust air opening less than half a meter from the outdoor inlet of next door dwelling.

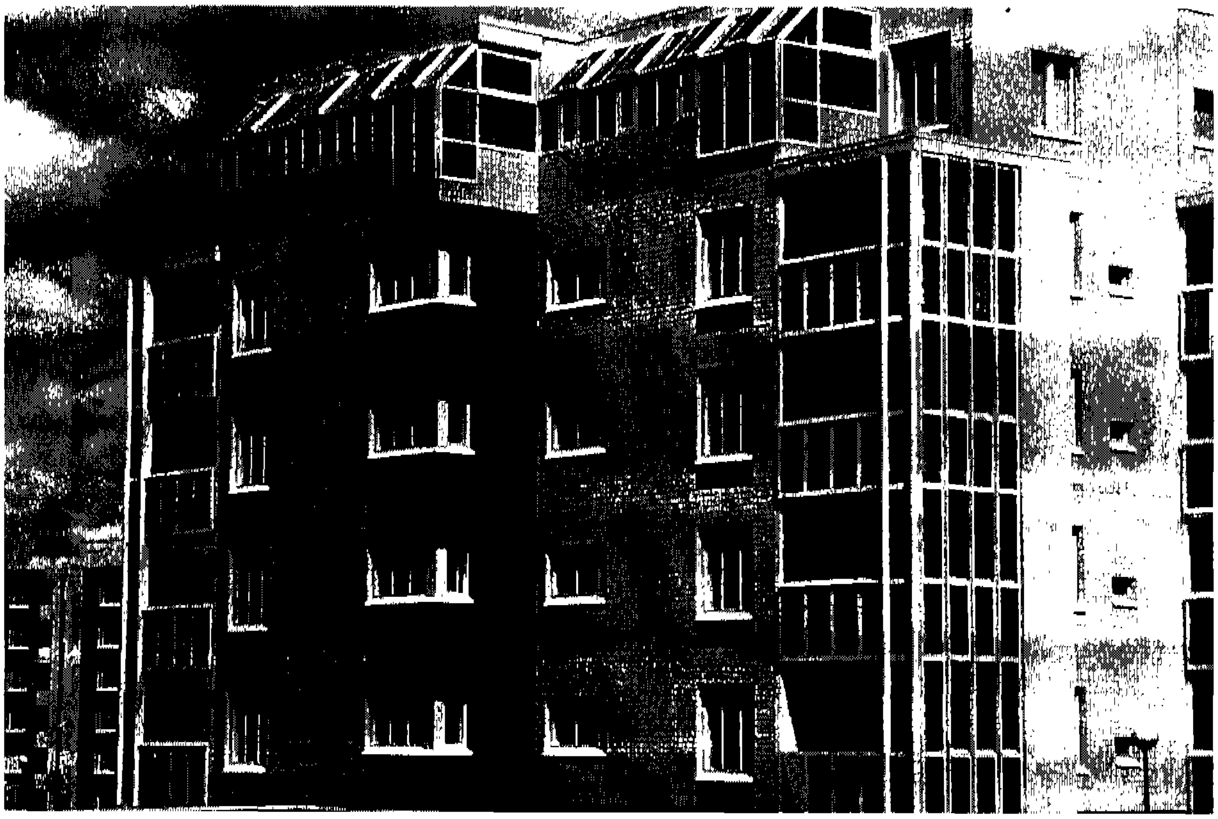

Figure 1. A block of flats with mechanical supply and exhaust ventilation unit in each flat and a wall exhaust solution. Outdoor air intakes and exhaust air openings are located in the same facade. Wall exhaust openings have longer shadows than outdoor air intakes.

2.2 Method

During this research, about 20 HVAC-engineers, janitors, building managers etc. involved with buildings having wall exhaust system were interwied. During these discussions no problems associated with wall exhaust occurred.

A self administrated questionnaire was mailed to all 352 dwellings. In the questionnaire, the daily use of ventilation unit, noise levels as well as odors and their sources in the dwellings were asked. The number of questionnaires returned was 200, which represents about 56 % response rate.

The questionnaires were recorded and analyzed with the SAS-computer package /3/. The prevalence of odors in different kind of wall exhaust solutions were tested with chi-square test. The same test was used for comparison of the prevalence of indoor air related problems with different ventilation system.

3.0 Results

3.1 Sufficiency of ventilation

According to 65 per cent of the occupants the ventilation of bedrooms and living room was sufficient during the heating season. The ventilation in them was also more often too high, especially in bedrooms, than too low. The ventilation of kitchen, bathroom and WC was more often too low than too high. No one judged the ventilation of sauna too high.

Occupants in dwellings with mechanical supply and exhaust ventilation system felt the ventilation of living and bedrooms more often sufficient than in dwellings with mechanical exhaust ventilation only. In those dwellings one third of the occupants felt ventilation of living and bedrooms too high during the heating season.

3.2 The use of ventilation units

The daily use of the ventilation units was relatively low, figure 2. The ventilation unit was never used in full speed in one eight of the dwellings. The median daily use of the full speed was 1.5 hours. The mean daily use of the full speed was 2 h and 15 minutes during heating season and 3 h and 15 minutes during summer.

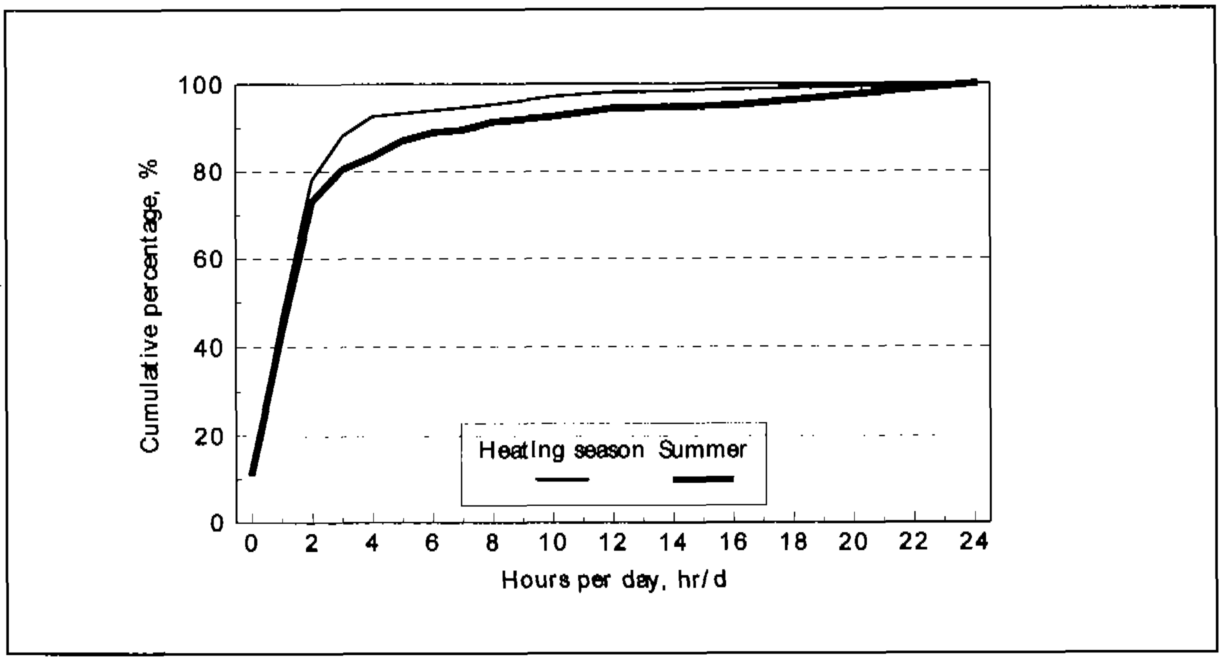

Figure 2. The distribution of the daily use of ventilation units in dwellings during heating season and summer.

About 80 per cent of the occupants did use the ventilation unit in minimum speed during the nights. Nearly 80 per cent used the ventilation unit in full speed during cooking. More than one third of the occupants used the ventilation unit in full speed during showers. The same number used the minimum speed. Even in the dwellings equipped with sauna one fifth of the occupants used the minimum speed during sauna baths.

3.3 Odors

Odor problems were named in 69 dwellings (35 %). This is quite a typical level occurring in Finnish dwellings. The most common types of odor were environmental tobacco smoke (ETS), cooking smell from the kitchen and exhaust gases from cars (Figure 3).

After cases where odors were from indoor sources, e.q. sewer gases and building materials, were excluded there were still 54 (27 %) dwellings with problems with outside odor sources. Most of those odors were cooking odor from staircase.

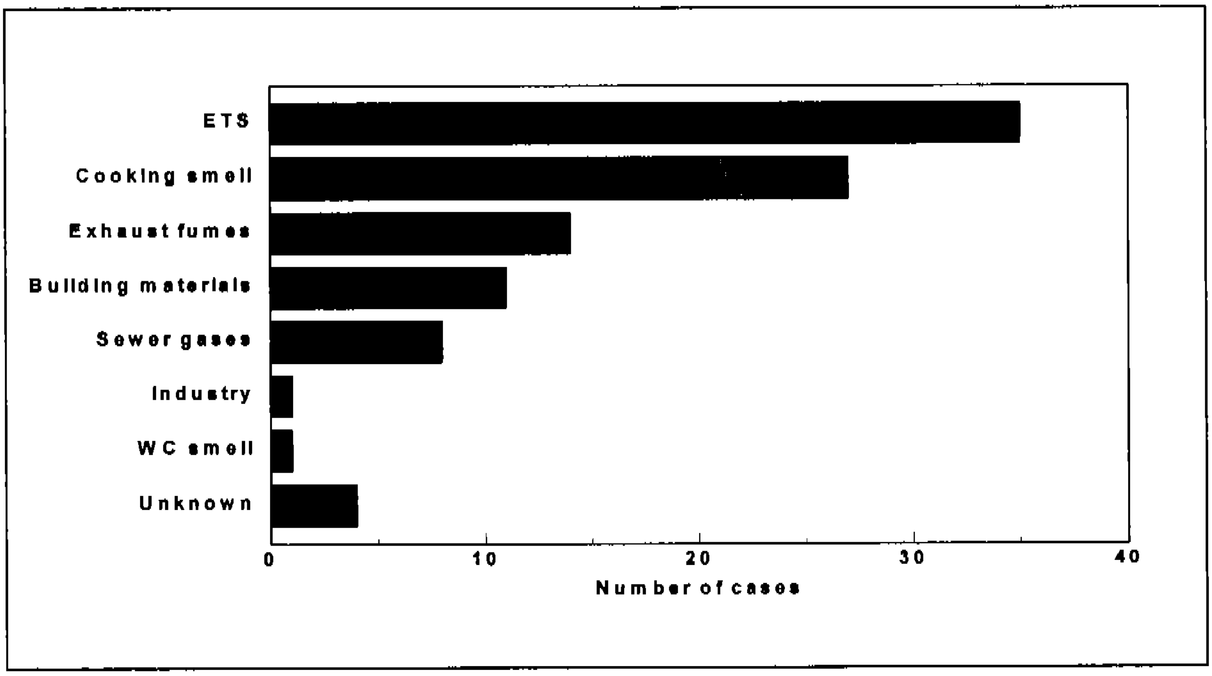

Figure 3. Odor types in dwellings.

Evident odors from outdoor occurred altogether in 32 dwellings (16 %). The most dominat sources were environmental tobacco smoke and car exhaust fumes. The main source of tobacco smoke was smoking on balconies in the vicinity of dwelling. There was only one case where the inhabitant proposed that the reason for cooking odor transfer was wall exhaust of the nearest neighbourhood.

It was not possible to find out any difference between different solutions concerning placement of the exhaust openings and air inlets and exhaust air design velocities.

The prevalence of odors was in the same level than in building with traditional way to conduct exhaust air from dwellings to roof /4/.

Dwellings with exhaust ventilation unit only had slightly higher prevalence of odors like cooking fumes and tobacco smoke from other dwellings or staircases than dwellings with supply and exhaust ventilation unit. The most likely reason for that is much higher under pressure between two dwellings, one dwelling with opened windows due to cooking or smoking and one dwelling with closed windows. Odors are transferred through cracks from polluted dwelling to another dwellings.

3.4 Noise

45 per cent of the occupants felt the noise level from ventilation unit disturbing with full speed and 10 per cent even with lower speed. Disturbance from ventilation noise was more common in dwellings with mechanical exhaust air unit only than in dwelling with mechanical exhaust and supply air unit. The main reason for that was the location of ventilation unit in dwellings. The exhaust air unit was placed over the cooker in the kitchen which often has straight contact with the living room. The supply and exhaust air units were placed in the bathrooms which were isolated from living and bedrooms.

There is a risk for too high noise level on the baloconies or in the yards when wall exhaust systems are used. This requires sufficient sound attenuator after exhaust fan. However, there were no complaints concerning too high noise levels outside caused by exhaust fans.

3.5 Draught

Almost 60 per cent of the dwellings were at least at times draughty. The main reasons for the draught sensation was too low room temperature and the windows.

More than 70 per cent of occupants in dwellings with exhaust ventilation system only reported of frequent or constant draught.

30 per cent of the occupants in dwellings with exhaust and supply ventilation system reported draught. 20 per cent of the occupants in dwellings with exhaust ventilation system and fresh air radiators reported draught. According to the occupants, The main reason for draught in these dwellings was too low room temperature, not ventilation system.

3.6 Air quality

Over 50 per cent of the occupants judged the air in their flats as fresh, 40 per cent as fresh or stale and less than 5 per cent as stale. The freshest air was reported in the dwellings with mechanical supply and exhaust air system (70 %) whereas the percentage among occupants living in dwellings with mechanical exhaust was 40 %. The air in the bedrooms was reported to be stale in the morning by 37 % of the occupants in the dwellings with supply and exhaust air ventilation and by 59 % of the occupants in the dwellings with exhaust air ventilation only.

There were no difference between dwellings with different ventilation system in airing habits.

3.7 Satisfaction and comfort of the occupants

65 per cent of the occupants were satisfied with the performance of ventilation system during the heating season and 75 per cent during the summer time. During the heating season, there was a great difference between occupants living in the dwellings with supply and exhaust ventilation and exhaust ventilation only. The percentages of satisfied with ventilation system was 83 and 48 %. During the summer time there was no difference between different systems.

60 per cent of the occupants said that their dwelling are comfortable during the heating season and 84 per cent during the summer time. During the heating season, only 43 per cent of the occupants in dwellings with mechanical exhaust only judged their dwellings as comfortable where as among the occupants in dwellings with mechanical supply and exhaust air ventilation system the percentage was 80 %. During summertime there was no difference between different systems.

4.0 Discussion

Results from these buildings with wall exhaust system did not show any things against it. It was not possible to find out any difference between different solutions concerning placement of the exhaust openings and air inlets and exhaust air design velocities.

Surprisingly, smoking on the balconies was the dominant odor problem in this study. Approximately 5 per cent of the inhabitants suffered greatly from the ETS from outdoor air. The location of fresh air intakes (supply and exhaust air unit) were in some buildings far too near the balcony of the neighbourhood.

The location of the parking lot was in some buildings far too near to the building itself. More attention should be paid to outdoor air inlet location in order to control more efficiently ETS and car exhaust.

In order to minimize odor problems in multistorey buildings, the tightness between dwellings must be improved, the pressure differences between dwellings should be decreased, more efficient local exhausts for cooking purposes should be developed and more tight doors between dwelling and staircases are needed.

Some of the occupants never use the full speed during cooking, showers and launder. The same problem has been found widely in Finnish flats with user controlled mechanical ventilation unit. More information must be given to occupants concerning the importance of ventilation.

Noise from ventilation units in full speed caused widely disturbance among occupants. Ventilation with lower noise levels are needed for night time cooling of dwellings. The placement of outdoor air intakes should be done more carefully; the fresh air inlets should not locate in the same facade with balconies.

This study included a few dwellings with fresh air radiators. These dwellings were judged as comfortable as dwellings with supply and exhaust air ventilation system. Of course much more research must be done to confirm this finding.

The feedback from the occupants showed quite clearly that in the Finnish climatic conditions ventilation systems where the outdoor air is preheated are much more preferred than older systems without preheated supply air.

Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

CONTROLLED NATURAL VENTILATION FOR COMMERCIAL AND INDUSTRIAL BUILDINGS

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TNO Building and Construction Research
Delft
THE NETHERLANDS

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Oslo, 1998

Controlled Natural Ventilation for Commercial and Industrial Buildings

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Abstract

The Dutch organization for applied scientific research TNO in Delft developed a system of Controlled Natural Ventilation (CNV). It is produced by the Dutch ventilation firm Brakel in Uden. The system controls ventilation grills and windows. Its purpose is:

- to compensate for fluctuating buoyancy forces (wind and temperature) so that natural ventilation flows are kept on set point value, independent of weather changes and changes in internal heat production;
- to optimize the air flow distribution over the building to get the highest possible ventilation efficiency;
- to restrict ventilation openings when draught risks occur.

The CNV system is based on a computer program that simulates ventilation. A special inverse version is derived that calculates the optimal ventilation openings for a specific building on each weather condition and for each ventilation set point.

The program needs input on local wind effects on the building. They are predicted with another new developed simulation tool, called the 'Cp-Generator'. This special computer program for prediction of wind pressure coefficients (Cp's) is build in as a module in the main program.

Extra features of the CNV system are:

- rain protection without decrease in flow rate,
- improved noise reduction,
- collaboration with mechanical ventilation,
- anticipation on opening doors,
- building leakage compensation,
- adjustment of both flow rate and direction to varying pollution or heat sources,
- smoother temperature control,
- special control for smoke ventilation.

Introduction

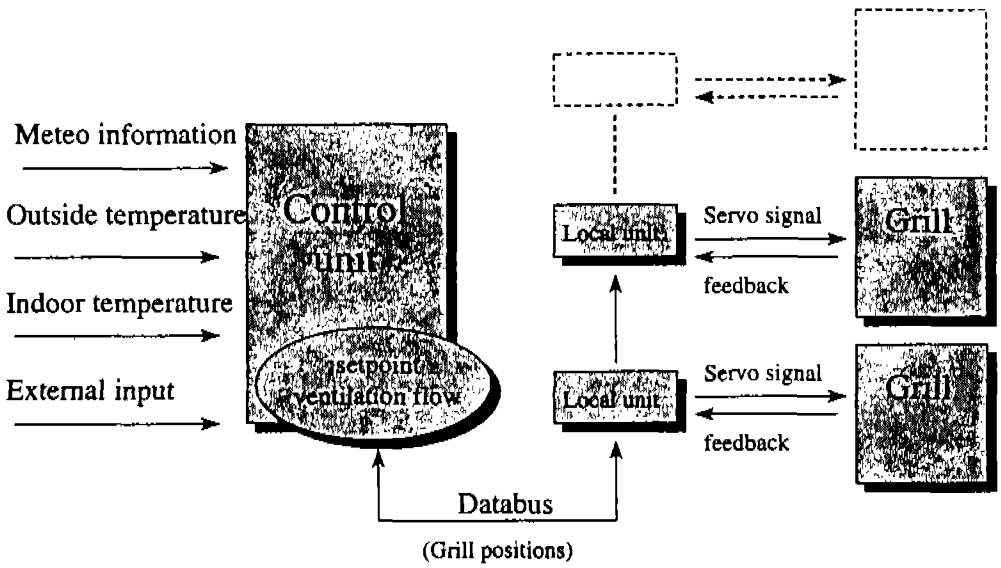
Natural ventilation is known as a cheap but poorly controlled system. In practice, a large variation of ventilation flows and directions occurs. As a consequence, users have to accept varying indoor air quality and draught.
To mitigate poor indoor air quality, in general natural ventilation shows an overshoot. Therefore, improving control also has an energy saving potential.
The main factor to be controlled is the wind. Combined with thermal buoyancy forces it causes the large 'unpredictable' fluctuations in ventilation.
To enable improved control, one large manufacturer recently came up with individual control per grill. However, this is just one of the necessary steps. The major steps are:

- to decide which position of each grill is the best in the present situation, depending on the users ventilation need;
- to automatically set each grill in its best position, however leaving the user in control over the ventilation set point.

The Dutch organization of Applied Scientific Research (TNO) in cooperation with the Dutch ventilation firm Brakel in Uden and its partner Helder engineering in Heeze developed this Controlled Natural Ventilation (CNV) system. Since late 1996 it is applied on a rapidly growing scale. The CNV system is described in this publication.

System Principle

The principle controller of the CNV system is a computerized ventilation simulation model, named 'TNO Regel'. It is a special version of the TNO ventilation model [1]. For a matrix of occurring situations the model 'TNO Regel' produces an output table with optimal grill positions. A control unit is fed with this data.



A local weather station measures the actual weather conditions (figure 1). Also the indoor temperature is measured. Together with the users ventilation set-point, it forms the input for a programmable logic controller (PLC). The input for this central control unit can also be generated by an external source.

The PLC looks up the occurring condition in the output table and calculates from the ventilation set point the optimal grill positions. The grill positions are send as an output signal to local grill controllers.
Each local controller recognizes its signal and uses it to set the correct grill position.

The ventilation model 'TNO Regel'

Basic calculation model

The calculation of ventilation is based on a number of equations. The physics behind them is explained using figure 2.
The wind causes an outdoor pressure distribution over openings in facades and the roof. The pressures over the height are influenced by thermal buoyancy. This will result in pressures P1, P2, P3, etc. compared to the indoor building pressure Pb. Each pressure difference over an opening causes a ventilation flow (q_v). The flow through an opening depends on its conductance (C) and the type of flow (n).

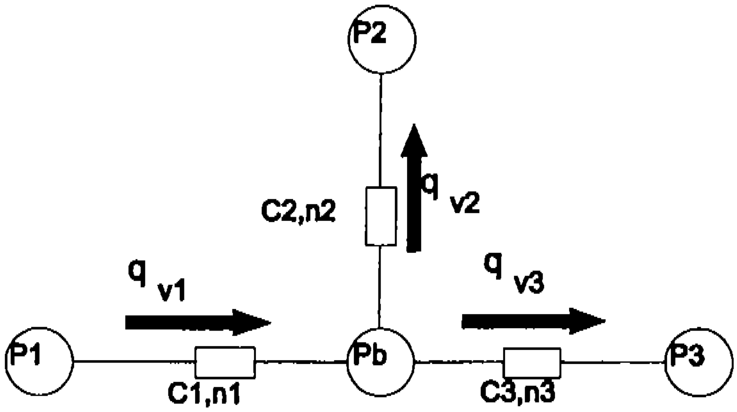


Figure 2 *Basic calculation model*

The flow per opening is described by:

$$q_{vx} = C_x \times (p_x - p_b)^n \quad \text{.......................................} \quad (1)$$

Also the sum of the incoming flows must equal the outgoing flows:

$$\Sigma\, q_v = 0 \quad \text{.......................................} \quad (2)$$

This means there are as many equations as unknown variables. Therefore, a solution can be found. The solver for this type of ventilation models often uses iterative calculation techniques.

Inverse calculation

Unlike the basic model, 'TNO Regel' is not a straight forward ventilation model that calculates ventilation flows from known openings. Its purpose is to calculate opening positions from a ventilation flow, set by the user.
One has to realize that this inverse calculation needs a calculation strategy. That is because there are numerous solutions to set the different ventilation openings, which all result in one and the same ventilation flow.
The basis of the calculation strategy is an optimization of the flow distribution over the building. The default is an evenly distributed flow, because this will minimize the occurrence of draught. However, the user may give his own preferences for the flow

distribution. If, for instance, a certain area of the building has a high contamination level, one defines nearby grills as exhaust openings and the other ones as supply openings, thus preventing the contamination to spread through the building.
This may rise the question whether and how it is possible to control ventilation flow directions in case of natural ventilation. It appears to be one of the benefits of CNV. To prevent cross flow according to the incidental wind direction, the windward openings are restricted and the roof openings where the lowest pressures (--) occur are enlarged (figure 3). This will lower the internal building pressure (-) below the leeward pressure (0), thus allowing supply at leeward side.

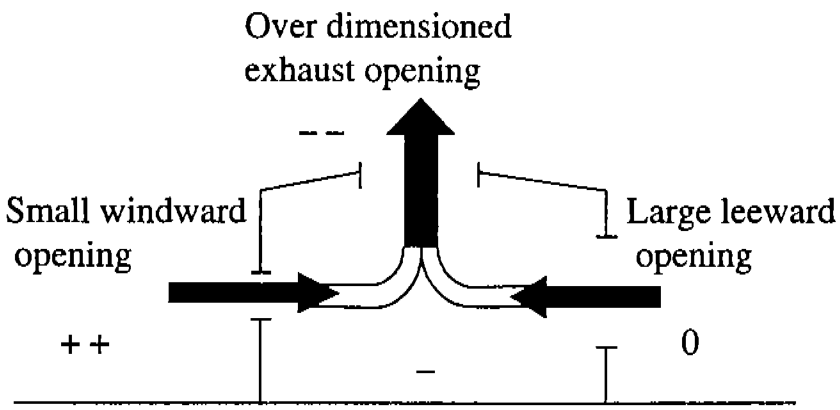


Figure 3 *Good ventilation design and control allows supply at leeward side*

The control of flow directions also needs a good design. The design is checked in the process of generating the output table. Apart from the optimal opening positions, a ventilation score is calculated. The score represents both whether the desired flow and flow distribution is met. In case of low ventilation scores the maximum grill openings and the grill locations have to be adjusted. A special add-on has been developed to guard the designer through this optimization process [2].

Input

The input for 'TNO Regel' is simple. For both the building and nearby obstacles the coordinates and wind orientations have to be given (figure 4). Also the locations of grills, doors, windows, leaks and other openings per facade and roof are transferred in a comparable manner. Each opening type is defined by its C and n-value. For some characteristic grill and window types

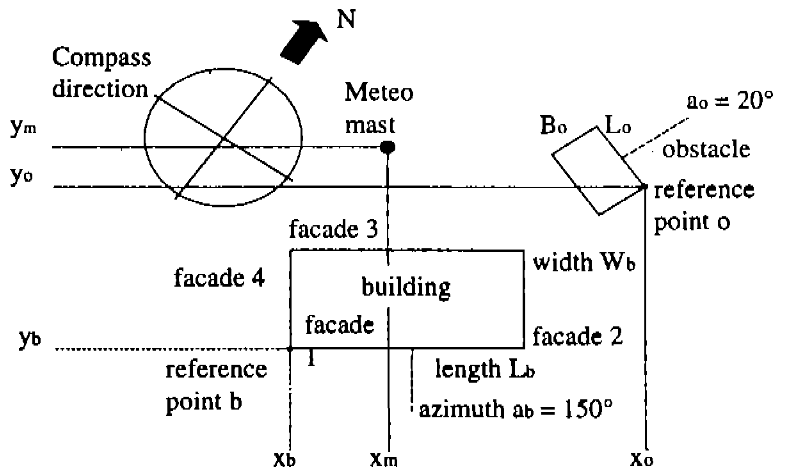


Figure 4 *The basic input consists of the main coordinates and measures of the building and surrounding obstacles*

the C and n-values are determined by measurements. Others are calculated from these measurement results and their geometry with an inaccuracy of about 10%.

Output table

For a matrix of weather conditions and flow settings the ventilation program 'TNO Regel' produces an output table with optimal grill positions and the ventilation score. Each output table is unique for a specific building with its surroundings and its grill positions and sizes.

The wind model 'Cp Generator'

A major input variable for the ventilation model is the wind pressure distribution over the building. The locally occurring wind pressure is presented as a factor to the dynamic pressure of the undisturbed wind, called Cp. In this way, the wind pressure is dimensionless.
Normally expensive wind tunnel measurement are necessary to accurately define the wind pressure coefficients (Cp's) for each opening at each wind direction. However, in this case a new computer program, called 'Cp Generator' is used [3].
The 'Cp Generator' is based on data of wind tunnel tests that were carried out systematically. The relationships derived from these data were programmed to calculate Cp's out of the simple input data mentioned before.
The process of Cp calculation is a fully background task of the program 'TNO Regel'.

Draught correction

A major disadvantage of traditional natural ventilation is the occurrence of draught. Draught problems can be highly reduced or even eliminated by:
1) preventing ventilation overflow;
2) optimizing flow distribution over the building (prevention of high local flows);
3) upward directing the incoming air by adjusting the positions of the grill strips;
4) decreasing incoming air flows at draught sensitive spots;
5) a higher positioning of ventilation openings in facades (increasing the mixing zone);
6) positioning air entrances in unoccupied zones;
7) preheating incoming air.

The CNV system takes care of a major draught reduction by performing the first four tasks. Tasks 1 and 2 are implicit tasks of the ventilation control system. Tasks 3 and 4 are extra. To perform these tasks, for each air entrance the curve of the incoming air is calculated. Using descriptive relations of air jets, the temperature and velocity at the upper limit of the occupied zone is calculated. The outcome is compared to draught criteria, to decide whether an additional draught correction is necessary. If so, the grill opening areas in case are decreased till the draught criteria are met. To compensate its effect on the ventilation flow, if possible, other grill opening areas where no draught risk occurs, are increased.

Known draught criteria are only valid for non-moving persons with rather low metabolic rates. Especially in industrial buildings, these conditions are exceeded. Hence, much higher draught limits are allowed. Therefore, the draught criteria used, are derived especially for this purpose (figure 5).

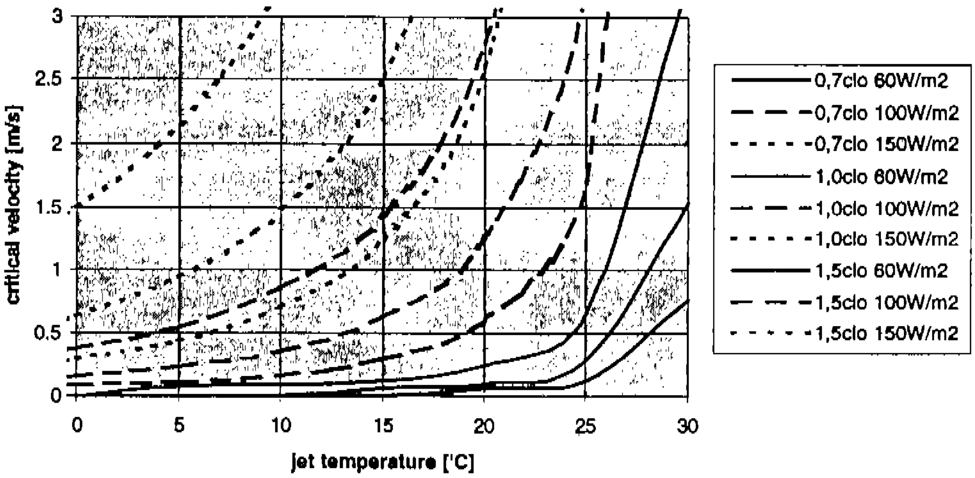


Figure 5 *Draught criteria for 30% dissatisfied*

The basis of the corrected draught criteria is a simple thermal exchange model of the skin, because skin sensors are responsible for the primary draught reaction. The velocity and temperature of the skins convective boundary layer of the neck is predicted from the metabolic rate and the clothing isolation on the one hand and the thermal environment (air temperature, air velocity and radiant temperature) on the other hand. Draught will occur when a critical skin temperature is descended.

Terrain roughness correction

A local weather station measures the actual weather conditions. The local wind pressures (p_{wind}) are calculated from the Cp-values, the wind velocity (v_{meteo}) measured on the local weather station and the specific mass of air (ρ), according to the formula:

$$p_{wind} = Cp \times \tfrac{1}{2} \rho \, v_{meteo}^2 \qquad \text{(3)}$$

The local wind velocity (v_{meteo}) will increase when the position of the local weather station is higher above the building and its surroundings. However, the wind pressure on a ventilation opening (p_{wind}) may not increase when a higher meteo mast is used. Therefore, 'TNO Regel' will compensate for this. It relates the calculated Cp-values to the position and height of the meteo mast.
The meteo correction on Cp's is based on a description of the velocity profile in the lower part of the boundary layer. Common expressions like the power law and Davenport's logarithmic relation (z_0 description) are not valid here. The descriptions are derived from velocity measurements in different terrain roughness' in the wind tunnel.

Description of major components

Local weather station

The local weather station has the following components:

- a cup anemometer to measure the local wind velocity, starting at 0.8 m/s;
- a rotating vane to measure the wind direction;
- a PT100 temperature meter, shielded for direct radiation;
- a conductive rain sensor.

The sensor signals are read electronically.

Indoor temperature sensors

The indoor temperature is measured at four spots in the room, using PT100 elements. Two spots are at 1 m above ground level, two spots are 1 m beneath the ceiling. The average value is used as indoor temperature in ventilation calculations. Also the temperature gradient over the height is used as a measure for the internal heat production.

The programmable logic controller AVC2080

A simple programmable logic controller (PLC) deals with the input of the weather station, the temperature sensors, the users input and any alternative external input, e.g. from a production capacity indicator, a rain detector, door position sensors, a smoke or fire detector and a fan use indicator.

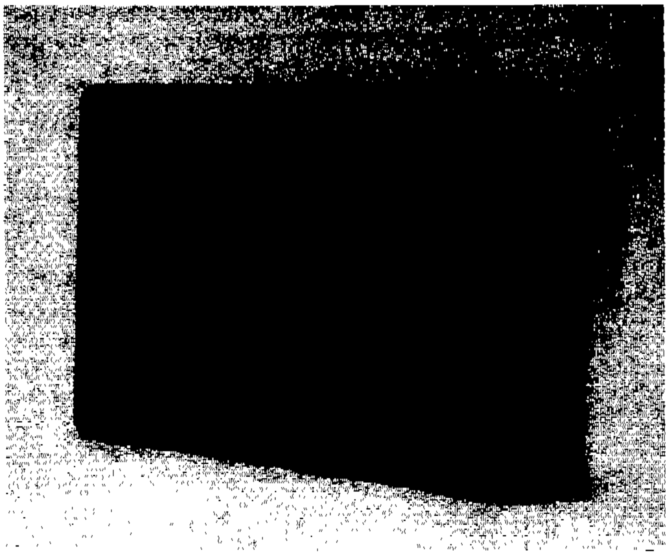


Figure 6 *The central control unit*

The user sets his ventilation requirement by turning a large knob on the central control unit (figure 6).
The user may also set his draught sensitivity, using small button tips. This will adjust the draught limit, used to restrict the grill openings in case of a draught risk.
The users settings are read on a display. Scrolling a menu with the button tips, the user is also able to show measured values on the display.

The PLC has some memory modules (E-proms), containing the output data from the 'TNO Regel' program. The data is specific for the case. There are no on-line ventilation calculations performed by the PLC. Major reasons for using this low intelligence local controller are cost reduction and software copy protection.
The PLC searches the memory for data nearest to the input information. For major variables interpolation is used. The grill positions for the occurring situation are the output.
The control speed is about once a minute.

The local grill controllers LU40

Each grill or group of grills is regulated by a local controller. Each local controller has its own code to recognize his information on the data bus. If the given grill position differs from the set position, a three-way pneumatic valve is activated to pressurize a two-way cylinder, allowing a positive or negative correction.
A potentiometer on the grill gives feedback over the actual position. If the desired position is reached, the pneumatic valve is set in its neutral position.
The local controllers make it possible to proportionally regulate the grills, still using cheap and fail-safe pneumatic adjusting devices.

System performance

Results of on-site measurements

To check the CNV system, an evaluation project is carried out [4]. The major actions are to measure ventilation flows and to deduce Cp-values from on-site measured pressure differences. This evaluation is still going on. Nevertheless, some first results can be shown already.
In general the ventilation flow ($q_{v\,real}$) still shows a considerable overshoot, compared to the set values (figure 7). The major cause is in the fast pressure fluctuations. This is shown by the continuous line in figure 7, showing the ventilation flow derived from the quarterly averaged measured pressure differences. If the ventilation model accounts for the fast fluctuations, a much better agreement with the set ventilation flow will occur.

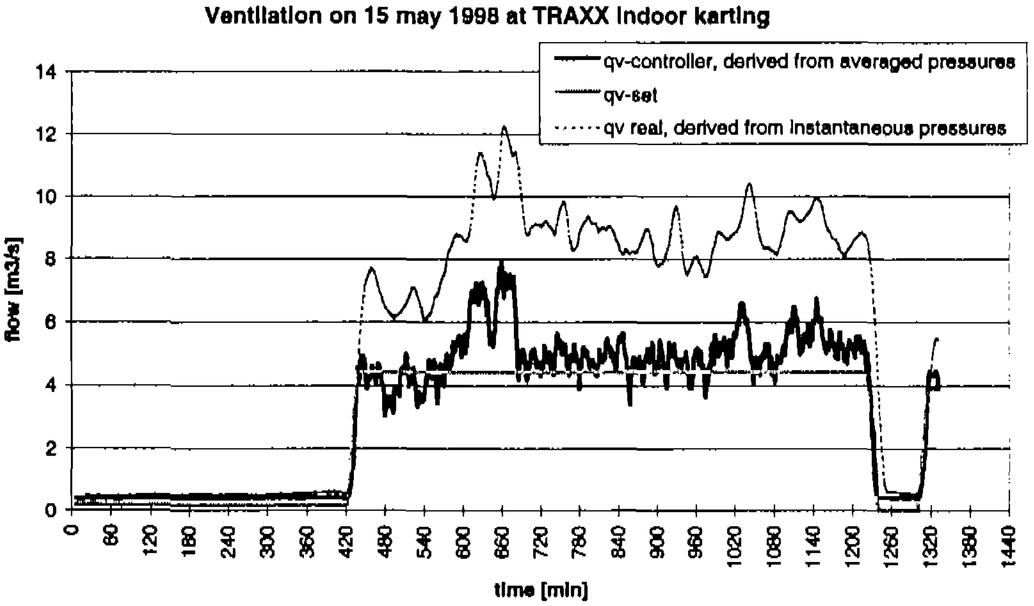


Figure 7 *Example of a typical evaluation result*

The large flow error due to pressure fluctuations is more or less caused by the control strategy, resulting in numerous large openings with pressure differences nearby zero. Figure 8 explains what happens if in this case the average pressure difference is taken in stead off the instantaneous one.

Accounting for a correction for the pressure fluctuations and some other minor errors, we consider the results as promising. The major goals, preventing large flow fluctuations and provide reproducible ventilation, appear to be fulfilled. The system, once fully developed, has the potential to keep ventilation flows constant within a 20% range.

Further analyses indicate that pre-calculated Cp-values, using the computer program 'Cp Generator' show a fairly good agreement with measured values (figure 9). Average Cp-values and global Cp-patterns do match reasonably. Only details are not predicted well.

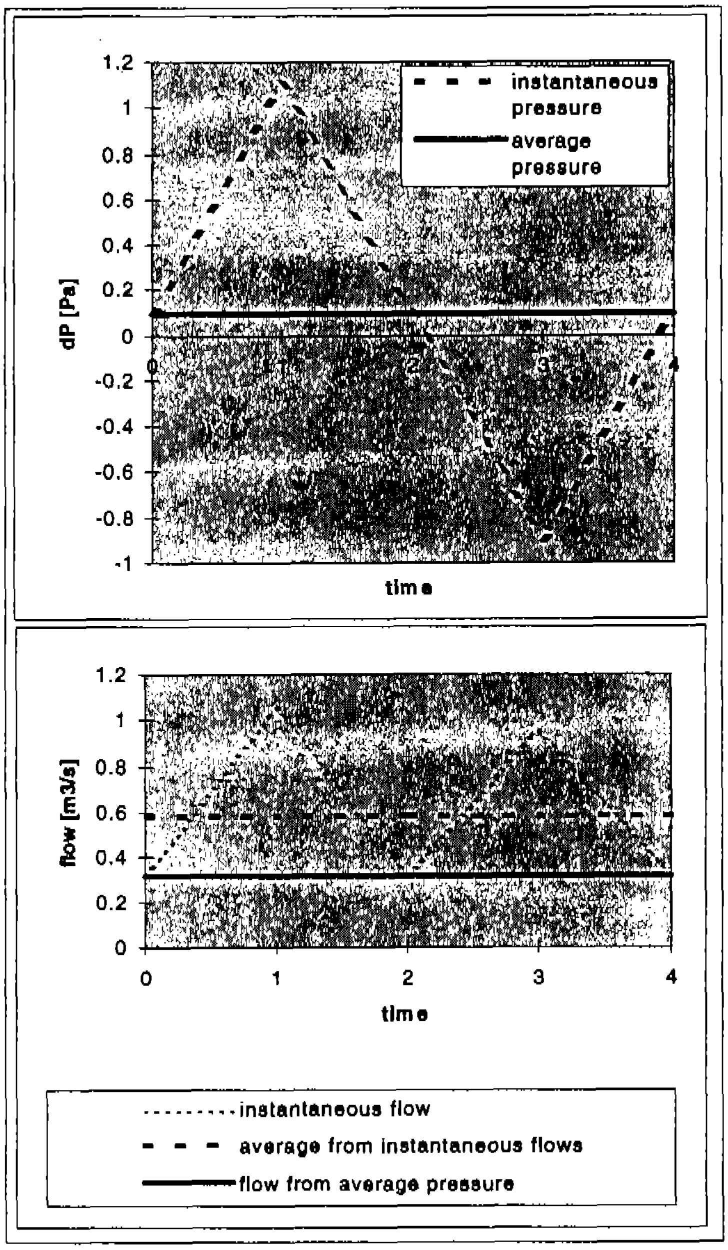


Figure 8 *Effect of pressure fluctuations on ventilation*

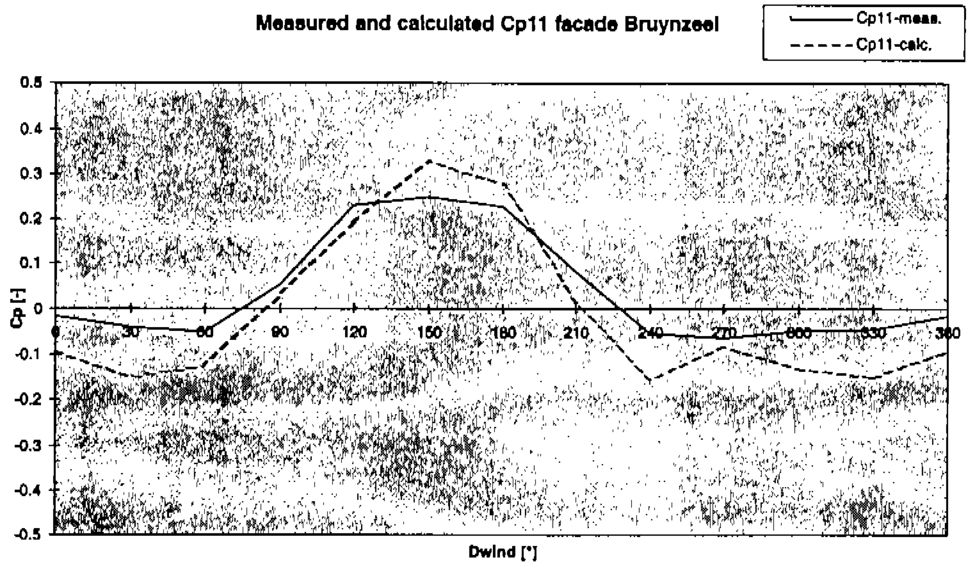


Figure 9 *Example of a comparison of measured and calculated Cp's*

User experiences

The users of CNV systems experience:

- better indoor air quality (IAQ);
- less draught;
- improved optimization between IAQ on the one hand and draught and energy losses on the other hand;
- mitigation of overheating problems.

An illustration of the improved optimization between IAQ and draught show the findings at an indoor cart track with a CNV system. This type of building is known as draughty or contaminated, but never ventilated well. CFD calculations for this case revealed that the mixing of ventilation air is improved significantly by application of a CNV system [5]. This is accomplished by the uniform distribution of ventilation air over the building exterior.
A further improvement of mixing is possible by the use of additional mixing fans, which are situated inside the building. User experiences confirmed these findings.

System costs

An important reason for the application of a natural ventilation system is its cost benefit compared to a simple mechanical ventilation system. One may wonder whether this still goes for the CNV system with its extras. To check this, the manufacturer did a cost analyses on actual project data. Figure 10 illustrates there is still an important saving on equipment costs of about 25%. The saving depends on the ventilation need (horizontal axis).
The cost comparison includes the provisions for heating the ventilation air and the room itself.
Costs are expressed in Dutch florins, which are about half a US dollar.

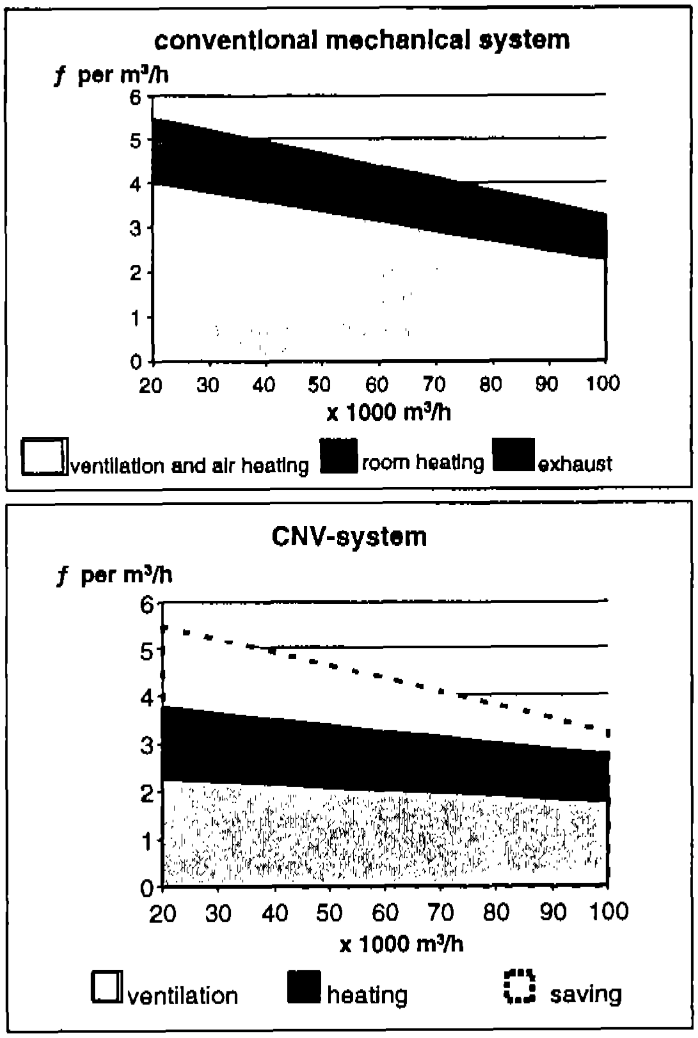


Figure 10 *Comparison of equipment costs for a simple conventional mechanical ventilation system and CNV*

Special applications

rain protection

One of the numerous special applications of CNV concerns rain protection. To prevent rain pouring in, grill openings are restricted. The restriction itself is not unique for CNV. However, compensating for the effect of the restriction on the ventilation flow is.
In case of rain especially roof openings will close. CNV will try and compensate this by increasing facade openings. The compensation is possible because natural ventilation systems are designed to function at minimal buoyancy forces. Often more advantageous conditions occur, so the ventilation goals can be realized even with a restricted number of openings.

noise reductive measures

Noise, whether coming in or going out, nowadays is a major problem. Additional dampers on the ventilation provisions may mitigate this problem. However standard dampers do highly restrict the air passage of the grills, so the number of (expensive) ventilation provisions increases to meet the ventilation need.
For this case the manufacturer of CNV offers a two way solution:

1. special developed low resistance dampers can be added on the grills;
2. CNV can be programmed with a special control regime, that squeezes grill openings at the noisy side as much as possible.

interaction with mechanical ventilation

In many cases some kind of (local) mechanical ventilation system may be present. The use of these fans will interfere with the CNV system. However, 'TNO Regel' is able to program CNV to work with these fans. In that case the fan use is given to the central control unit as external input.

anticipating on opening doors

Just as CNV may anticipate on fan use it may on the use of doors, if it is programmed for it. The door positions are send to the control unit.

building leakage compensation

Especially at low flow rates, interference by building leaks may cause CNV not to function properly. If building leaks are estimated in the design stage, CNV can be programmed to account for it.
Advantages of the leakage compensation are:

- minimizing uncontrollable infiltration, therefore minimizing energy loss;
- optimizing flow control, due to minimized cross flow.

automatic set-point adjustment

In the basic version of CNV the ventilation set point is adjusted manually. In many cases this will do, due to the reproducibility of the ventilation setting. This makes it possible to learn which ventilation set point is adequate in which situation.
In more complex situations however, an automatic adjustment of the ventilation set point may be preferred. Any external signal is suitable. Examples are temperature control, production dependent control or signals from a building control system.
The automatic setting can be overruled manually by the user.

local preheating

In case of temperature critical or draught sensitive situations, local preheating of the incoming air is advisable. The CNV system recognizes whether air intake or air exhaust will occur and at what rate. Therefore it is able to define how much preheating at which grill has to take place. The information is also used to close down at exhaust, thus preventing unnecessary energy loss.

area venting

Sources of contamination or heat may vary not only in size, but also in time and per spot. Depending on this, not only the ventilation flow set point of CNV may be adjusted dynamically, but also the major flow direction and flow distribution.
To accomplish this, first each grills contribution to the total exhaust and supply is redefined. Next, the output table for this alternative is calculated and loaded too in the central control unit. The output of different sensors over the place determine whether the basic flow regime or the alternative one has to be used.
A special type of area venting concerns smoke exhaust in case of fire. Depending on where the fire is, smoke may be directed different, better enabling people to evacuate. The advantage of CNV over a mechanical system in such cases is clear. It needs no extra distribution system, nor expensive extra components.

Conclusions

Controlled Natural Ventilation (CNV) is a system with all the benefits of natural ventilation, but without its major disadvantages. Moreover, the CNV system clearly shows some extra advantages, like a higher ventilation efficiency due to a more uniform distribution of ventilation air.

CNV prevents large flow fluctuations and provides reproducible ventilation. Once fully developed, it is expected that ventilation flows may be kept on set values within a 20% range.

The CNV system positions between conventional natural ventilation and mechanical ventilation. Due to its interesting possibilities, in a number of special cases it even may be qualified better than mechanical ventilation.
Nevertheless, CNV is about 25% cheaper than a simple mechanical ventilation system. Hence, it is expected that the application of natural ventilation may be widened by CNV.

The information that is produced standard in the design stage, allows a guaranteed performance of CNV systems.

Acknowledgement

The authors wish to thank their home fronts for lending them the extra time needed for the proper development of the CNV system and its parts.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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VENTILATION RELIABILITY – AN EVALUATION TOOL FOR DOMESTIC VENTILATION

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ABSTRACT

Ventilation reliability – An evaluation tool for domestic ventilation

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One of the objectives of IEA Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems" is to produce user-friendly tools suitable for pre-evaluation of different ventilation strategies for domestic buildings. Several tools, covering different aspects of domestic ventilation, have been developed in the project. One of these tools concerns ventilation reliability.

Ventilation reliability means – in general – the probability that the chosen ventilation system performs in an acceptable way for a certain building in a certain climate, between scheduled maintenance measures.

In the annex work, ventilation reliability has been treated in the following ways:

- Reliability as indicated by perceived indoor air quality, covered by the IAQ-tool
- Reliability as indicated by flow rate stability for a "perfect" system, but influenced by climatic conditions, air tightness of the envelope, window airing user patterns, air supply devices, local fans etc)
- Reliability as indicated by performance over time influenced by the quality of the ventilation system components and maintenance intensity.

This paper will summarise the results of the analyses of the last two items, ie. flow rate stability and performance over time. A simplified as well as an advanced tool for the pre-evaluation of ventilation strategies will be reported.

Please Note: The full paper will be published in the Supplement to the Proceedings

VENTILATION TECHNOLOGIES IN URBAN AREAS

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EXPERIMENTAL IMPACT VALUATION OF FOULING ON EXTRACT AIR TERMINAL DEVICES PERFORMANCES: AN ACCELERATED ARTIFICIAL FOULING APPROACH

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ABSTRACT

A humidity controlled air flow terminal device works as a humidity sensor : its opening surface varies according to relative humidity inside a room in order to match air flow rate to pollution. These components are fouling up when used during several months.

In a laboratory, an air flow with a high rate of particules is fouling up five identical air devices in a few hours. Considering a constant relative humidity, the impacts on two devices are similar : it seems that artificial fouling tests can be reproduced. These first tests show a high sensitivity to fouling at low relative humidities (small openings devices) and nearly no sensitivity at high relative humidities (large openings devices).

The characteristic curve of a humidity controlled air flow device represents the air flow rate in function of the relative humidity at a set pressure loss. The curves of four devices are compared before and after the accelerated fouling : the most important absolute air flow reduction is about 3.1 l/s. This maximum is always reached near the point of RH = 60 %. The corresponding relative reduction varies from 14 % to 23 %. The nearly same relative reductions are measured at the point of RH = 60 % with two years old fouled up devices from real dwellings although the fouling is visually very different. Thus, the accelerated artificial fouling experiments seem to be representative of real cases.

SYMBOLS AND UNITS

Q	Air flow rate (in l/s)
ΔQ	Air flow rate reduction (in l/s)
m	Total mass of generated particules (in g)
C	Particules concentration (in mg/m^3)
T	Temperature (in deg C)
ΔP	Pressure loss of a device (in Pa)
RH	Relative humidity

1- INTRODUCTION

As generally known, the extract air terminal device which is the only visible component of a mechanical ventilation system is clearly fouling up when used during several months. The impact is a possible deterioration of its performance. Considering real dwellings, the valuation of this impact is still possible. But the generalization of the results is difficult because some basic parameters are out of control, such as the users behaviour on a long time period or the local pollution.

This paper studies an experimental method of accelerated fouling of air terminal devices in a laboratory. The method is similar to the one used to test filters performances : an air flow with a high rate of particules is fouling up the component in a few hours. The performance of a clean component is compared to the performance of the same component once fouled up. The main purpose is to reproduce the same results under similar experimental conditions. Furthermore, these results are qualitatively compared to the performance of dirty components from real dwellings in order to check the representativeness. If these tests are successfull, this method would allow the comparative valuation of the air terminal devices performances with standard fouling conditions.

2- TYPE OF TESTED AIR TERMINAL DEVICES

Two types of extract air terminal devices are generally used in France. An automatic controlled air flow device keeps a nearly constant air flow rate whatever the pressure loss. A humidity controlled air flow device adjusts its opening surface to the relative humidity inside the room where the device is set up. Therefore, the air flow rate is an increasing function of relative humidity.

Humidity controlled air flow device is supposed to be more sensitive to fouling because its average opening surface is lower than the one of an automatic controlled air flow device. So, only humidity controlled air flow devices are tested here.

3- EXPERIMENTAL SET UP

Figure 1 represents the accelerated fouling experimental bench. The air flow contains a high concentration of particules. The ASHRAE dust is used. Its composition is given by the standard EN 779 [1] used to test filters performances : silica, carbon and fibrous cotton.

Because of the low air flow rates (about 5 to 20 l/s), the duct is set up on a vertical position in order to avoid the particules sedimentation.

The pressure difference between the both sides of the device is maintained constant at $\Delta P = 100$ Pa (± 2 Pa). Air relative humidity, RH, is controlled by a steam generator upstream the experimental bench. The relative humidity precision is about ± 5 %. The air temperature near the tested components varies between 19 and 23 deg C. On this temperature range, the devices performance is not dependent on air temperature. The air flow rate measurement is performed downstream the experimental bench with a propeller flowmeter.

Five identical air terminal devices are tested (called A, B, C, D and E).

4- REPRODUCTIBILITY TESTS

The method must be reproductible. For two identical air terminal devices tested with the same experimental conditions, the results must be similar.

Each air terminal device is fouling up on the experimental bench at a constant relative humidity. The tests are done with two relative humidity values : RH = 30 % and RH = 70 %. Figure 2 shows the air flow rate, Q, in function of the total mass of particules generated, m. The dust generator allows to control the particules mass flow. At each measurement point, the particules mass flow is adjusted taking into consideration the air flow rate in order to achieve a particules concentration C = 200 mg/m^3. Then, the mass flow is maintained constant until the next measurement. At this new point, the mass flow is adjusted again in order to have C = 200 mg/m^3. Therefore, between two measurement points, the particules voluminal concentration is increasing as the air flow tends to decrease due to the fouling.

Tests seem to be reproductible : at each relative humidity value, the curves shapes of two identical air devices are comparable. The fouling reduces the air flow rate at RH = 30 % but not at RH = 70 %. Thus, air terminal devices seem to be more sensitive to fouling when relative humidity (or their opening surface or the air flow rate) is low. Nevertheless, the curve shape of the B device is unexpected. The first three points are suspect due to possible experimental problems : perhaps, the device was still adjusted its opening surface to the relative humidity during the fouling and its steadyness was not achieved. Even so, the B device was fouled up and used for the next test (cf. section 6).

5- HUMIDITY CONTROLLED AIR FLOW DEVICES PERFORMANCES

The humidity controlled air flow device performances are represented by the characteristic curve which shows the air flow rate in function of the relative humidity at a constant air temperature (for our tests : T = 20 deg C).

The experimental bench used for these tests is set up according to the standard ISO 5219 [2]. The pressure difference between the both sides of the device is maintained at ΔP = 100 Pa. It takes more than one hour and a half to reach the steadyness of the relative humidity in the room calorimeter at each measurement point. The air flow measurement error is less than 0.15 l/s. The relative humidity error is less than 1.3 %.

First, these tests were done considering an increasing humidity and then considering a decreasing one. The goal is to see a possible hysteresis. Both the characteristic curves were so similar that we decided to present only the increasing humidity curve.

6- ACCELERATED FOULING TESTS

Figures 3 to 6 represent the characteristic curves of the A, B, C, and D air terminal devices before and after the accelerated fouling, and after the cleaning (the E device was only used in the course of the reproductibility tests). A sensible air flow reduction is noticed on the whole humidity range once devices are fouled up. The maximum absolute air flow reduction is about 3.1 l/s. This maximum is always reached near the point of RH = 60 % (table 1). At this point, the relative reduction varies from 14 % to 23 %. We also note that the fouled up devices curves are nearly linear whereas the cleaned up devices curves are hardly concav. Besides, the performances of an air device once cleaned up are nearly as good as they were when the device was new.

At RH = 30 % and RH = 70 %, we note unexpected differences between these characteristic curves and the results of the reproductibility tests (figure 2). A sufficient explanation was not found. The only one is about humidity during the accelerated fouling experiment which were performed with no normative requirements : a proper steadyness of the relative humidity and its stability were not sure. Besides, The relative humidity precision was not as good as it was during the devices performances tests.

Device	**Fouling conditions**		**Air flow rate reduction between a clean device and the same device once fouled up**		
	RH	**m (g)**	**Maximum ΔQ (l/s)**	**Corresponding RH**	**Corresponding ΔQ/Q**
A	70 %	31	3.3	58 %	19 %
B	70 %	23	2.3	60 %	14 %
C	30 %	10	3.2	60 %	19 %
D	30 %	12	3.6	60 %	23 %

Table 1 : Fouling conditions and air flow rate reduction between a clean device and the same device once fouled up

7- TESTS VALIDATION

The tests validation goal is to study the characteristic curves of fouled up air terminal devices used in real dwellings. We compare qualitatively these tests to the results of the accelerated fouling experiment.

Three devices called F, G and H which have been used for two years are taken from bathrooms of real dwellings. They are identical to the accelerated fouled up devices. Their fouling is fibrous and very different from the laboratory accelerated fouling which is finer. The characteristic curves of the F, G and H cleaned up devices cannot be compared to the ones of A, B, C and D new devices due to differences in initial calibration.

The figures 7 to 9 represent the characteristic curves of F, G and H devices when they are fouled up and again when they are cleaned up. Considering the three devices, the fouling impacts are similar although the three fouling aspects are not equivalent (table 2). The maximum air flow reduction is about 2.1 l/s. Contrary to the laboratory fouling tests, the reduction is nearly constant in the whole humidity range. At RH = 60 % (the laboratory fouling results show that the absolute air flow reduction achieves the maximum at this point), the relative air flow reduction varies from 15 % to 21 %. These relative reductions are comparable to the ones of the artificial fouling.

Considering all these components, the accelerated artificial fouling impact is similar to a real fouling impact. Nevertheless, the fouling aspects are visually different (fibrous fouling and finer one).

Device	Fouling visual diagnosis	Air flow rate reduction between the fouled up device and the same device once cleaned up		
		Maximum ΔQ (l/s)	Average ΔQ (l/s)	ΔQ/Q at RH = 60 %
F	Fibrous, high and homogeneous	2.1	1.7	19 %
G	Fibrous and low	2	1.6	15 %
H	Fibrous, high and confined (aggregates around the opening)	2.1	1.8	21 %

Table 2 : Fouling visual diagnosis and air flow rate reduction between the fouled up device and the same device once cleaned up

8- CONCLUSION

This study is a laboratory experimental approach of the impact valuation of the fouling on humidity controlled air flow devices performances. Considering the air terminal devices tested here, the representativeness is satisfactory. The reproductibility tests are also satisfactory but they need more control of the experimental conditions. This first work shows that the impact valuation of the fouling on air terminal devices performances is possible with standard (and possibly normative) experimental conditions. Nevertheless, we carrefully notice that only eight air terminal devices were tested here.

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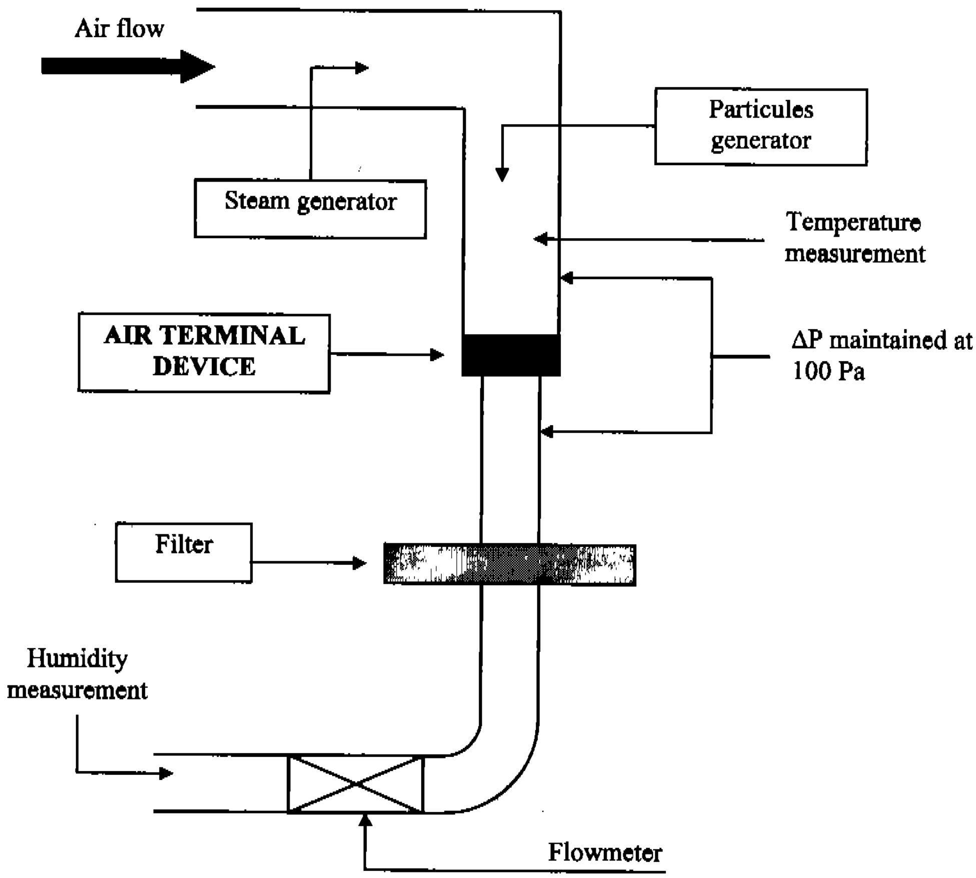


Figure 1 : Accelerated fouling experimental bench

Figure 2 : Reproductibility tests at ΔP=100 Pa and a constant relative humidity : air flow rate in function of the total mass of generated particules

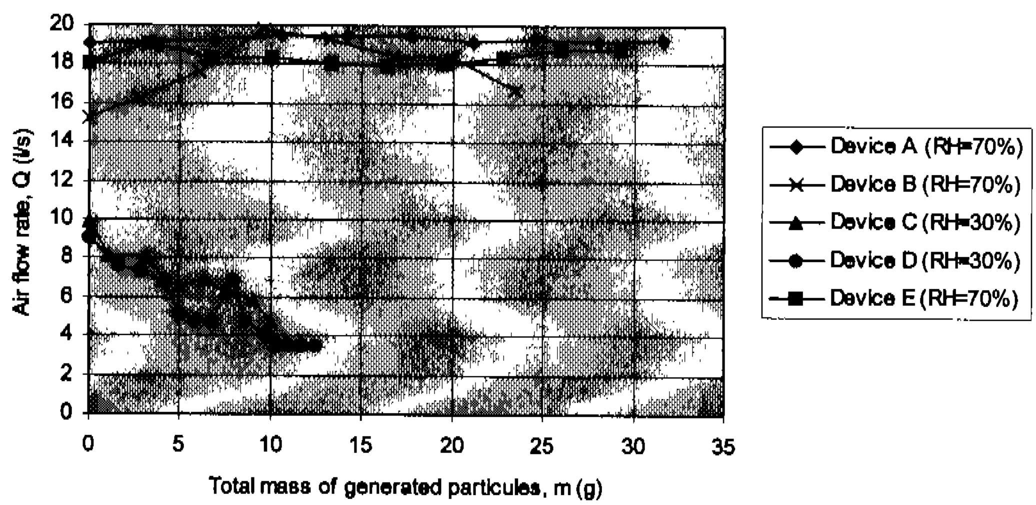


Figure 3 : Characteristic curves of the device A (ΔP=100 Pa) before and after the accelerated fouling

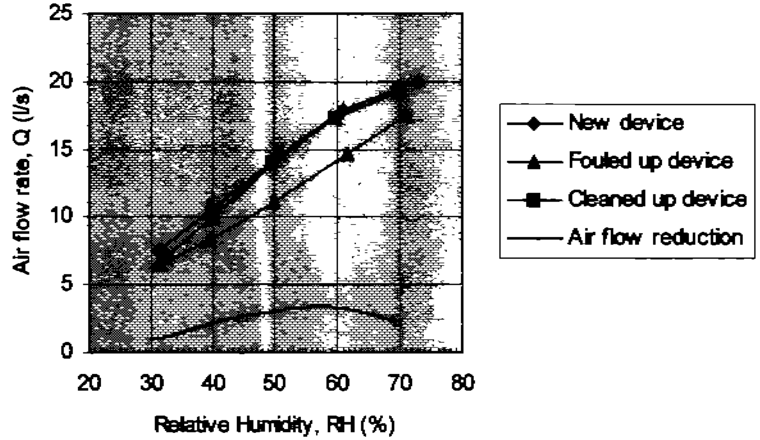


Figure 4 : Characteristic curves of the device B (ΔP=100 Pa) before and after the accelerated fouling

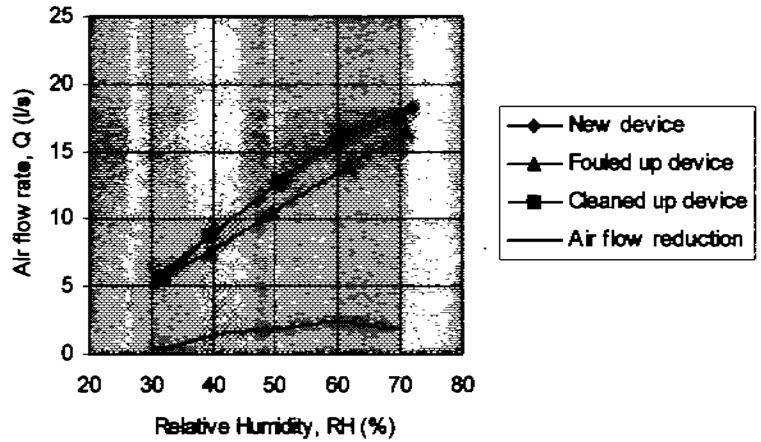


Figure 5 : Characteristic curves of the device C (ΔP=100 Pa) before and after the accelerated fouling

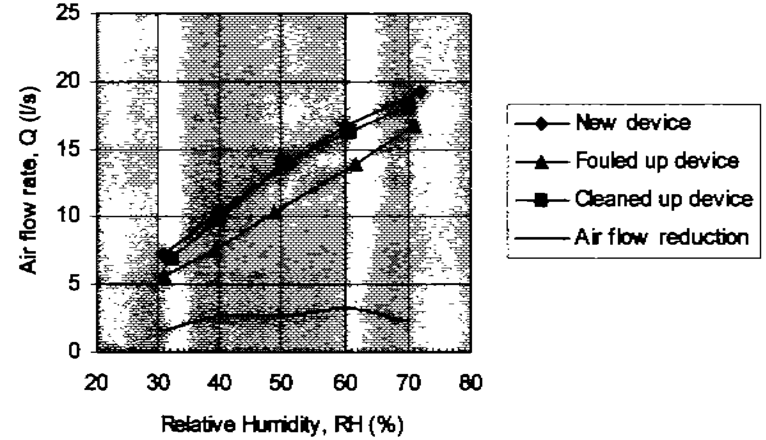


Figure 6 : Characteristic curves of the device D (ΔP=100 Pa) before and after the accelerated fouling

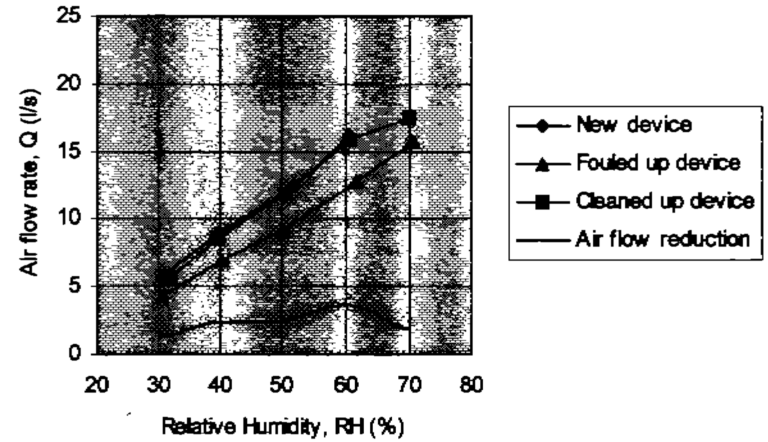


Figure 7 : Characteristic curves of the device F (ΔP=100 Pa) fouled up in a real dwelling and after the cleaning

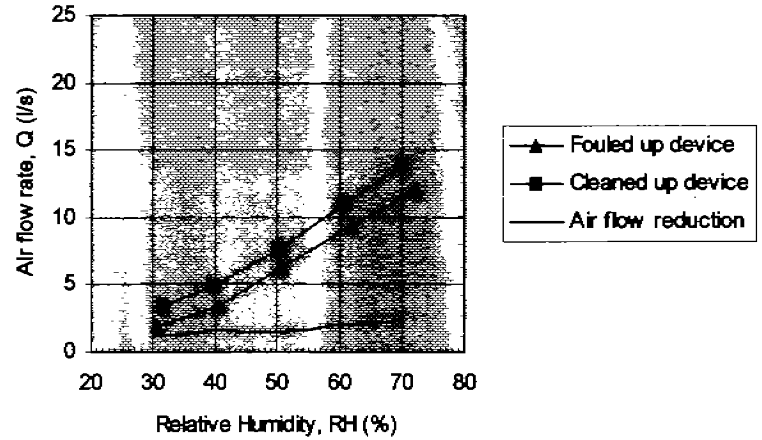


Figure 8 : Characteristic curves of the device G (ΔP=100 Pa) fouled up in a real dwelling and after the cleaning

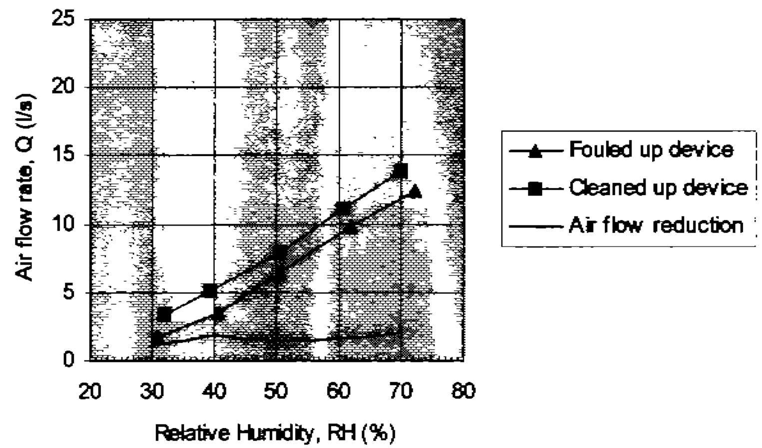
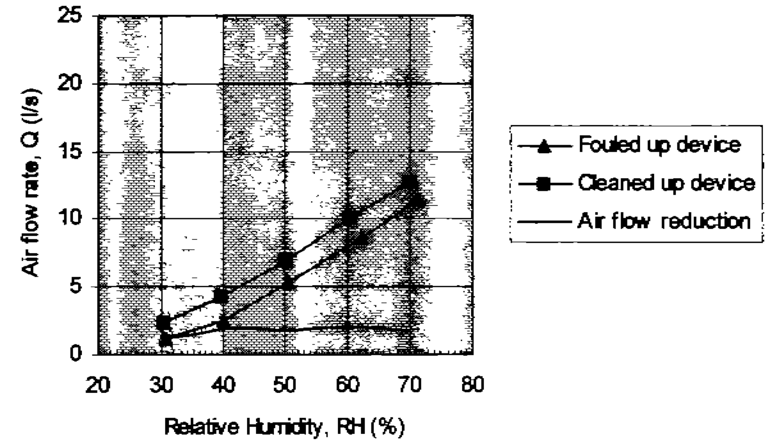


Figure 9 : Characteristic curves of the device H (ΔP=100 Pa) fouled up in a real dwelling and after the cleaning



VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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ANALYSIS OF DUCT SYSTEMS FOR VARIABLE VENTILATION FLOW RATES

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Analysis of duct systems for variable ventilation flow rates.

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SYNOPSIS

Ventilation systems using variable airflow are useful in urban areas. Due to outdoor pollution and the indoor load from pollution or thermal sources, it is important to vary the airflow. This must be done without disturbing the control of the total distribution. To analyse such problems, there is need for a design aid. This paper presents a first version of a modular simulation program working in the IDA environment. The program is based on a set of individual component-models in the NMF (Neutral Model Format) language. To illustrate the usefulness, a first series of simulations of a supply air system with the main characteristics of a real office building, using two different control strategies is presented. To compare the control strategies, a test sequence of changes in the heat load of the rooms is created. Using this test sequence, it is possible to detect a difference in performance of the control systems regarding stability, ability to maintain a constant value, and ability to avoid interaction between separate controllers at different parts of the duct system.

1. INTRODUCTION

Ventilation systems using variable airflow are useful in urban areas, since it is important to minimise the flow of polluted air into the building and to vary the airflow due to the indoor load from pollution or thermal sources. One of the most difficult tasks in the design of variable air volume (VAV) systems is to vary the airflow to individual rooms without disturbing the control of the total distribution; components like dampers must work well together with the rest of the ductwork also in transient flow states. Duct systems for variable ventilation flow rates contain a large number of closed-loop controls with more or less close interaction. The controls can be divided into two main groups: those controlling the air handling unit and those controlling the supply of air throughout the building. Most of the studies in this area deal either with the dynamic performance of a single air-handling unit or with demand controlled ventilation. However, some interesting examples of studies of interaction between different parts of the control system include Haves, P., (1994 and 1995), and ASHRAE (1997) ,where the model of the air distribution system is simplified and the purpose is to study the interaction between components in the air handling unit and the terminal units. When performing a more detailed study of air distribution, it is not possible to make large simplifications of the duct system as in the three studies mentioned. The complexity of the duct layout must remain, which implies a large number of equations to be solved simultaneously resulting in slow execution speeds. The interaction of the control-loops largely depends on the design of the duct system and the selection of terminal units. An unstable system for airflow control can sometimes be stabilised to the price of a control system of high complexity and a high cost of investment. This paper introduces a first version of a simulation aid, made to study the duct system and the interaction between the terminal units, the air handling unit (fan and dampers), and control system. The simulation aid is based on a set of individual component models written in NMF code (Sahlin, P. 1996), (Sahlin, P., Bring, A., Sowell. E.F., 1996). The simulation aid will be presented in chapter 2, and an example of simulation of a supply air duct system of a small office building together with a discussion about controllability will be presented in chapter 3 and 4. The simulation aid presented is developed to be used in further studies about design of duct systems and control systems for variable airflow with the purpose to support design of simple systems.

2. DESCRIPTION OF THE SIMULATION AID

The simulation aid that will be described can be divided in two parts: a library of individual components written in NMF code that can be used in several simulation environments, and the IDA simulation environment. In this paper I will concentrate on the library of components and just briefly discuss the simulation environment.

2.1 The library of components

The library can be divided in three parts: duct system, control system, and boundary. In this paper I will only describe the models briefly, the full NMF code will be found in a future publication. Some of the components are created specially for this application and will be treated in the text below, while others are collected from other sources

The duct system so far contains eight component models: straight duct, elbow, transition, tee (converging and dividing), damper, fan, and point of measurement. All of the components, except point of measurement, have their origin in the IDA MAE (Multizone Air Exchange) application (Bring, A., Sahlin, P., 1993). Although some changes have been done in some of them, they will not be presented here. The model Point of measurement is simply used to supply data from an arbitrary point in the duct system, to the control system without modification of the other duct components. It contains no equations or parameters. The model consists of four links that can be connected to the transducers below and two links for letting the air pass by.

The control system so far contains eight component models, actuator/motor, transducers (temperature, pressure, flow, and X-transducers), PI-controller, and two models used to compare and select max or min value of a number of signals. The motor or actuator has constant speed. The actuator starts moving when the difference between current out signal (damper position) and in signal from the controller exceed specified limits and stops when the difference falls below other limits. To describe the actuator, values of the dead bands, the normalised inverted speed, and the minimum actuator position must be supplied. If the dead band is set too large it makes the system harder to control, and if it is to small the simulations will be harder to perform due to numerical problems. All the transducers used (temperature, pressure, flow, and X) are based on the same structure. The out signal is between 0 and 1. The lumped capacitance method is used and simplified to a single time constant. To describe the model the values of the time constant, and maximum measured value allowed must be supplied. The X transducer is used for the measurement of moisture or any pollutant in the air. The model of the PI-controller is a classic one with anti wind-up and the out signal limited to a value between 0 and 1. The tuning of the controllers is crucial for the behaviour of the simulated system. The models, findmax and findmin are used to select maximum or minimum value of a varying number of in signals and send the value to the control unit. The component must be connected to at least three signals. There are no parameters to supply.

The boundary so far consists of one model of a room – and a simple model – outside air properties, used as an air intake. The room model serves as a boundary for the duct system. The thermal performance of the room is modelled by the use of the lumped capacitance method for simulating the wall temperatures. The air is modelled as perfectly mixed, and in this model the exhausted air always equals the supplied. The model, outside air properties, contain no equations, only parameters used to set the outside pressure, temperature, and pollution. When no air-handling unit is available, these values are used to set the values of the treated air.

2.2 The IDA simulation environment

The IDA simulation environment consists of three separate parts: NMF-translator, IDA-modeller, and IDA-solver. They are all needed when creating an application, but when it is finished you only use the IDA-modeller and -solver. The NMF - translator is used to create Fortran files from NMF models to be used by the IDA solver or other simulation platforms such as TRNSYS and HVACSIM+. The NMF-translator is also used to create files that can be used together with the IDA modeller. IDA- modeller is the graphical interface, which helps to put individual components together into larger systems. (Sahlin, P., 1993). From this platform you create your system model and perform all simulations, it also supplies the possibility to create scenarios for the simulation by using text files or tables. It is also possible to log any variable to an arbitrary number of output files. It is possible to use a toolbox in the modeller to create tailored applications. The most important part of the environment is the IDA - solver, the variable step length solver of systems of differential and algebraic equations (Sahlin, P., Bring, A., 1991).

3. SIMULATIONS OF AN OFFICE BUILDING

3.1 The simulated duct system

To illustrate what kind of problems that can be studied with the simulation aid, simulations of a simplified model of the duct system of an existing office building have been performed. The building has two floors, each with 6 rooms. In the real building, the air is supplied to the office rooms by VAV terminals and exhausted through CAV (Constant Air Volume) dampers in the sanitary areas and VAV dampers in the corridor. The air handling units are located at the top floor and each unit supplies 15 m^3/s air at a pressure of about 1600 Pa. From each unit the air is distributed through three main ducts, located at the south, central, and northern parts of the building. The air temperatures are equal for all mains and kept constant at 15 degrees. The initial strategy is to control the fan rather slowly, the dampers in the branches fast and the VAV Terminal units quite slowly. The variable speed control of the fan is set to maintain a constant pressure of 350 Pa at the top of the main ducts. There is one pressure transducer at the top of each main, figure 1, the pressures are compared and the lowest one selected to control the system. At the beginning of each horizontal branch there is a damper for maintaining a constant pressure of 200 Pa at the end of the branch. These dampers have linear characteristics and fast actuators. The room temperature is maintained constant at 20 degrees by separate dampers in each room. In this small study, only the supply system was simulated.

One difficult part of the design is to tune the control system. A modified Ziegler-Nichols method (Sørensen, B.R., Novakovic, V., 1996) was used as a first estimation and after that corrections of the gain and integration time were made. The integration time was set to a number close to the inverted speed of the actuator and the gain was set to the best value of a few tested by simulation. Each time the pressure of the dampers was changed, tuning of the controllers, and start up calculation had to be performed, which was very time consuming.

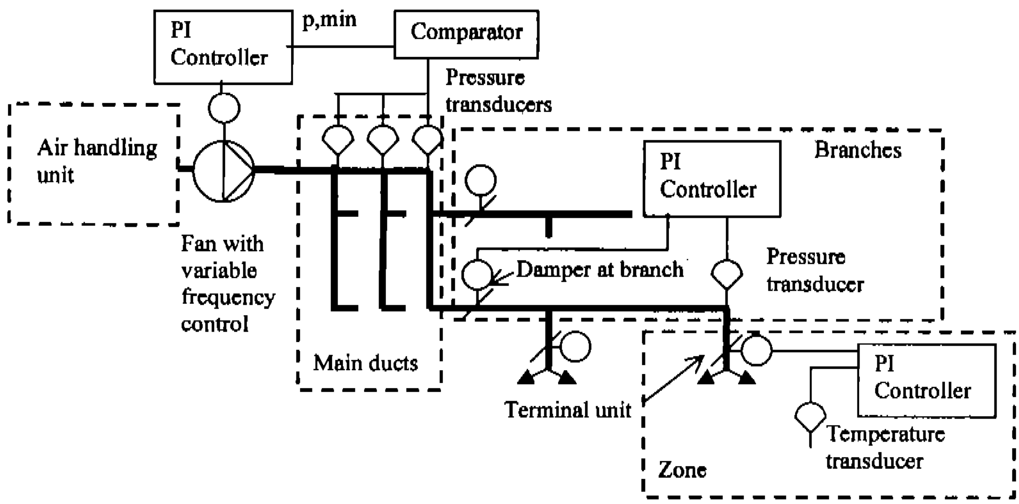


Figure 1: Scheme of the control system.

The duct system was designed to meet the load of 150 W of equipment, 80 W lightning, and 1 person. The dampers of the branches were designed to a pressure drop of about 150 Pa (40 % open) and the terminal units to a pressure drop of 200 Pa (55 % open).

3.2 Simulated scenarios

A question that often is raised is; what are the characteristics of a good controlsystem? An easy way to make an evaluation is to integrate the error in some way, usually its squared or absolute value (Schmidtbauer, B., 1988). It is not only the error that is important, usually one wants to minimise the signals from the controllers to avoid unnecessary movements of the dampers (Glad, T., Ljung, L., 1997). According to ASHRAE 825-RP (ASHRAE, 1996) you can look at two levels, first at one single loop and then at the entire building. In the first case, the common criteria of good control, static accuracy, rise time and stability can be used. If you look at the entire building, it becomes more complex. There are often compromises to be made among contradictory conditions. Among the factors you can select, such as criteria of good control, are discomfort, energy use, maintenance, and investment. There are also more factors to be considered as noise level and disturbing frequencies generated by modulation of dampers and the fan speed. A measure that includes several of the factors mentioned could be to perform a simulation of a sequence of relevant changes in the load of the rooms and measure the time at which the controlled quantity is stable and within its wanted limits. This is illustrated in figure 2. The controlled variables of a VAV system are usually the room temperature or the pressure at some point in the duct system. By summarising the time ranges t1, t2... that correspond to periods of stability and accuracy, and dividing the sum by the total time, you reach a figure that approaches one when the control is perfect. There will always be some fluctuations in the system. This implies that a system must be considered as stable when the fluctuations of the variables are within a specified limit. The accuracy is also defined, as a limit within which the variable should stay. When looking at the fluctuation of the pressure of the terminal units, if the limits are expressed as a fraction of the nominal pressure loss of the dampers, the change in supplied mass flow during the fluctuations will be the same, not depending of the actual nominal pressure loss. A fast and stable control should give a high figure. A high figure could also imply low wear of the equipment and small risk of unwanted disturbing frequencies.

The simulations were started at a steady state condition and the changes of the heat load in the rooms were then applied to the system. The duct system was designed for 70% of the maximum load used in the sequence. Besides the original control, one simplified control strategy with the pressure control of the branches removed was tested.

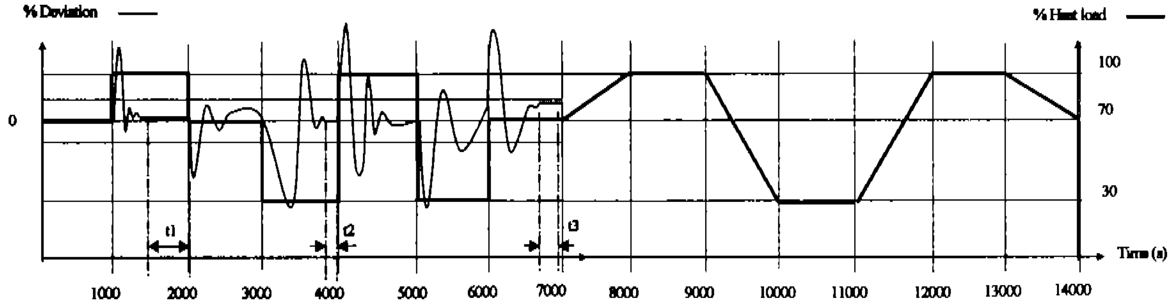


Figure 2: Sequence of heat load and a fictitious example of a deviation of one controlled variable.

For each of the two control strategies, five simulations were performed each, using the nominal pressure over the dampers as presented in table 1. For one set of pressure, simulations were performed using different gain (k) of the controllers of the terminal units.

Table 1: Summary of the scenarios simulated.

	Scenario, pressure in branch and terminal unit				
Damper in branch	150	300	50	50	150
Terminal unit	200, k=20	50, k=5	300, k=20	300, k=5	25, k=5

3.3 Performing simulations

When using the IDA simulation environment such a duct system is quite easy to model. Looking at the graphic interface of IDA Modeller, it is easy to follow the structure of the model (see figure 3). The simulations are controlled by supply of data through input files. The parameters of the individual components can be changed by the use of a dialog box that appears when the symbols are opened. Before any changes in load are put to the system it has to be in a steady state. It usually takes long simulation runs to create a steady state starting condition, but it is possible to make one start up calculation that can be used as a starting condition for all simulations.

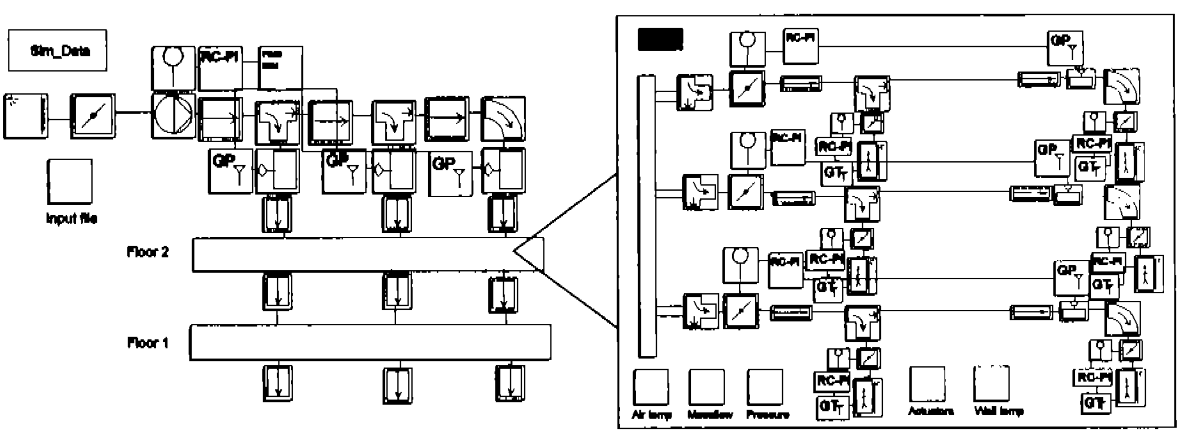


Figure 3: The model of the building as it looks in IDA Modeller. When one floor is opened the duct system appear.

4. RESULTS

Table 2 shows the scenarios simulated, the first scenario is presented in more detail. Figure 4 shows what happens to the temperatures of the rooms when the sequence, figure 2, is used for one room at each floor. As you can see, there is no problem to keep the room temperature within rather strict limits, but this does not say much about the movements of the dampers and changes in fan speed. Figure 5 shows the pressure before the terminal units: This figure more clearly shows the dynamics of the system. The control has a problem to maintain a constant pressure of 200 Pa. It is also possible to detect some interaction between the different controls.

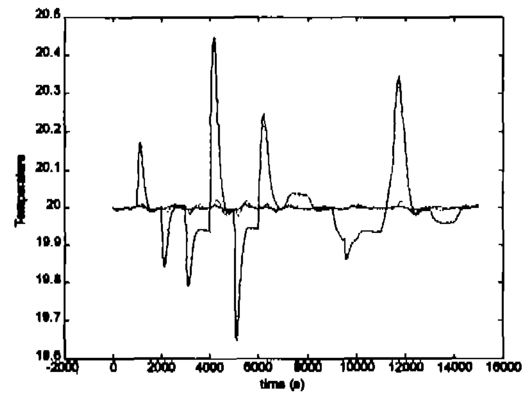
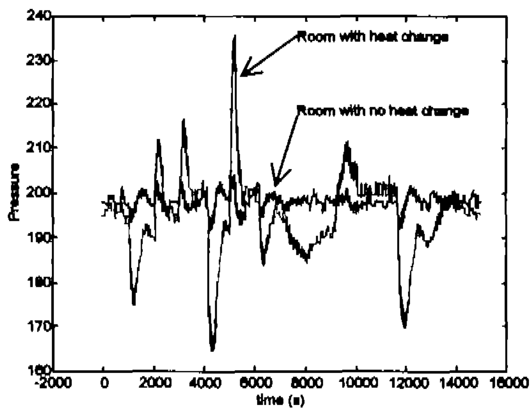


Figure 4: Temperatures of the rooms when using the test sequence of one room at each floor.



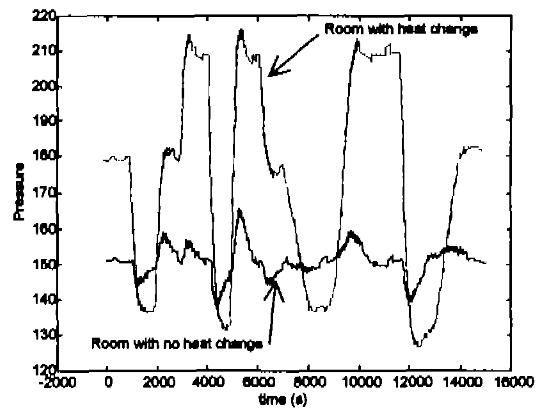


Figure 5: Pressure before the room dampers, original control and simple control respectively.

Figure 4 and 5 show the result, with 150 Pa pressure loss over the damper in the branch, and 200 Pa pressure loss at the terminal unit. A calculation was performed of the ratio described in chapter 3.2, with the limit set to 10%, and with the assumption that the system is stable if the oscillation is below 1%. The result when looking at the pressure before the terminal unit of the room with changes of heat load become 0.29 for the original system and 0.44 for the simplified system, and in a branch where no changes take place the result become 0.75 and 0.32 respectively. When looking at the simplified system, and analysing the pressure before the room with the changes, it is not possible to use the limit of 10%, since the pressure loss in this system should vary when the flow increases. It can also be seen from the simulations that there is almost no difference in the behaviour of the fans in the two simulated systems. A summary of the result is presented in table 2. When looking at the mean values of the two ratios for each control algorithm the best performance for the original control is obtained at scenario 4, and for the simplified control at scenario 3. Least interaction between controllers at different branches is obtained for the original control at scenario 2, and the most stable pressure is obtained for the simplified control at scenario 2.

Table 2: Summary of the performance ratios for the systems simulated.

	Scenario, pressure in branch and terminal unit				
	1	2	3	4	5
Pressure at damper in branch	150	300	50	50	150
Pressure at terminal unit	200,k=20	50, k=5	300,k=20	300, k=5	25, k=5
	Performance ratio, pressure loss of terminal units				
Original control, heat change	0.29	0.13	0.29	**0.645**	0.05
Original control, no heat change	0.75	**0.88**	0.44	0.77	0.46
Mean value:	*0.52*	*0.505*	*0.365*	***0.705***	*0.255*
Simple control, heat change	0.44	0.09	**0.53**	0.284	0.04
Simple control, no heat change	0.32	0.75	0.683	**0.77**	0.52
Mean value:	*0.38*	*0.42*	***0.605***	*0.525*	*0.28*

5. DISCUSSION

There is still a lot of work remaining until the tool presented can be used by engineers in the daily work, but the examples in this study show its usefulness. One of the advantages of this tool is the possibility for the user to select which aspects to study; any variable can be logged to a file during the simulations, and any variable can be kept constant. The simulation time for systems like those presented is about an hour for a 266 MHz computer, but the developments of the computers will shorten this time in a near future. The tool still lacks some important models such as discrete controllers, and more advanced room models that support multiple supply and exhaust terminals and leakage to other rooms and to the environment.

As discussed in chapter 3.2, the definition of good control depends on what system of the building that is studied. The duct system is something in between the entire building and a single control-loop. For this, a test sequence of the kind presented here seems appropriate. When calculating performance ratios for different control strategies and scenarios, the values obtained are in such a wide range that it is possible to detect differences between the systems. If the compared systems behave slowly, the time between the changes in load must be increased if it should be possible to detect the difference between them. The evaluation of the test sequence was performed manually, but a model for estimation of such control deviation will be added to the next version of the library of components.

When looking at the overall performance by using the sum of performance ratios, it looks like the original control with most of the pressure loss located at the terminal unit and a moderate gain at the controller produces the best result. Least interaction between controllers located at different branches seems to be obtained when using the original control with most of the pressure loss located at the damper at the branch. Also, as when looking at the best overall performance, moderate gain. Although this study is based on too few simulated scenarios, it is evident that the behaviour of the duct system is determined by distribution of the pressure loss throughout the system. As could be expected, the interaction between controls at different branches depends largely on the pressure loss at the damper located at the branch. At a constant sum of the pressure loss at the branch and terminal unit there probably exists a combination that gives the highest performance ratio when looking both at an individual controller and at the distribution of disturbances throughout the duct system. Also the tuning of the controllers plays an important role. To change the gain of the controllers will often have the same effect as changing the nominal pressure loss of the dampers.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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ON NATURAL VENTILATION OF A BUILDING WITH TWO OPENINGS

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SYNOPIS

Analytical solutions are derived for calculating natural ventilation flow rates in a single-zone building with two openings when no thermal mass is present. In these solutions, the independent variables are the heat source strength and wind speed, rather than given indoor air temperatures. Three air change rate parameters α, β and γ are introduced to characterise respectively the thermal buoyancy force, the conduction heat loss effect, and the wind force. The wind can either assist the buoyancy force or oppose it. For assisting wind the flow is always upwards and the solutions are straightforward. For opposing wind, the flow can be either upwards or downwards depending on the relative strengths of the two forces. In this case the solution for the flow rate as function of the heat source strength presents some complex and unusual features.

LIST OF SYMBOLS

A	area of the envelope
A_b	area of the bottom opening b
A_t	area of the top opening t
A^*	effective opening area
C_d	discharge coefficient
C_p	pressure coefficient
E	total heat power
E_i	heat from people, equipment and lighting
E_s	direct solar gain through windows
g	gravitational acceleration
h	height between two vertical openings t and b
ΔP_w	wind pressure
q	volumetric flow rate
T_i	air temperature in the building
T_o	outdoor air temperature
$T_{solair,j}$	sol-air temperature acting on building element j
V	wind speed
U_j	U-value of building element j

Greek Symbols

α	thermal buoyancy air change parameter
β	fabric loss air change parameter
γ	wind air change parameter
ρ	air density

Subscripts

o	outside
i	inside
b	bottom opening
t	top opening
w	wind

1. INTRODUCTION

Natural forces, in particular thermal buoyancy and wind, drive natural ventilation of buildings. There are a large number of governing factors in natural ventilation, such as wind speed and direction and its turbulence, the size and position of ventilation openings, heat sources, the envelope conductance, and so on. Accurate prediction of natural ventilation rate at the design stage is often very difficult. The key to accurately predicting natural ventilation rates lies in predicting the combined effects of the two natural forces.

In most design problems, the thermal buoyancy force and ventilation flow rates are interdependent. Most studies in the past only considered situations where the indoor air temperatures are given, see Foster and Down (1987) and Andersen (1995). Etheridge and Sandberg (1984) and Etheridge and Stanway (1988) presented two excellent parametric investigations. Their studies used a non-dimensional approach to demonstrate the relative importance of the various parameters. However, they assumed that the indoor air temperatures are known. In reality, the heat sources, wind speed and thermal conductance of the building envelope are known, and the indoor air temperature and ventilation flow rate are derived from these parameters.

Although numerical methods can now be used to revisit the work of Etheridge and Sandberg (1984) and Etheridge and Stanway (1988), and produce non-dimensional graphs for the simple cases they considered, an analytical method is preferred before a full numerical study is carried out. It is much easier to carry out a parametric study with analytical solutions than with numerical experiments.

Most today's design codes on natural ventilation still adopt simple semi-analytical solutions (i.e. the hot air column model and cross-wind ventilation model) as design tools, such as those in the CIBSE design guide (CIBSE, 1988) and BS5925 (1991). These formulae have been shown to provide reasonable estimates of natural ventilation flow rates in many situations. However, one of the difficulties in using them is that the indoor air temperature must be known beforehand. One has to adjust the air temperature used after the natural ventilation flow rate has been calculated. To obtain a consistent estimate of both flow rate and indoor air temperature is quite often practically not possible.

This paper derives analytical solutions for the ventilation rate in a single-zone building with two openings, considering the effect of buoyancy force, wind force and heat conduction loss through the building envelope, and their interactions. We assume that there is no thermal mass in the building. If thermal mass is included, no analytical solution exists. However, the model can still apply to some practical buildings such agricultural (e.g. livestock) and industrial buildings with relatively low thermal mass. When ventilation air flow rates are very large, then the thermal mass may also be neglected. The effect of thermal mass on natural ventilation will be a subject of a future paper.

2. NATURAL VENTILATION DRIVEN BY COMBINED THERMAL AND WIND FORCES

Consider a simple building with two openings at different vertical levels on opposite walls, as shown in Figure 1. There is an indoor source of heat, E_i, and solar radiation acts on the building via a sol-air temperature for the opaque elements and solar heat gain through windows. The wind force can assist or oppose the thermal buoyancy force. We assume that the indoor air is fully mixed, i.e. the air temperature is uniform. This assumption is generally not valid for thermal buoyancy force-dominated flows. Wind turbulence effects are not included.

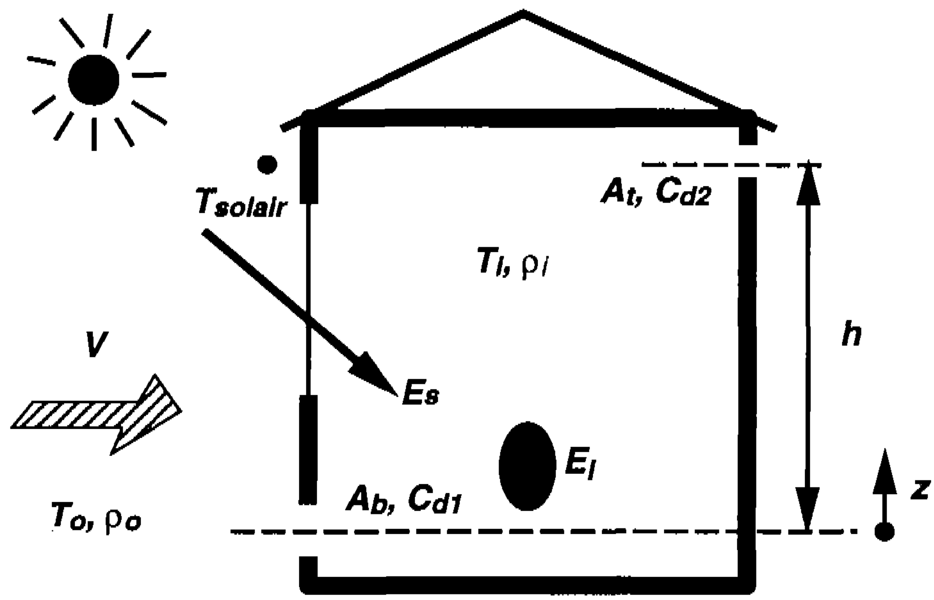

Figure 1. A two-opening building with solar radiation through windows

A heat balance on the building gives

$$\rho c_p q(T_i - T_o) + \sum_j U_j A_j (T_i - T_{sol-air,j}) = E_i + E_s . \tag{1}$$

This can be rearranged as

$$\rho c_p q(T_i - T_o) + \sum_j U_j A_j (T_i - T_o) = E \tag{2}$$

where

$$E = E_i + E_s + \sum_j U_j A_j (T_{solair,j} - T_o). \tag{3}$$

Depending on the arrangement of ventilation openings, wind can assist or oppose the thermal force in natural ventilation. Two extreme situations are considered here, i.e. fully assisting and fully opposing. These occur when there are only two ventilation openings. It appears that no analytical solutions exist for more than two openings.

2.1 Assisting Wind Force

It is easy to show that the flow rate is given by

$$q = C_d A^* \sqrt{2gh\frac{T_i - T_o}{T_o} + 2\Delta P_w}\ , \tag{4}$$

where the effective opening area, A^*

$$A^* = \frac{A_t A_b}{A_t^2 + A_b^2}\ , \tag{5}$$

and the wind pressure, ΔP_w

$$\Delta P_w = \frac{1}{2}C_{p1}V_1^2 - \frac{1}{2}C_{p2}V_2^2\ , \tag{6}$$

where the subscripts 1 and 2 refers to two ventilation openings. ΔP_w is always non-negative. In equation (4) the C_d values are assumed to be the same for both openings. It is fairly easy to derive the formula for non-equal C_d values. Both A_b and A_t are free-opening areas. In deriving equation (4), the power-law equation was used to describe the relationship between the flow rate and the pressure difference.

Substituting equation (2) into (4) gives, after some manipulation,

$$q^3 + 3\beta q^2 - 3\gamma^2 q - 2\alpha^3 - 9\gamma^2\beta = 0 \tag{7}$$

where

$$\alpha = \left(C_d A^*\right)^{\frac{2}{3}}\left(\frac{Egh}{\rho c_p T_o}\right)^{\frac{1}{3}}, \tag{8}$$

$$\beta = \frac{\sum_j U_j A_j}{3\rho C_p} \tag{9}$$

and

$$\gamma = \frac{1}{\sqrt{3}}\left(C_d A^*\right)\sqrt{2\Delta P_w}\ . \tag{10}$$

The three air change parameters α, β and γ quantify respectively the effects of heat gains, fabric heat losses, and wind. For a perfectly insulated building, when the buoyancy force acts alone, $q = \sqrt[3]{2}\alpha$, and when the wind force acts alone, $q = \sqrt{3}\gamma$.

There are many graphical ways to present the ventilation flow rate as a function of these parameters. Examination of equation (7) shows that the solution can be easily presented in a non-dimensional graph, shown in Figure 2. The important non-dimensional ratio α/γ is a relative measure of the two driving forces.

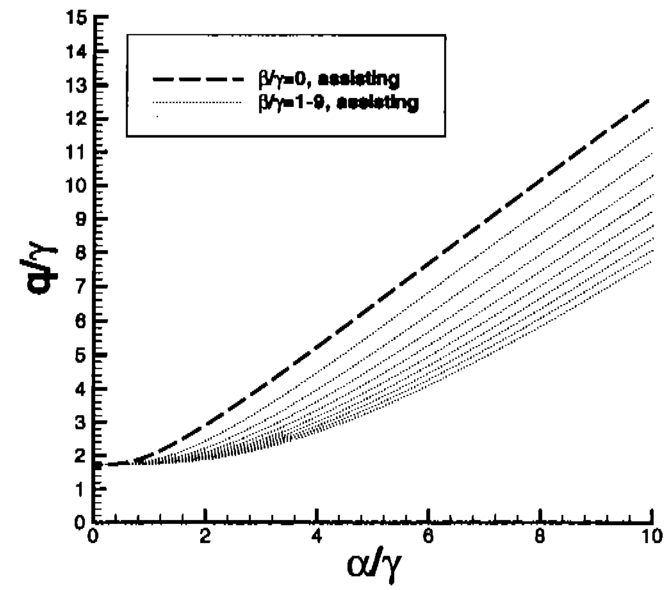

Figure 2. Natural ventilation graph for combined forces with assisting winds – non-dimensional flowrate vs heat gain parameter at varying insulation levels.

The effect of heat loss on ventilation flow rate is not linear. There is a big drop in the ventilation flow rate as β increases from 0. As β further increases, the resulting rate reduction in the ventilation flow rate slows down.

Figure 2 is not applicable when $\gamma = 0$, i.e. no wind. A much simpler graph can be produced (not shown here). For design purposes, Figure 2 is easy to use. A similar graph can be produced for indoor air temperature. Air temperatures can also be calculated by equation (4).

2.2 Opposing Wind Force

So far the flow direction considered is always upward. When the wind force opposes the thermal buoyancy force, i.e. the wind blows onto the upper opening, the flow can be either upward or downward, depending on the relative strength of the forces. Again, it can be shown that

$$q = C_d A^* \sqrt{\left| 2gh\frac{T_i - T_o}{T_o} - 2\Delta P_w \right|} \, . \tag{11}$$

Applying the heat balance equation gives

$$q^3 + 3\beta q^2 = \left|2\alpha^3 - 3\gamma^2 q - 9\gamma^2\beta\right|. \tag{12}$$

For upward flows, the buoyancy force is stronger and $2\alpha^3 > 3\gamma^2 q + 9\gamma^2\beta$. We have

$$q^3 + 3\beta q^2 + 3\gamma^2 q - 2\alpha^3 + 9\gamma^2\beta = 0. \tag{13}$$

For downward flows, the wind force is stronger and $2\alpha^3 < 3\gamma^2 q + 9\gamma^2\beta$. We have

$$q^3 + 3\beta q^2 - 3\gamma^2 q + 2\alpha^3 - 9\gamma^2\beta = 0. \tag{14}$$

Analytical solutions of equations (13) and (14) can be written down, but in practice they are best solved numerically. However, an analysis of the behaviour of the flow rate as a function of α (the heat source) reveals that the general form of the solution must be as shown in Fig. 3.

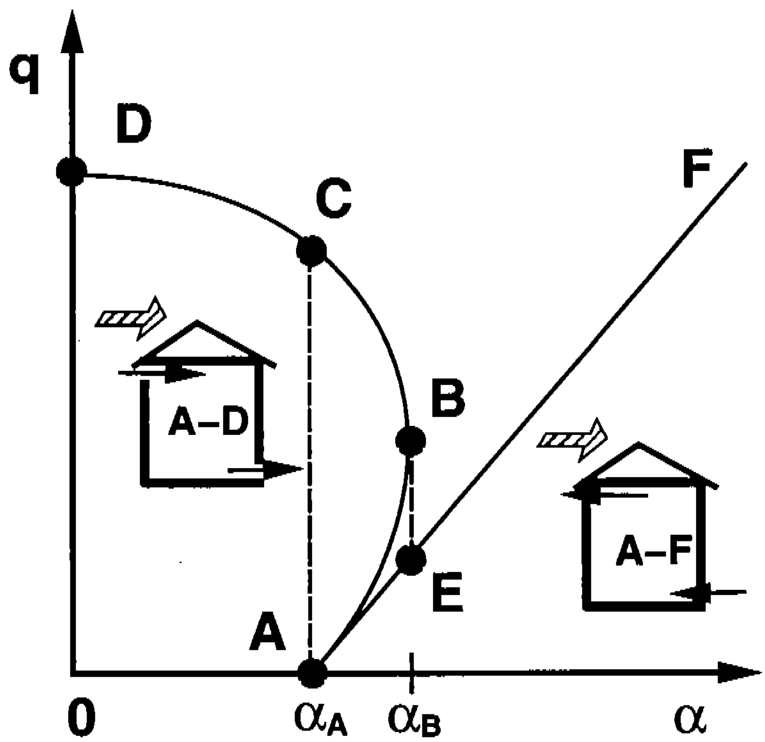

Figure 3. A sketch of the analytical solutions of equation (12)

This shows some interesting features. For example, consider a very low value of α, so that the flow is definitely downward. As α increases, we move to the right along the downward flow curve, and the flow rate decreases. However, when α reaches a critical value, α_B, denoted by point B in Fig. 3, an interesting phenomenon occurs. If α increases slightly, the flow direction reverses to upward flow and the flow rate drops to a lower value, denoted by point E. If α then increases further the flow rate increases, as would be expected. However, if at E α now decreases, the upward flow rate can decrease to zero at $\alpha = \alpha_A$, at point A. If α further decreases, then the flow reverses to downward, and the ventilation flow rate jumps to point C.

Furthermore, for $\alpha_A < \alpha < \alpha_B$, there appear to be three possible flow rates for a given value of α: two downward flows and one upward flow.

Finally, the state of the system represented by the curve A-B is unusual. Here the flow direction is downward, but an increase in α results in an increase in the flow rate. This is counter-intuitive: because the wind is opposing the buoyancy force and is stronger (i.e. downward flow), one would expect that an increase in the buoyancy force would result in a decrease in the flow rate, not an increase. Thus the curve A-B may a non-physical region, even though points on this curve do satisfy the original equations. It is not clear what other constraint can be applied to show that A-B is non-physical.

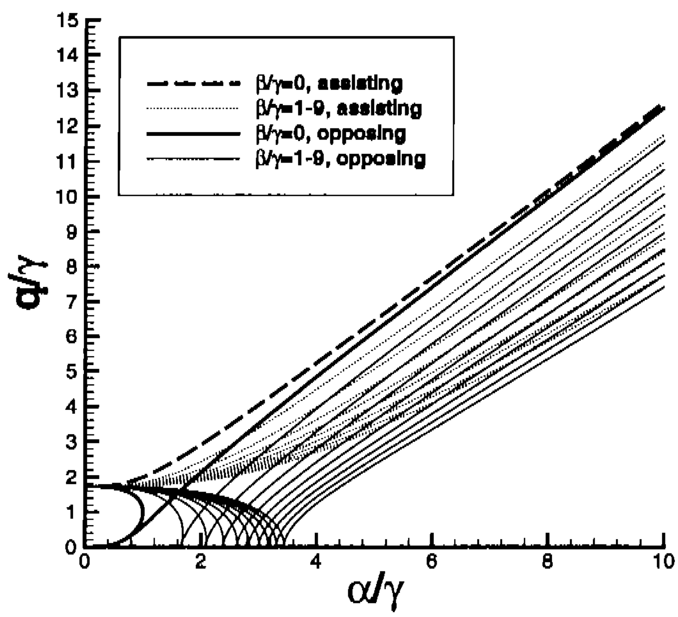

Figure 4. Non-dimensional flowrate vs heat gain parameter at varying insulation levels for both assisting and opposing winds.

3. CONCLUSIONS

Analytical solutions are derived for natural ventilation in a single-zone building with two openings. Three air change parameters are introduced: the thermal buoyancy air change parameter α, the wind air change parameter γ and the fabric loss air change parameter β. We believe that simple ventilation graphs, such as Figures 2 and 4, can be used for design purposes. By analysing these graphs and the underlying equations, the following conclusions can be drawn.

- The ventilation flow rate is simply proportional to α or γ, when each driving force acts alone.
- The effect of heat loss through the building envelope is very significant. The ventilation flow rate drops sharply as β increases from 0. The rate of change in the ventilation flow rate slows down when β further increases. When a wind force is present, the heat loss effect also interacts with wind-induced flows. This is due to the fact that heat loss depends on indoor air temperature, which is again controlled by the ventilation flow rate.
- When the wind force opposes the thermal buoyancy, for a certain range of α values, there appears to be three possible flow rates for a given value of α: two downward flows and one upward flow, depending on whether α has been increasing or decreasing, i.e. the system exhibits hysteresis.
- With an opposing wind force, for a given set of α and γ values it is possible that two different β values can result in the same ventilation flow rate, but in opposite flow directions.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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Top-down natural ventilation of multi-storey buildings

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Top-Down Natural Ventilation of Multi-Storey Buildings

SYNOPSIS

We examine natural ventilation in buildings with multiple storeys, each storey linked to a common chimney or atrium, and ventilated using 'top-down chimneys' to draw in relatively unpolluted air from openings located high above street level. Two significant issues relating to ventilation design and management are addressed. First, the common stack provides connections between every storey and, consequently, the ventilation of each storey cannot be calculated in isolation, but must be calculated simultaneously for all storeys. Second, the introduction of the top-down chimney results in frictional losses whose magnitude depends, in part, on the chimney length and cross-sectional area. We develop a simple theoretical model to quantify the effect each storey has on others, and to predict how ventilation openings should be resized in order to overcome the pressure losses associated with the top-down chimney. The model describes the thermal stratification and ventilation flow rate in each storey, and leads to design curves for the sizing of vents to achieve the required ventilation. We focus on steady-state displacement ventilation and compare our theoretical predictions with paradigm small-scale laboratory experiments in a model two-storey building. Our study indicates that, with careful design, top-down ventilation of multi-storey buildings is a realistic strategy in urban environments.

LIST OF SYMBOLS

a	opening area (m^2)	n	number of storeys
A^*	'effective' opening area (m^2)	P	pressure (Pa ≡ $kgm^{-1}s^{-2}$)
c	plume entrainment constant	Q	ventilation flow rate (m^3s^{-1})
c_p	specific heat capacity of air ($Jkg^{-1}K^{-1}$)	Re	Reynolds number = UD/ν
C	loss coefficient	T	temperature (K)
D	diameter of top-down chimney (m)	U	mean velocity (ms^{-1})
E	power of internal heat gains (W)	V	volume of storey (m^3)
f	friction factor ≡ 5×10^{-3}		
g	acceleration due to gravity (ms^{-2})	β	coefficient of thermal expansion (K^{-1})
h	interface height (m)	ν	kinematic viscosity of air (m^2s^{-1})
H	room height (m)	ρ	density of air (kgm^{-3})
L	length of top-down chimney (m)		
M	stack (chimney or atrium) height (m)		

Subscripts - refer to quantities:

inlet	at top-down chimney inlet	*TDC*	in top-down chimney
$j\,(=0, 1,.., n-1)$	on j^{th} storey	*total*	along a complete flowpath
stack	in stack (chimney or atrium)		

1. INTRODUCTION

Despite the energy-saving potential of natural ventilation systems, there remain a number of technical barriers restricting their implementation. These barriers must be overcome before natural ventilation is more widely accepted in urban areas as a realistic alternative to forced ventilation or air conditioning. The recent NatVent programme[1], involving seven European countries with temperate or cold climates, has identified the main technical barriers. These include problems associated with air and noise pollution in urban areas, the need for natural ventilation to provide a 'constant' supply of fresh air independent of short-term fluctuations in the driving pressure forces, and the need for controlled passive cooling. In cold climates, there may also be a requirement for heat recovery from natural ventilation, which could otherwise result in unacceptably high energy consumption. In this paper, we consider a technique for implementing natural ventilation in the urban environment which may significantly reduce the impact of noise and air pollution within the building.

In temperate and cold climates, in which indoor air is typically at a higher temperature than ambient (outdoor) air, the primary goals of ventilation are to supply fresh air and to remove excess heat generated within the building by internal and solar heat gains. A desirable and efficient mode of natural ventilation in these climates is displacement ventilation, in which cool ambient air is introduced at low levels, and warm, stale air leaves at high levels. The position of the openings leads to the development of a stratification within the space. The air inlets are typically at street level where pollution and noise levels may be high, which raises concerns over indoor air quality, and may restrict the use of natural ventilation. One possible solution to this problem, as described by Gage, Hunt and Linden[2] for a single space, is to locate the air inlet well above street level, where the air is typically less polluted, and to draw the air down through a duct before releasing it at low levels in the room. This strategy is aptly referred to as 'top-down' ventilation, as it involves drawing air from the *top* of the building *down* into the ventilated spaces below, see figure 1.

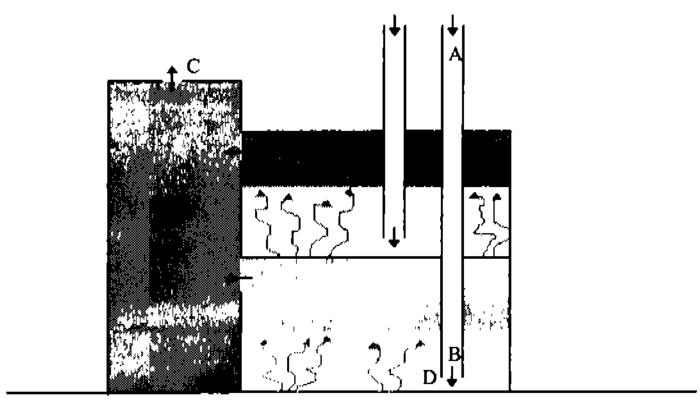

Figure 1. Schematic of top-down ventilation in a two-storey atrium building.

Although, at first sight, the concept of top-down natural ventilation may seem counter-intuitive, ambient air *will* be drawn down the top-down chimney (TDC), providing the difference in internal pressure between the top (A) and bottom (B) of the TDC is greater than the difference in pressure inside the building between the high-level ventilation outlet (C) and the base of the TDC (D). Internal heat gains in the room cause a build-up of warm air near the ceiling and in the stack, which provides this pressure difference. Note that it may be necessary to insulate the TDC from solar/internal heat gains to ensure that the air inside the TDC remains close to ambient temperature. Top-down ventilation in a single-storey space has been successfully demonstrated by Gage *et al.*[2], who simulated these flows at small scale in water tanks.

If there are constant heat gains within an enclosure, then, after some time, a steady situation develops, in which the rate of heat input is balanced by the rate of removal of heat by the ventilation flow. For a single room, this steady flow has been modelled by Linden, Lane-Serff and Smeed[3], using a point source of heat on the floor to represent heat gains, and was shown to depend on the 'effective' opening area A^* of the inlet and outlet. However, in multi-room buildings, the flowpaths between inlets and outlets are typically more complex, and may link numerous internal spaces. When there are isolated flowpaths through a building, in which each inlet/outlet pair is connected by an independent air route, the effective area A^* of each flowpath may be calculated by summing the areas of the individual openings:

$$\frac{1}{A^{*2}} = \sum_{j=1}^{n} \frac{1}{2C_j a_j^{\,2}}, \tag{1}$$

where a_j ($j = 1, 2, .., n$) is the area of the j^{th} opening and C_j is the loss coefficient associated with that opening. Expression (1) is valid for flows driven by any combination of stack and wind effects[4,5,6].

In this paper we extend the work of Linden *et al.*[3] and Gage *et al.*[2] by considering the application of top-down ventilation to multi-storey spaces that are linked by a common atrium or chimney. Linking a number of storeys via a common stack creates a more complex building geometry, in which flowpaths are no longer isolated, but merge and divide. This poses a number of questions at the design stage such as '*how should the openings be sized on each floor to meet ventilation requirements*?' and '*how does the ventilation of one floor affect the ventilation of another*?' To address these and related questions we develop a simple theoretical model and determine the airflow rate and temperature stratification on each storey of a generic stack ventilated building. Additionally, in top-down ventilation, poor design may result in significant pressure losses through the relatively tall TDCs which link the lower storeys to inlet vents at the top of the building. Here, the additional losses associated with the use of TDCs are quantified.

Model predictions are presented in the form of design curves which may be used to determine the opening areas and TDC dimensions subject to specific design criteria, such as heating loads and required air changes per hour. These design curves are intended as an aid in the initial planning of multi-storey buildings with top-down ventilation. The predictions are compared with the results of salt-bath laboratory experiments, in which natural ventilation flows were produced in a small-scale model of a generic two-storey building.

2. THEORETICAL MODEL FOR NATURAL VENTILATION OF MULTI-STOREY BUILDINGS

2.1 Loss in driving pressure due to top-down ventilation

If the driving stack forces are weak, as is often the case, then any additional pressure losses that arise in top-down ventilation are undesirable, and designers must aim to minimise these losses in order to make best use of the driving forces. We now quantify the magnitude of the frictional pressure drop ΔP along a TDC, and illustrate its dependence upon the chimney's aspect ratio L/D, where L is the length and D the diameter. For a cylindrical TDC drawing air of density ρ at a mean velocity U into a building, the frictional pressure loss may be expressed as

$$\Delta P = \frac{2Lf\rho U^2}{D}, \tag{2}$$

where the friction factor f decreases with Reynolds number Re and may be approximated by

$$f = \begin{cases} 16/\mathrm{Re} & \text{for } 10 < \mathrm{Re} < 2000 \text{ - laminar flow;} \\ 0.079\,\mathrm{Re}^{-1/4} & \text{for } \mathrm{Re} < 10^5 \text{ - turbulent flow,} \end{cases} \tag{3}$$

see Ward-Smith[7]. Here, the Re is based on the diameter of the TDC and mean air velocity, *i.e.* $\mathrm{Re} = UD/\nu = 4Q/\pi\nu D$, where Q is the ventilation flow rate. Since $U = Q/a_{TDC}$, we may write

$$\Delta P = \frac{\rho Q^2}{2C_{TDC}a_{TDC}^2} \quad \text{with} \quad C_{TDC} = \frac{D}{4Lf}, \tag{4}$$

where $a_{TDC} = \pi(D/2)^2$ is the cross-sectional area of the TDC and C_{TDC} is a loss coefficient for the TDC. From (4), it is clear that frictional losses increase with f, increase as the TDC area a_{TDC} decreases and increase as the aspect ratio L/D of the TDC increases.

In most building ventilation applications, we anticipate that the flow in the TDC will be turbulent and, in this case, relatively large changes in Re result in only small changes in f: for $5000 < \mathrm{Re} < 10^5$, f lies in the range $9.4\times10^{-3} > f > 4.4\times10^{-3}$. Therefore, we make the approximation that the friction factor f is independent of Re, over the range of Re expected, and take $f = 5\times10^{-3}$. This simplifies the representation of frictional losses, since C_{TDC} is then a constant for a given aspect ratio L/D.

In a similar way to (1), the two pressure loss contributions from the TDC, due i) to flow through the TDC inlet, of area a_{inlet} and loss coefficient C_{inlet}, and ii) to the frictional losses described above, can be shown to combine as a 'TDC effective area' A^*_{TDC} given by

$$1/{A^*_{TDC}}^2 = 1/2C_{inlet}a^2_{inlet} + 1/2C_{TDC}a^2_{TDC}. \quad (5)$$

Thus, the use of a TDC effectively reduces the area of the openings along the entire flowpath, and will therefore decrease the ventilation flow rate.

Frictional losses may be neglected only when they are significantly less than losses through the openings. As an example, for a TDC with $D = 0.5$ m and $L = 6$ m (*i.e.* $L/D = 12$), the loss coefficient $C_{TDC} = 4.2$. Then, from (5), for a typical loss coefficient of $C_{inlet} = 0.63^2 \approx 0.4$ at the inlet of the TDC, friction can only be neglected if $a_{inlet} \ll 3.2a_{TDC}$, *i.e.* only if there is a considerable constriction at the inlet. In most situations, therefore, friction in the TDC is likely to be sufficiently large that it should be taken into consideration.

2.2 Relationship between ventilation flow rate and stratification along a flowpath

For a two-storey building with displacement ventilation (figure 1), there are two flowpaths; the first through the lower storey and into the stack, and the second through the upper storey and into the stack. The solution for the steady airflow rate and stratification is then given by two simultaneous equations, one for each flowpath, and by conservation of volume and heat fluxes. By extension, n simultaneous equations can be written for a building with n storeys connected to a common stack. If each flowpath j comprises a TDC with effective area A^*_{jTDC}, leading into the j^{th} storey with effective area A^*_j and out into a common stack of height M with outlet effective area A^*_{stack}, then the n simultaneous equations describing the flow are

$$Q_j^2\left(\frac{1}{A_j^{*2}} + \frac{1}{{A^*_{jTDC}}^2}\right) + Q^2_{stack}\left(\frac{1}{{A^*_{stack}}^2}\right) = \frac{\Delta T_j}{T}g\left(H_j - h_j\right) + \frac{\Delta T_{stack}}{T}g\left(M - \sum_{k=0}^{j} H_k\right), \quad (6)$$

for $j = 0,\ldots n-1$, where the ventilation flow rate and the excess air temperature (*i.e.* temperature above ambient) in the stack are

$$Q_{stack} = \sum_{j=0}^{n-1} Q_j \quad \text{and} \quad \Delta T_{stack} Q_{stack} = \sum_{j=0}^{n-1} Q_j \Delta T_j, \quad (7)$$

on the assumption that the stack is well-mixed. Here, H_j, h_j, ΔT_j and Q_j are the room height, interface height (*i.e.* depth of cool lower layer), excess temperature of the warm upper layer and ventilation flow rate for storey j, respectively. The effective area A^*_j of the j^{th} storey is calculated by summing the upper and lower internal opening areas, as in (1).

2.3 Modelling heat gains

In this simple model, we assume that the heat gains within each storey can be represented as a point source of heat. In some situations this assumption may represent a considerable simplification, as heat gains may arise from distributed sources or from numerous interacting localised sources of heat. However, as in Linden *et al.*[3], we shall see that this simplification allows us to develop approximate design guidelines and rules of thumb. From the theory of turbulent plumes[8], the temperature departure from ambient ΔT_j and volume flow rate Q_j, at a vertical height h_j from a heat source of strength E_j, are

$$\Delta T_j/T = \left(\beta E_j/\rho c_p\right)^{2/3} / c g^{1/3} h_j^{5/3} \quad \text{and} \quad Q_j = c\left(g\beta E_j/\rho c_p\right)^{1/3} h_j^{5/3}, \qquad \text{(8a,b)}$$

respectively, where $c \approx 0.14$ is a parameter dependent upon entrainment into the rising thermal plume. The height h_j of the interface for the displacement flow within each storey is given by the solution of (6) and (7), with Q_j and ΔT_j given by (8), and may be obtained numerically using a simultaneous root-finding algorithm. Once h_j has been determined, Q_j and ΔT_j are found from (8), and the number of air changes per hour in the j^{th} storey (ACH_j) is given by $ACH_j = 3600Q_j/V_j$, where V_j is the volume of the j^{th} storey.

3. LABORATORY VALIDATION

In order to simulate natural ventilation flows in linked spaces, experiments were performed using a small-scale transparent Perspex model of a simplified two-storey building. The model was immersed upside-down in a large tank of fresh water, and heat gains were modelled as a continuous release of dense salt solution through a small opening in the floor of each storey. Salt solution is denser than the surrounding fresh water and creates a turbulent plume analogous to a thermal plume rising from a heat source in air. For clarity, the experimental observations will be described as if for a heat source in air. A food dye added to the salt solution distinguished it from the ambient (uncoloured) fresh water. A number of holes (diameter 2.0 cm) made in the ceiling of each storey connected to a common stack and acted as high-level outlet openings. An open-ended cylindrical tube (diameter 5.2 cm) represented the TDC and extended from the top of the model to close to the floor of each storey. An experiment began by supplying salt solution to the plumes. Following an initial transient period a steady flow was established and measurements of interface height and salinity (and hence temperature[3]) were made in each storey. When the experiment was illuminated from behind, the fraction of light transmitted through dyed regions could be directly related to the dye concentration, and hence to the fluid density. The motion of the ambient fluid was visualised by releasing a patch of neutrally buoyant dye into the region of interest.

The dye concentration image in figure 2, taken during a typical experiment, shows the geometry of the model building and the steady displacement flow established in both storeys, when there are heat gains in each. The common stack fills with warm air which descends to the level of the upper openings of the first storey. The heat gains and inlet/outlet areas in the two storeys are identical, however, the stratification height in the ground floor is greater than in the first floor. This implies a greater airflow rate and a reduced upper layer air temperature in the ground floor, and is a consequence of the enhanced stack at this floor.

By simulating heat gains in just one of the storeys, a displacement flow was set up only in the heated storey. However, due to the common stack, a ventilating flow was induced in the unheated storey — ambient air was drawn into this storey via its TDC and exhausted through the upper openings. A general upward movement of air through this storey was observed.

It was observed that ambient fluid at street level was not, in this quiescent environment, drawn into the TDCs. In contrast, street-level air was drawn into the model when low-level inlets were connected directly to the ambient.

4. DESIGN CURVES

For a specified building geometry and heating load, (6), (7) and (8) can be solved to give a prediction of stratification and *ACH*, within the assumptions of the model. In addition, the strength of the theoretical model is that it can aid an understanding of how the variable quantities scale with the controlling parameters of the flow. For example, the ventilation flow rate Q in single room ventilation driven by a heat source of power E is always equal to some scalar multiple of $(g\beta E/\rho c_p)^{1/3}H^{5/3}$, although the multiplication factor will vary with other flow parameters. It is therefore useful to explore how the *non-dimensional* variable $Q/(g\beta E/\rho c_p)^{1/3}H^{5/3}$ behaves. In this section, we present several design curves in non-dimensional variables.

As shown in §2.1, the addition of a TDC can reduce the effective opening area of a flowpath and so reduce the ventilation flow rate. However, expression (5) can be used to calculate by what amount the effective area of a particular storey A^*_j must be *increased* to compensate for the addition of a TDC, and so give no net reduction in flow rate:

$$\%\text{ increase in } A_j^* = \left[\left\{1-\left(A_j^*/A_{inlet}^*\right)^2 - \frac{L}{100D}\left(A_j^*/a_{TDC}\right)^2\right\}^{-1/2} - 1\right] \times 100. \qquad (9)$$

This is displayed graphically in figure 3, and it is evident that the necessary change is greater for large aspect ratio ($L/D \gg 1$) TDCs and increases as the inlet effective area decreases.

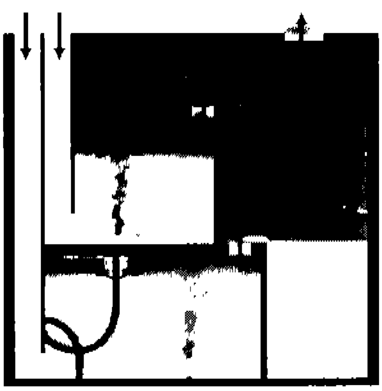

Figure 2. Inverted dye concentration image showing displacement flow in a two-storey enclosure. The common stack is on the RHS. The height of each storey $H = 14$ cm.

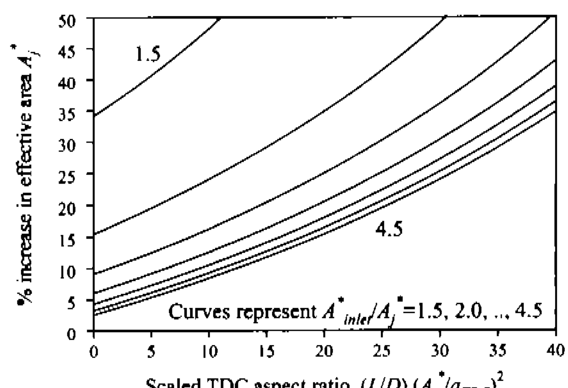

Figure 3. Percentage increase in A^*_j needed to compensate for the addition of a TDC of area a_{TDC}, aspect ratio L/D and inlet effective area A^*_{inlet}.

We now consider a possible design criterion for a multi-storey office building with a common stack. In our 'test' building, all the storeys are the same height $H_j = H$, and are subject to the same heating load $E_j = E$. One likely design requirement is for an equal ventilation flow rate on each storey, *i.e.* $Q_j = Q$. However, the lower storeys have a significantly larger driving force than the upper storeys due to the depth of the warm layer within the stack above them. The design solution is to choose larger opening areas for the higher storeys, to compensate for the reduced stack. From (6), it can be shown that, in this special case, the 'total' effective area A^*_{jtotal} for the flowpath through the j^{th} storey is

$$\frac{1}{A^*_{jtotal}} = \sqrt{\frac{1}{A^{*\,2}_{jTDC}} + \frac{1}{A^{*2}_j} + \frac{n^2}{A^{*\,2}_{stack}}} = \sqrt{\frac{1}{A^{*\,2}_{j(TDC+storey)}} + \frac{n^2}{A^{*\,2}_{stack}}}, \qquad (10)$$

as n equal streams of air must share the common stack exit. Here, $A^*_{j(TDC+storey)}$ is the combined effective area for the TDC and rooms on the j^{th} storey. From (6), the ratio of the total effective areas for the j^{th} and ground storeys required to give identical airflow rates through each storey is given by

$$\frac{A^*_{jtotal}}{A^*_{0total}} = \sqrt{\frac{M-h}{M-h-jH}}. \tag{11}$$

Assuming the required airflow rate Q is specified, the interface position can be found from (8b), and hence, the area ratio (11) determined. Figure 4(a) illustrates this ideal area ratio in a two-storey building as a function of the specified airflow rate for various chimney heights M/H. For small stack heights the area of the first floor vents need to be considerably larger than for the ground floor. As the stack height increases, the area ratio approaches 1 and is less dependent on the specified airflow rate. For a typical stack height of $M/H \approx n$, it can be shown from (11) that for a building with many storeys, *i.e.* $n \gg 1$,

$$\frac{A^*_{jtotal}}{A^*_{0total}} \approx \sqrt{\frac{1}{1-j/n}} \text{ and hence } \frac{A^*_{j(TDC+storey)}}{A^*_{0(TDC+storey)}} \approx \left[1-\frac{j}{n}-\frac{j}{n}\left(\frac{nA^*_{0(TDC+storey)}}{A^*_{stack}}\right)^2\right]^{-1/2} \tag{12a,b}$$

The effective area ratio (12b) is displayed in figure 4(b) for various stack outlet areas. As the stack outlet area decreases, the area ratio between the j^{th} and the ground floor rapidly increases, *e.g.* for the top storey of a 9-storey building (*i.e.* $j/n = 0.89$), reducing the dimensionless stack area from 4 to 3 more than doubles the required area ratio (from $A^*_{j(TDC+storey)}/A^*_{0(TDC+storey)} \approx 4.2$ to 9.0). The figure clearly shows how sharply the effective opening area $A^*_{j(TDC+storey)}$ must increase with each additional storey, until, at the highest storeys, it is no longer possible to match flow rates with the lowest storeys.

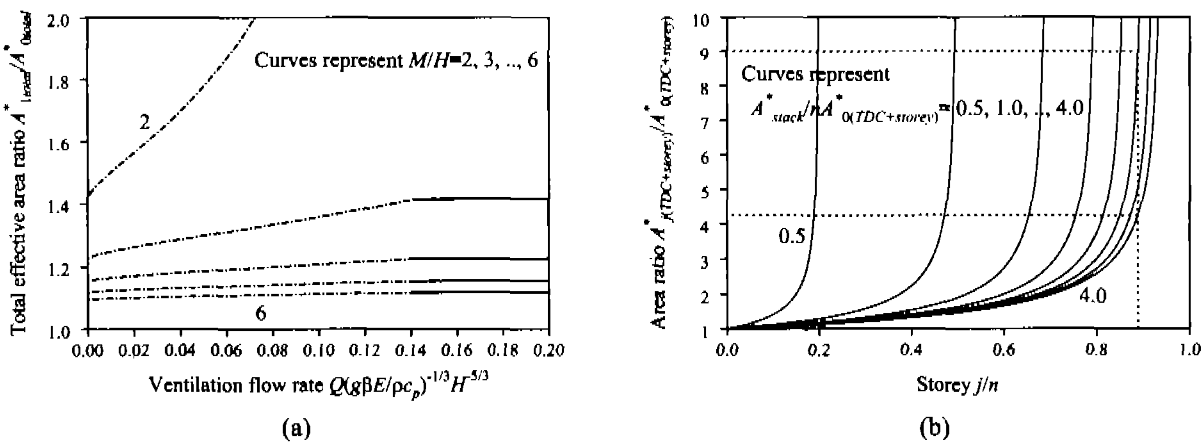

Figure 4. Ratio of effective areas necessary for equal ventilation flow rates on each story. (a) For a two-story building: $A^*_{1total}/A^*_{0total}$, for various M/H and Q. The solid lines indicate that the interface has reached the level of the upper openings. (b) For an n-storey building: $A^*_{j(TDC+storey)}/A^*_{0(TDC+storey)}$, for storey j/n and various dimensionless stack areas $A^*_{stack}/nA^*_{0(TDC+storey)}$.

5. CONCLUSIONS AND IMPLICATIONS TO BUILDING DESIGN

Stack-driven top-down displacement ventilation of multi-storey linked spaces has been examined through the development of a theoretical model and complementary small-scale laboratory experiments in water tanks. The experiments allow the ventilating flows on each storey to be clearly visualised, and demonstrate that top-down displacement ventilation in multi-storey spaces is a feasible strategy, which avoids the introduction of street-level contaminants into the building. The theoretical model provides a useful tool for predicting airflow rates and thermal stratification on each storey and in the stack. Furthermore, it may be used for selecting the opening areas required to give equal ventilation in each storey — a common requirement in multi-storey office spaces.

A potential drawback of top-down ventilation is the reduction in the natural driving forces due to frictional losses in the TDC. These losses may be approximated as a Re-independent loss coefficient giving an "effective area" for the TDC, and are shown to increase as the aspect ratio L(length)/D(diameter) of the TDC increases. Design guidelines are presented in the form of non-dimensional curves. These depict the required percentage increase in the vent area of a storey in order to compensate for the addition of a TDC, with no net loss in pressure.

In conclusion, we have shown that the modelling techniques developed for single-spaced enclosures may successfully be applied to more complex multi-compartment linked spaces. Work is currently in progress to further validate and extend the theory in order to model the direct ventilation of an atrium and to predict the stratification within it.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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NATURAL VENTILATION IN URBAN ENVIRONMENTS

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NATURAL VENTILATION IN URBAN ENVIRONMENTS

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Abstract

The present paper discusses issues related to the potential of natural ventilation techniques when applied to urban environment and in particular to buildings located in canyons. The paper discusses the specific phenomena related to air flow processes in urban canyons and presents some of the existing methods to calculate the wind speed distribution into the canyons.

Wind speed and temperature data have been collected through experiments carried out in ten different urban canyons presenting different characteristics, during summer 1997. The collected data have been used to evaluate the potential of natural ventilation in the ten canyons for single and cross ventilation configurations. It is found that, mainly during the day period, this is seriously reduced because of the important decrease of the wind speed inside the canyon. Air flow reduction may be up to ten times the flow that corresponds to undisturbed ambient wind conditions.

1. Introduction

Air flow around isolated buildings is well known. This is characterized by a bolster eddy vortex due to flow down the windward facade, while behind there is a lee eddy drawn into the cavity of low pressure due to flow separation from the sharp edges of the building top and sides, and further downstream is the building wake characterized by increased turbulence but lower horizontal speeds than the undisturbed flow.

The air flow patterns in urban canyons has received important attention during the last years. Natural Ventilation of buildings located in urban canyons is seriously reduced because of important decrease of the wind velocity inside the canyons.

Natural ventilation of buildings is due either to the wind forces or to the temperature difference between the indoor and outdoor environment or in a combination of both. Design of urban buildings to improve natural ventilation potential should consider the appropriate wind data and not routine meteorological observations collected in open fields. Also, the specific temperature regime in a canyon should be considered.

Knowledge of the air speed inside urban canyons is of high importance for passive cooling applications and especially for naturally ventilated buildings. Various methods, simplified or detailed have been proposed to calculate the wind speed inside a canyon.

Nakamura and Oke, (1989), have suggested the following simple linear form to calculate the mean horizontal wind speed, u_h, inside a canyon :

$$u_h = p\, u_{roof}$$

where p is a diminuation factor which depends on H/W and the measurement levels. They show for wind speeds up to 5 m/sec, with H/W = 1, and canyon centre and above roof measurements at heights of about 0.06 H and 1.2 H respectively that p=2/3. They also found that at smaller H/W, p approaches unity and shelter is lost.

Nicholson, (1975), has proposed a simple model to calculate the vortex circulation produced in street canyons when wind blows perpendicular to the street. By applying mass conservation techniques in the layer of air between the building's height, h_b, and the height at which the effects of the canyon on the overlying flow become negligible h', he proposed the following expression to calculate a representative speed of the upward-current of the vortex, w_m, :

$$w_m=2(h'-h_b)([u]_B - [u]_A) / W$$

where the '[ ]' symbol implies an average value across the layer from h_b to h', W is the canyon width and subscripts A and B refer to the locations.

According to Mills, (1993), h', may be calculated from :

$$h'=(0.15\ d + z_o\ h_b) / (0.15+z_o)$$

where d is the zero plane displacement. Also, values of z_o can be obtained from Tables.

Yamartino and Wiegand, (1986), has proposed a more representative canyon velocity calculated from the following expression :

$$V_c = (w_m{}^2 + u_m{}^2)^{0.5}$$

where w_m is wind speed due to vortex and u_m represents the along canyon flow at mid canyon height, (Mills, 1993), shows the relationship between V_c ,w_m and u_m with ambient wind azimuth.

Paciuk, (1975), based on wind tunnel experiments trying to identify the effects of building height and distances between buildings on the wind speed in the open spaces between the buildings, when the buildings are perpendicular to the wind direction, has developed a formula predicting the relative wind speed.

$$V_{r(u,h)} = 10 + (66(1-e^{-0.08h}))e^{-0.18D/W}$$

where :

$V_{r(u,h)}$ is the relative wind speed expressed as the percent of the wind at the same height well in front of the first line of buildings.

D is the distance traveled by the wind in meters, D=n(b+W)-0.5W

b is the depth of the buildings in meters

n is the serial number of the space, (downwind)

h is the height of the buildings in meters

and W is the width of the spaces between buildings in meters.

The formula indicates that as the wind approaches an urban area of long buildings with uniform height perpendicular to the wind direction, the initial turbulence 'agitation' over the first lines of buildings declines gradually toward a uniform wind speed in the spaces between the buildings. The analysis of the data suggested that the rate D/W is what determines the rate of drop in the air velocity towards an asymptotic value of about 10 % of the 'free' wind speed.

2. Air Flow In Urban Canyons

Urban canyons are characterized by three main parameters, H, the mean height of the buildings in the canyon, W, the canyon width, and L the canyon length. Given these parameters, the geometrical descriptors are limited to three simple measures. These are the ratio H/W, the aspect ratio, L/H and the building density j=Ar/A1 where Ar is the plan of roof area of the average building and A1 is the 'lot' area or unit ground area occupied by each building. Air flow phenomena associated with urban canyon are extensively discussed in (Santamouris, 1999)

When the predominant direction of the airflow is approximately normal (say ± 30 degrees), to the long axis of the street canyon, three type of air flow regimes are observed as a function of the building (L/H), and canyon (H/W), geometry. When the buildings are well apart, (H/W>0.05), their flow fields do not interact. At closer spacing, the wakes are disturbed and the flow regime is known as "Isolated Roughness Flow". When the height and spacing of the array combine to disturb the bolster and cavity eddies, the regime changes to one referred to as wake inteference flow. This is characterized by secondary flows in the canyon space where the downward flow of the cavity eddy is reinforced by deflection down the windward face of the next building downstream. At even greater H/W and density, a stable circulatory vortex is established in the canyon because of the transfer of momentum across a shear layer of roof height, and transition to a "skimming" flow regime occurs where the bulk of the flow does not enter the canyon.

Skimming regime is the most common in urban areas. Under these conditions the air flow in the canyon can been seen as a secondary circulation feature driven by the above roof imposed flow. If the wind speed out of the canyon is below some threshold value the coupling between the upper and secondary flow is lost, and the relation between wind speeds above the roof and within the canyon is characterized by a considerable scatter. According to many studies, carried out in almost symmetrical canyons where 1<H/W <1.4, the threshold value is between 1.5-2 m/sec. In all these studies higher wind speeds have been found to produce a stable vortex circulation within the canyon. For lower wind speeds thermal as well as mechanical influences may play an important role in the canyon circulation.

Parallel ambient flow generates a mean wind along the canyon axis, with possible uplift along the canyon walls as airflow is retarded by friction by the building walls and street surface. Regarding the relation between the free stream wind speed, U, and the along canyon velocity, v, it is reported that the along canyon wind component, v, in the canyon is directly proportional to the above roof along canyon component, through the constant of proportionality that is a function of approach flow azimuth.

The more common case in the urban environment, is that where the air flows at a certain angle relative to the long axis of the canyon. Unfortunately the existing research on this topic is considerably smaller compared to the scientific information for perpendicular and along the canyon flows, but it is known that when the flow above the roof is at some angle of attack to the canyon axis, a spiral vortex is induced along the length of the canyon, similar to a cork -screw type of action For intermediate angles of incidence to the canyon long axis, the canyon airflow is the product of both the transverse and parallel components of the ambient wind, where the former drives the canyon vortex and the later determines the along canyon streching of the vortex.

3. Experimental Procedure

Experiments have been performed in ten different canyon presenting dissimilar layout, orientation, anthropogenic heat and vegetation. The characteristics of the canyons are given in (Santamouris et al, 1997). Measurements have been performed between June and September 1997.

Three types of measurements have been performed :

a) Air temperature measurements. Miniature ambient air sensors have been used. The sensors were shielded inside a white painted wooden cylinder opened form the two parts to permit air circulation. The length of the cylinders was 20 cm while their internal and external diameter was 9 and 8 cm respectively. Sensors were completely protected from solar radiation. The cylindrical wooden boxes including the sensors have been fixed in the exterior facades of the buildings and in various heights in the canyon. The distance between the cylindrical box and the exterior wall was between 5 cm to 2 m, thus the temperature sensors were between 12 to 205 cm from the walls. Measurements were performed every 15 minutes. In some canyons, ambient temperature measurements were also performed using a digital hand thermometer at the mid width of the canyon on an hourly basis.

b) Surface temperature measurements. An infrared thermometer equipped with a laser beam has been used. The surface temperature of the exterior facades of the buildings, through a cross section of the canyon where the air temperature sensors were placed, is measured, (default section). Measurements are performed from the bottom to the top of both facades of the canyon using a step of 3-3.5 m. Additional measurements have been performed in some cross sections of the canyon where different than the default section materials are used. All measurements have been performed from the street level. The pavement and road temperature were measured as well at five different points along the width of the canyon. All measurements have been performed in an hourly basis during day and night.

c) Wind speed measurements. A three axis anemometer has been used to measure the three components of the wind speed inside the canyon. The anemometer was mounted on the exterior facade of a building in the canyon and in distance of 1 - 2 m from the wall. A cup anemometer has been also placed on the top of the canyon and in a distance of 6 m from its top level to measure the wind speed and direction out of the canyon. Measurements have been performed every 12 seconds.

The exact type of measurements performed in each of the ten canyons are given in (Santamouris et al, 1997).

4. Results

Experiments in ten deep canyons during the summer 1997, have shown that mean wind speed inside the canyon rarely exceeds 1 m/sec, independently of the free wind speed above the buildings. Figure 1, shows as an example, the variation of the air speed inside and outside a canyon having an aspect ratio close to one, during the whole experimental period.

In order to evaluate the natural ventilation potential of urban buildings, as well as its possible decrease because of the canyon related phenomena, simulations of the air flow processes have been carried for ten different canyons where wind speed and temperature data have been collected. Two configurations have been considered. A single as well as a cross ventilation configuration. A typical zone of 36 square meters, and 144 m3, having a window of 1.5 x 1.5 m, in each canyon facade is also considered.

Two types of simulations have been performed for each configuration. The first was based on the wind and temperature data measured inside the canyon, while the second one was based on the undisturbed temperature and wind speed measured over the buildings. Comparison of both simulation results should permit to assess the decrease of the natural ventilation potential in urban canyons.

Simulations have been performed using the AIOLOS natural ventilation simulation code, (Allard, 1998). The used software is well validated in the frame of the PASCOOL research project against a high number of experiments, (Limam K., Allard F., Dascalaki E, 1997).

Figures 2-3, give the air flow rate for the ten canyons, and for the single side and cross ventilation configuration respectively. The two flow rates, one when the ambient temperature and wind speed is used, and the second corresponding to the inside canyon measured data, are given. Analysis of the results permits to extract the following conclusions.

a) During the day time, when the ambient wind speed is considerably higher than wind speed inside the canyon and inertia phenomena dominate the gravitational forces, the natural ventilation potential in single and cross ventilation configurations is seriously decreased inside the canyon. In practice this happens when the ambient wind speed is higher than 4 m/sec. For single side ventilation configurations the air flow is reduced up to five times, while in cross ventilation configurations the flow is sometimes reduced up to ten times.

b) During the day time and when the ambient wind speed is lower than 3-4 m/sec, gravitational forces dominate the air flow processes. In this case the difference in wind speed inside and outside the canyon, do not play any important role and especially in single side configurations.

c) During the night time the ambient wind speed is seriously decreased and is comparable to the wind speed inside the canyon. In this case the air flow calculated for inside and outside the canyon is almost the same.

d) The calculated reduction of the air flow inside the canyon is mainly a function of the wind direction inside the canyon. When the ambient flow is almost vertical to the canyon axis, the flow inside the canyon is almost vertical and parallel to the window. In this case a much higher pressure coefficient correspond to the conditions outside the canyon, and thus a much higher flow is calculated when the ambient conditions are considered and inertia forces are dominating. When the ambient flow is parallel to the canyon axis, a similar flow is observed inside the canyon, thus the pressure coefficients are almost similar.

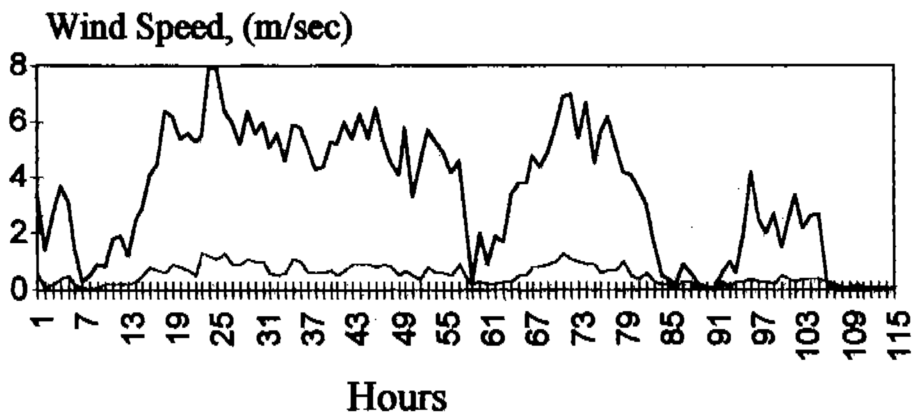

Figure 1. Measured wind speed inside and outside a representative urban canyon in Athens.

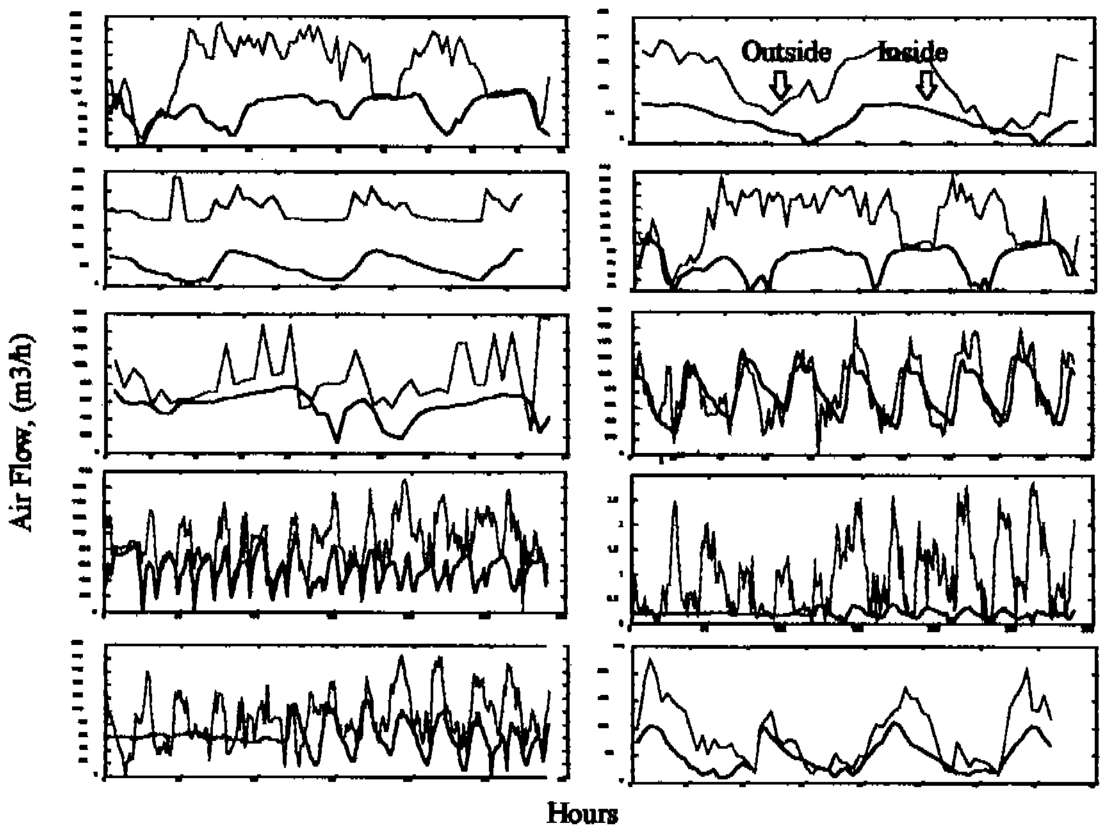

Figure 2. Air flow rates calculated for ten different canyons and for single side building configurations.

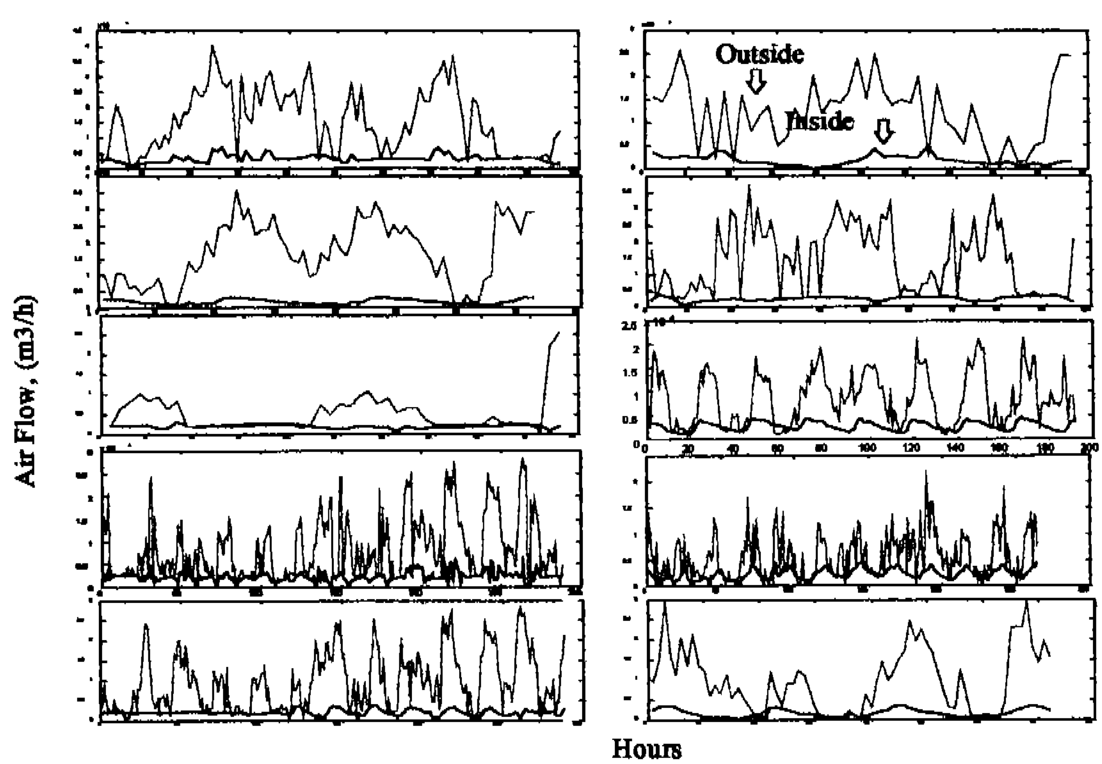

Figure 3. Air flow calculated for ten different canyons and for cross ventilation configurations

5. Conclusions

An assessment of the potential of natural ventilation in urban areas and in particular in urban canyons are presented. Calculations are based on air flow and temperature measurements taken in ten different canyons in Athens. A very serious reduction of the natural ventilation potential inside canyons is calculated especially during the day period. Compared to the air flow rates when undisturbed ambient meteorological data are used, air flow rates inside canyons may be reduced up to ten times. Further research work to better understand air flow processes in urban canyons is necessary.

6. Acknowledgment

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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NATURAL VENTILATION BY THERMAL BUOYANCY WITH SEVERAL OPENINGS AND WITH TEMPERATURE STRATIFICATION

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SYNOPSIS
Based on the fundamental flow equations, a set of formulas is derived for air velocities, temperature differences and ventilation rates in relation to number of openings, opening areas, net heat input, building geometry, and temperature stratification. The use of the formulas is illustrated on a three-storeyed office building.

LIST OF SYMBOLS

A	opening area	m^2
C_c	contraction coefficient	
C_d	discharge coefficient	
H	vertical distance between centre of highest and lowest placed opening	m
H_j	vertical distance between neutral pressure plane and centre of opening j	m
H_1^*	weighted height related to bottom opening	m
H_N^*	weighted height related to top opening	m
H_N^{**}	weighted height related to top opening taking temperature differences into account	m
$L_{j,n}$	vertical distance between the centres of opening j and n	m
N	number of openings (or opening levels)	
N_1	number of openings below neutral plane	
R	gas constant (= 287 J/kg K)	J/kg K
T	absolute temperature	K
c_p	specific heat capacity of air (= 1007 J/kg K)	J/kg K
g	gravity acceleration (= 9.82 m/s^2)	m/s^2
p	pressure	Pa
q_v	ventilation rate, volume flow	m^3/s
v	air velocity	m/s
ϵ	temperature factor	
ρ	air density	kg/m^3
ζ	resistance coefficient	
ψ	flow coefficient (= 1 + ζ)	
Φ_s	net heat input	W
$\Phi_{s\epsilon}$	modified net heat input	W
Δ	difference in pressure or temperature	

Subscripts

c	contracted
i	indoor
m	mean
o	outdoor
1	lowest inlet
N	highest outlet

1. INTRODUCTION
The formulas available for natural ventilation by thermal buoyancy are usually derived for rooms with two openings and an assumed uniform indoor temperature. However, in practice temperature stratification always occurs, and you have more than two openings when buildings of several storeys are considered.

Few references are available where the temperature stratification is taken into account, but they concern only one-storeyed buildings.

The theory for natural ventilation by thermal buoyancy in buildings with uniform indoor temperature and with two or several opening levels is treated in detail in Andersen (1995). Thermal buoyancy with indoor temperature stratification and with two openings is treated in Andersen (1998).

This paper describes the theory for natural ventilation by thermal buoyancy in buildings with several opening levels and with indoor temperature stratification.

2. THEORETICAL CONSIDERATIONS

By thermal buoyancy, the indoor and outdoor pressures are equal at a certain level called the neutral plane. Through the openings below the neutral plane, air flows inward due to a negative indoor pressure, and through the openings above the neutral plane, air flows outward due to a positive indoor pressure. By using the mass balance equation, the position of the neutral plane can be determined.

2.1 Uniform temperature

For two openings and with uniform indoor temperature the position of the neutral plane can with good approximation be determined by (Andersen, 1995):

$$H_1 = \frac{H}{1 + (A_1/A_2)^2} \tag{1}$$

where H_1 is the vertical distance from the centre of inlet to the neutral plane, H is the vertical distance between the centres of inlet and outlet, and A_1 and A_2 are the areas of inlet and outlet, respectively.

With uniform temperature and several openings you introduce a "weighted" height H_1^* determined by (Andersen, 1995):

$$H_1^* = H_1 \left[1 + \sum_2^{N_1} \frac{C_{dj}\, A_j\, \sqrt{H_j}}{C_{d1}\, A_1\, \sqrt{H_1}} \right]^2 = H_1 \left[1 + \sum_2^{N_1} m_j n_j q_j^{1/2} \right]^2 \tag{2}$$

where H_1 is the distance from the lowest inlet to the neutral plane, and where the neutral plane position is found by solving the mass balance equation iteratively. Further, N_1 is the number of openings below the neutral plane and:

$$m_j = C_{dj}/C_{d1}, \text{ and } n_j = A_j/A_1, \text{ and } q_j = H_j/H_1 = (H_1 - L_{1,j})/H_1$$

where again C_d is the discharge coefficient and $L_{1,j}$ is the vertical distance between the bottom opening and opening number j.

2.2 Temperature stratification and several openings

By temperature stratification, the neutral plane moves upward compared to the position by uniform temperature. But the pressure differences and thereby the air velocities in the openings stay approximately unchanged because of the increasing temperature difference above the neutral plane and the decreasing temperature difference below the neutral plane. The approximation is best for the top and bottom openings and less good for the openings closest to neutral plane. However, these openings contributes the least to the ventilation rate. Therefore, the position of the neutral plane can be found approximately from the mass balance equation assuming uniform indoor temperature.

An approximate solution of the basic flow equations is shown in Table 1. The formulas in column I of the table are identical to those valid for a uniform temperature equal to the mean indoor temperature T_{im}, and with a temperature difference $\Delta T = \Delta T_m = T_{im} - T_o$. The weighted height H_1^* is determined by Equation 2.

In column II, a modified net heat input $\Phi_{s\epsilon}$ is the independent variable (cf Appendix A):

$$\Phi_{s\epsilon} = \frac{\Phi_s}{\Delta T_N / \Delta T_m} = \frac{\Phi_s}{\epsilon} \tag{3}$$

where Φ_s is the net heat input, and ΔT_N is the temperature difference at the top opening. The factor ϵ takes into account that a smaller ventilation rate is required compared to the uniform temperature situation because of the bigger temperature difference at the outlet. Similar to H_1^* one has:

$$H_N^* = H_N \left[1 + \sum_{N_1+1}^{N-1} \frac{C_{aj}\, A_j \sqrt{H_j}}{C_{aN}\, A_N \sqrt{H_N}} \right]^2 = H_N \left[1 + \sum_{N_1+1}^{N-1} m_j n_j q_j^{1/2} \right]^2 \tag{4}$$

where H_N is the vertical distance from the neutral plane to the centre of the top opening, N is the total number of openings (or opening levels), and:

$m_j = C_{dj}/C_{dN}$, and $n_j = A_j/A_N$, and $q_j = H_j/H_N = (H_N - L_{1j})/H_N$

Moreover a new weighted height H_N^{**} is introduced defined by (cf Appendix A):

$$H_N^{**} = H_N \left[1 + \sum_{N-N_1}^{N-1} \frac{C_{dj}\, A_j\, H_j^{1/2} \Delta T_j}{C_{dN}\, A_N\, H_N^{1/2} \Delta T_N} \right]^2 = H_N \left[1 + \sum_{N-N_1}^{N-1} m_j n_j q_j^{1/2} r_j \right]^2 \tag{5}$$

where $r_j = \Delta T_j / \Delta T_N$.

The resistance and the contraction properties of the openings are given by the discharge coefficient C_d, and a flow coefficient ψ is introduced. The relationship between the coefficients is (Andersen, 1996):

$$C_d = C_c\, C_v = C_c/\psi^{1/2} = C_c/(1 + \zeta)^{1/2}$$

2.2.1 Four openings. For a building with four openings as shown in Figure 1, the neutral plane position is determined by the mass balance equation. The indoor temperature is assumed to be uniform, the neutral plane position is assumed to be somewhere between openings 2 and 3 and the air velocities are dependent on the square root of the distance to the neutral plane, cf Table 1. One gets:

$$A_1\, H_1^{1/2} + A_2\, (H_1 - L_{12})^{1/2} - A_3\, (H - L_{34} - H_1)^{1/2} - A_H\, (H - H_1)^{1/2} = 0 \tag{6}$$

Table 1. Formulas by serveral openings and temperature stratification.

Row no.		Column I Formulas based on the mean temperature difference ΔT_m [1) 3)]	Column II Formulas based on the modified net heat input Φ_{se} [1) 2) 3) 4) 5) 6) 7)]
1	Inlets pressure dif., Δp_j (Pa)	$\rho_u \, \Delta T_m \, g \, H_j \, / \, T_{im}$	
2	air velocity, $v_{k,J}$ (m/s)	$\left[\dfrac{2 \, \Delta T_m \, g \, H_j}{\psi_j \, T_{im}}\right]^{1/2}$	$0{,}038 \left[\dfrac{\Phi_{se}}{C_{dN} \, A_N}\right]^{1/3} \left[\dfrac{1}{H_N^{**}}\right]^{1/6} \left[\dfrac{H_j}{\psi_j}\right]^{1/2}$
3	Outlets pressure dif., Δp_j (Pa)	$\rho_i \, \Delta T_m \, g \, H_j \, / \, T_u$	
4	air velocity, $v_{k,J}$ (m/s)	$\left[\dfrac{2 \, \Delta T_m \, g \, H_j}{\psi_j \, T_u}\right]^{1/2}$	$0{,}039 \left[\dfrac{\Phi_{se}}{C_{dN} \, A_N}\right]^{1/3} \left[\dfrac{1}{H_N^{**}}\right]^{1/6} \left[\dfrac{H_j}{\psi_j}\right]^{1/2}$
5	Temperature dif., ΔT_m (K eller °C)		$7{,}3 \cdot 10^{-5} \, T_o \left[\dfrac{\Phi_{se}}{C_{dN} \, A_N}\right]^{2/3} \left[\dfrac{1}{H_N^{**}}\right]^{1/3}$
6	Ventilation rate, q_V (m³/s)	$C_{d1} \, A_1 \left[\dfrac{2 \, \Delta T_m \, g \, H_1^{*}}{T_{im}}\right]^{1/2}$	$0{,}039 \, (\Phi_{se})^{1/3} (C_{dN} A_N)^{2/3} (1/H_N^{**})^{1/6} (H_N^{*})^{1/2}$
7	Top outlet area, A_N (m²)		$6{,}2 \cdot 10^{-7} \, \dfrac{\Phi_{se}}{C_{dN}} \left[\dfrac{1}{H_N^{**}}\right]^{1/2} \left[\dfrac{T_u}{\Delta T_m}\right]^{3/2}$
8			$140 \, \dfrac{q_V^{3/2}}{(\Phi_{se} \, H_N^{**})^{1/2} \, C_{dN}}$ [6)]
9	Bottom inlet area, A_1 (m²)		$\dfrac{q_{V_{II6}}}{C_{d1}} \left[\dfrac{T_{im}}{2 \Delta T_m \, g \, H_1^{*}}\right]^{1/2}$ [7)]

[1)] ΔT_m and T_{im} are the temperature difference and the indoor temperature, respectively, equidistant from top and bottom openings.

[2)] Φ_{se} is determined by Equation 3.

[3)] H_1^{*} is determined by Equation 2.

[4)] H_N^{*} is determined by Equation 4.

[5)] H_N^{**} is determined by Equation 5.

[6)] q_V is a required ventilation rate.

[7)] $q_{V,II6}$ is determined by Formula II6 of the table. (Column II, row 6)

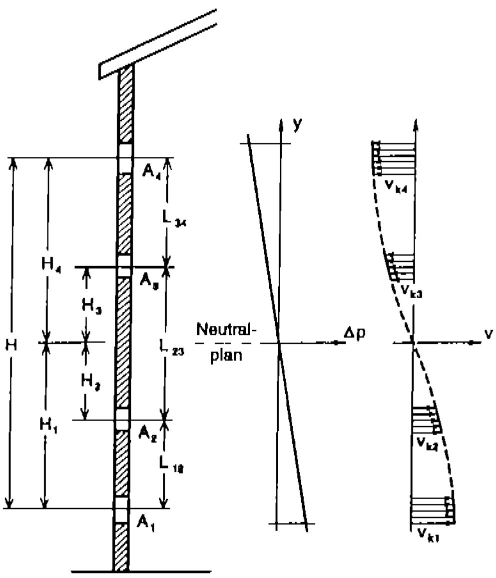

Figure 1. Pressure differences and air velocities by natural ventilation through four openings.

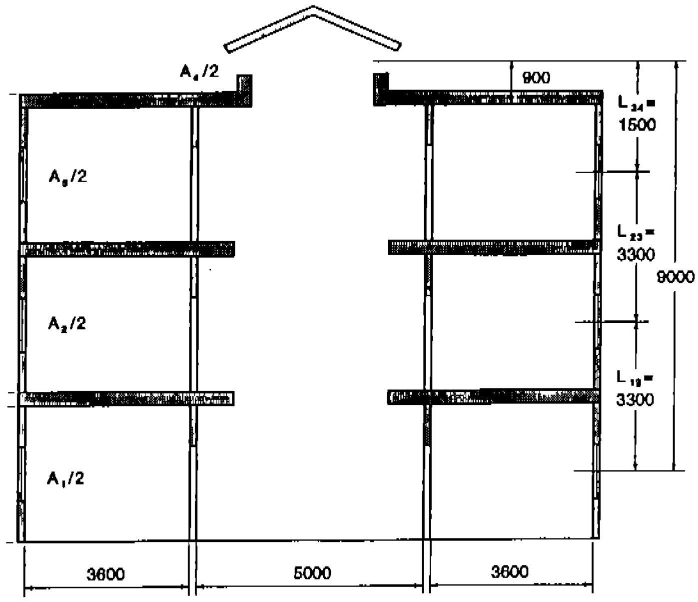

Figure 2. Cross section through three-storyed office building.

where the distance H_1 is the only unknown quantity. The weighted height H_1^* is determined by:

$$H_1^* = H_1 \left[1 + \frac{C_{d2}\, A_2\, \sqrt{H_2}}{C_{d1}\, A_1\, \sqrt{H_1}} \right]^2 = H_1\left(1 + m_2\, n_2\, q_2^{1/2}\right)^2 \tag{7}$$

where H_1 is found from Equation 6, and where:

$m_2 = C_{d2}/C_{d1}$, and $n_2 = A_2/A_1$, and $q_2 = H_2/H_1 = (H_1 - L_{12})/H_1$

The weighted height H_4^{**} is determined by:

$$H_4^{**} = H_4 \left[1 + \frac{C_{d3}\, A_3\, H_3^{1/2} \Delta T_3}{C_{d4}\, A_4\, H_4^{1/2} \Delta T_4} \right]^2 = H_4\left(1 + m_3\, n_3\, q_3^{1/2} r_3\right)^2 \tag{8}$$

where $H_4 = H - H_1$ and where:

$m_3 = C_{d3}/C_{d4}$, and $n_3 = A_3/A_4$, and $q_3 = H_3/H_4 = (H_4 - L_{34})/H_4$, and $r_3 = \Delta T_3/\Delta T_4$

3. EXAMPLE. OFFICE BUILDING

Figure 2 shows the cross section of a three-storeyed office building with an atrium at the centre. It should be ventilated by natural ventilation and the design situation is a calm summer day with an out-door temperature of 25°C and a required indoor temperature in the offices not exceeding 29°C. A 3 m long section of the building is considered.

The indoor temperature stratification is assumed to have a vertical gradient of 0.2 K/m and the openings are sharp-edged with a discharge coefficient of $C_d = 0.62$. Further the net heat input is calculated to $\Phi_g = 2250$ W.

There are openings in four levels, and the necessary opening areas are requested for the following three situations:

- Equally big opening areas at each opening level.
- Graduated opening areas for getting fresh air into all offices.
- Graduated opening areas for getting the same amount of fresh air into all offices.

3.1 Equally large opening areas

With the window opening area of each office equal to $A_1/2$, and with the opening area at the top of the atrium being A_1, the opening area at each of the four opening levels is equal to A_1.

It can be assumed that the neutral plane is placed somewhere between the openings at second and third floors. The position can then be determined by the following mass balance equation, cf Equation 6, with $A_1 = A_2 = A_3 = A_4$:

$$H_1^{1/2} + (H_1 - 3.3)^{1/2} - (6.6 - H_1)^{1/2} - (9.0 - H_1)^{1/2} = 0 \tag{9}$$

The equation is satisfied for $H_1 = 4.7$ m and this is in accordance with the previously assumed position between openings 2 and 3. With an indoor temperature at third floor of 25 + 4 = 29°C, one gets $\Delta T_m = 3.6$ K and $\Delta T_4 = 4.5$ K, so that $\epsilon = 4.5/3.6 = 1.25$ and $\Phi_{g6} = 2250/1.25 = 1800$ W (cf Equation 3).

The height H_4^{**} is determined by Equation 8 with $H_4 = 9.0 - 4.7 = 4.3$ m, $m_3 = n_3 = 1.0$, $q_3 = 1.9/4.3 = 0.44$, and $r_3 = 4.0/4.5 = 0.89$. One gets:

$$H_4^{**} = 4.3\,(1 + 0.44^{1/2} \cdot 0.89)^2 = 10.9 \text{ m}$$

The opening areas are determined by Formula II 7 (column II, row 7) in Table 1:

$$A_4 = A_1 = 6.2 \cdot 10^{-7}\ \frac{1800}{0.62} \left[\frac{1}{10.9}\right]^{1/2} \left[\frac{298}{3.6}\right]^{3/2} = 0.41 \text{ m}^2$$

The required opening area of the windows then becomes $A_1/2 = A_4/2 = 0.21$ m^2.

3.2 Fresh air in all offices

By equally big opening areas, one gets an inward air flow in the window openings at the first and second floors, and outward flow in the window openings at third floor. Fresh air into the third floor offices can be obtained by moving the neutral plane up above the third floor openings.

With the neutral plane at third floor ceiling level, one gets $H_4^{**} = H_4 = 1.0$ m. The distance from ceiling to centre of the top opening is increased by 0.1 m to take the bigger top opening area into account. $Q_{g6} = 1800$ W and $\Delta T_m = 3.6$ K are unchanged. The required opening area in the top of the atrium is then determined by:

$$A_4 = 6.2 \cdot 10^{-7} \frac{1800}{0.62} \left[\frac{1}{1.0}\right]^{1/2} \left[\frac{298}{3.6}\right]^{3/2} = 1.4 \text{ m}^2$$

or 0.7 m_2 for each of the two hatches on the atrium roof.

The opening areas of the windows can be determined by Formula II 9 in Table 1, when the ventilation rate has been determined by Formula II 6 with $H_N^{**} = H_N^* = 1.0$ m, and the weighted height H_1^* by Equation 2. For the ventilation rate one gets:

$$q_v = 0.039 \cdot 1800^{1/3} (0.62 \cdot 1.4)^{2/3} \cdot 1.0^{1/6} \cdot 1.0^{1/2} = 0.43 \text{ m}^3/\text{s}$$

For H_1^* one gets, when $H_1 = 9.0 - 1.0 = 8.0$ m, $m_2 = m_3 = n_2 = n_3 = 1.0$, $q_2 = 4.8/8.0 = 0.60$, and $q_3 = 1.5/8.0 = 0.19$:

$$H_1^* = H_1 (1 + m_2 n_2 q_2^{1/2} + m_3 n_3 q_3^{1/2})^2 = 8.0 (1 + 0.60^{1/2} + 0.19^{1/2})^2 = 39.1 \text{ m}$$

Finally one gets:

$$A_1 = \frac{0.43}{0.62} \left[\frac{302}{2 \cdot 3.6 \cdot 9.82 \cdot 39.1}\right]^{1/2} = 0.23 \text{ m}^2$$

or about 0.12 m^2 per window.

3.3 Equal amounts of fresh air

With equal big inlet openings one gets the biggest air flow rate through the windows at first floor. Equally big air flow rates can be obtained by graduating the inlet areas.

The air flow rate through each of the six window openings should be $q_{vw} = 0.43/6 = 0{,}07$ m^3/s. The neutral plane shall still be placed at the level of third floor ceiling.

The air velocities in the windows can be determined by Formula I 2 in Table 1 where $\psi_j = 1 + \varsigma_j = 1 + 0.1 = 1.1$, ($\varsigma_j$ being the resistance coefficient) and one gets:

$$v_{cj} = \left[\frac{2 \cdot 3.6 \cdot 9.82 \cdot H_j}{1.1 \cdot 298}\right]^{1/2} = 0.46\, H_j^{1/2} \qquad (11)$$

For one window (with index j) with the contraction coefficient assumed to be $C_c = 0.65$, the required opening area is determined by:

$$A_{jw} = \frac{q_{vw}}{C_c\, v_{cj}} = \frac{0.07}{0.65 \cdot 0.46\, H_j^{1/2}} = \frac{0.23}{H_j^{1/2}}$$

For each floor, one gets the following opening areas per window:

First floor, $H_j = 8.1$ m: $A_{1w} = 0.08$ m^2
Second floor, $H_j = 4.8$ m: $A_{2w} = 0.11$ m^2
Third floor, $H_j = 1{,}5$ m: $A_{3w} = 0.19$ m^2

3.4 Summarizing

The calculation results are summarized in Table 2. The table shows the temperatures occurring at the different floors and at roof level as well as the required opening areas by the three different conditions considered in the calculations.

Table 2. Required opening areas in three-storyed office building by various conditions.

Level	1 floor	2 floor	3 floor	Roof
Temperature difference, ΔT_j, K	2.5	3.4	4.0	4.5
Required opening areas by:				
equal opening areas, m^2	0.21	0.21	0.21	0.21
fresh air in all offices, m^2	0.12	0.12	0.12	0.70
equal amount of fresh air, m^2	0.08	0.11	0.19	0.70

APPENDIX A

The theoretical air velocity (i.e. no friction loss) in the openings is determined by:

$$v_{theo\,j} = \left[\frac{2\Delta T_m\, g\, H_j}{T_{im}}\right]^{1/2} \tag{A1}$$

By stationary conditions the net heat input is equal to the heat removed by the air flow through the outlets:

$$\Phi_t = \sum_{N_1+1}^{N} c_p\, \rho_{im}\, q_v\, \Delta T_l = \sum_{N_1+1}^{N} c_p\, \rho_{im}(C_{dj}\, A_j\, v_{theo\,j})\Delta T_j$$

$$= \sum_{N_1+1}^{N} c_p\, \rho_{im}\, C_{dj}\, A_j\, (2\,\Delta T_m\, g\, H_j/T_u)^{1/2}(\Delta T_j/\Delta T_N)\Delta T_N$$

$$= c_p \rho_{im} C_{dN} A_N (2\,\Delta T_m\, g\, H_N/T_u)^{1/2}\Delta T_N \left[1 + \sum_{N_1+1}^{N-1} \frac{C_{dj}}{C_{dN}}\frac{A_j}{A_N}\left[\frac{H_j}{H_N}\right]^{1/2}\frac{\Delta T_j}{\Delta T_N}\right]^2$$

$$= c_p\, \rho_{im}\, C_{dN} A_N \left(2\Delta T_m\, g\, H_N^{**}/T_u\right)^{1/2}(\Delta T_N/\Delta T_m)\Delta T_m$$

or:

$$\frac{\Phi_t}{\Delta T_N/\Delta T_m} = \frac{\Phi_t}{\epsilon} = \Phi_{tc} = c_p\, \rho_{im}\, C_{dN} A_N \left(2\,\Delta T_m\, g\, H_N^{**}/T_u\right)^{1/2}\Delta T_m \tag{A2}$$

This equation can be solved with regard to ΔT_m. When using $\varrho_{im} = p_{im}/(R\, T_{im})$ one gets:

$$\Delta T_m = \left[\frac{R}{c_p p_{im}}\right]^{2/3} \left[\frac{1}{2g}\right]^{1/3} \left[\frac{T_{im}}{T_o}\right]^{2/3} \left[\frac{\Phi_{st}}{C_{dN} A_N}\right]^{2/3} \left[\frac{1}{H_N^{**}}\right]^{1/3} T_o$$

By introducing R = 287 J/kgK, c_p = 1007 J/kgK, p_{im} = 101300 Pa, g = 9.82 m/s^2, and by assuming $T_{im}/T_o \cong 1.03$, one finally gets:

$$\Delta T_m = 7.5 \cdot 10^{-5}\, T_o \left[\frac{\Phi_{st}}{C_{dN} A_N}\right]^{2/3} \left[\frac{1}{H_N^{**}}\right]^{1/3} \tag{A3}$$

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

VENTILATION PERFORMANCES IN NEW BELGIAN DWELLINGS

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SYNOPSIS

A systematic analysis of recently constructed dwellings in the Flemish Region has been undertaken within the SENVIVV-project (1995-1998) [1]. In total 200 dwellings have been examined in detail. The study involved various aspects: energy related building data (thermal insulation level, net heating demand, installed heating power, etc.), indoor climate (temperature levels in winter and summer), building airtightness, ventilation, appreciation of the occupants, etc. This paper focuses on the findings concerning ventilation facilities in the investigated dwellings. The following aspects were investigated:

- Required air flow rates according to the Belgian ventilation standard;
- Presence of ventilation facilities in the investigated dwellings;
- Air flow rates of the installed ventilation devices;
- Facilities for intensive ventilation.

The study revealed that the prescriptions in the ventilation standard are usually not well adopted.

1 INTRODUCTION

Each year about 35 000 new dwellings are constructed in the Flemish region (northern half of Belgium). During the nineties a standard related to ventilation and building regulations related to thermal insulation came into force. As little was known about the building practice and the compliance with the new regulations, a thorough study [1] was set up to examine the energetic performances of new dwellings. From 1995 to 1998, 200 representatively selected houses and multifamily buildings were investigated in detail. This paper discusses the findings concerning the ventilation facilities in the investigated dwellings.

2 BELGIAN VENTILATION STANDARD

2.1 General

The Belgian standard NBN D50-001 (March 1992) [2] describes the requirements for ventilation in dwellings. In the Flemish region this standard is not compulsory (except for social housing), but every standard has to be seen as a rule of good practice, and as a consequence the performances have to be comparable with the requirements of the standard.

The philosophy of the standard is that a good ventilation consists of different aspects, represented in Figure 1. The items in a grey box are discussed in this paper.

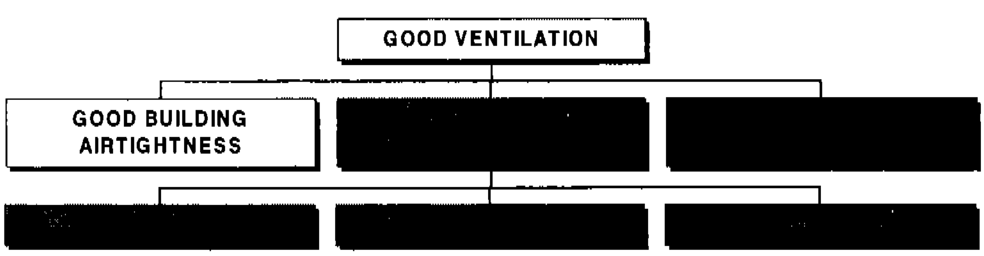

Figure 1: Elements for a good ventilation strategy, according to the Belgian ventilation standard

2.2 Facilities for basic ventilation

The ventilation standard prescribes which air flow rates have to be achievable in different types of rooms: these are called the nominal air flow rates (see Table 1).

Basic requirement: for all rooms	1 l/s.m² floor area (3.6 m³/h.m²)	
Additional requirement:	*Minimum*	*Can be limited to...*
Living rooms	21 l/s (75 m^3/h)	42 l/s (150 m^3/h)
Bedroom, study, etc.	7 l/s (25 m^3/h)	10 l/s (36 m^3/h.person)
Bathroom, washhouse, separate kitchen, etc.	14 l/s (50 m^3/h)	21 l/s (75 m^3/h)
Open kitchen (combined with living room)	21 l/s (75 m^3/h)	----
Toilet	7 l/s (25 m^3/h)	----

Table 1: Required air flow rates according to the Belgian ventilation standard NBN D50-001

Ventilation can be achieved as well in a mechanical way as in a natural way. In the case of natural ventilation the nominal air flow is defined as the air flow rate through the device at a pressure difference of 2 Pa. The crucial point is that the philosophy of the standard is always respected: supply in 'dry' rooms (living rooms, bedrooms, studies,...), exhaust from the 'humid' rooms (bathroom, toilet, washhouse, kitchen,...) and transfer from the dry rooms to the humid rooms through registers in doors (also a gap under the door is accepted) or inner walls (possibly via corridors). The standard requires that the sum of the transfer openings of a room has a nominal air flow rate (at 2 Pa) of 14 l/s for a kitchen and 7 l/s for all other rooms. The ventilation standard gives also requirements for the ventilation of garages, attics, basements, etc. These are not mentioned here.

2.3 Facilities for intensive ventilation

During cooking, painting and other specific activities the air flow rates provided by the facilities for basic ventilation are not sufficient to guarantee an acceptable indoor air quality. Also during periods with a lot of sunshine additional ventilation can help to avoid excessive overheating. Therefore the Belgian ventilation standard requires the presence of facilities to increase the air flow rate during certain periods of time.

- Kitchens without external doors or windows that can be opened: intensive ventilation (cooker hood) of at least 200 m³/h.
- Bedrooms, studies, playrooms, living rooms and kitchens: requirements for the minimal area of windows and doors that can be opened.
  - ⇨ Openings in only one facade: sum of the openings = at least 6.4% of the floor area
  - ⇨ Openings in two facades: sum of the openings = at least 3.2 % of the floor area and a uniform distribution of the openings over both facades (at least 40 % per facade).

3 RESULTS OF THE STUDY AND DISCUSSION

3.1 Required air flow rates at house level

The required air flow rates according to the Belgian ventilation standard were calculated for all rooms of the investigated dwellings. For each house the total supply air flow rate and the total exhaust air flow rate were calculated. The result is represented in Figure 2.

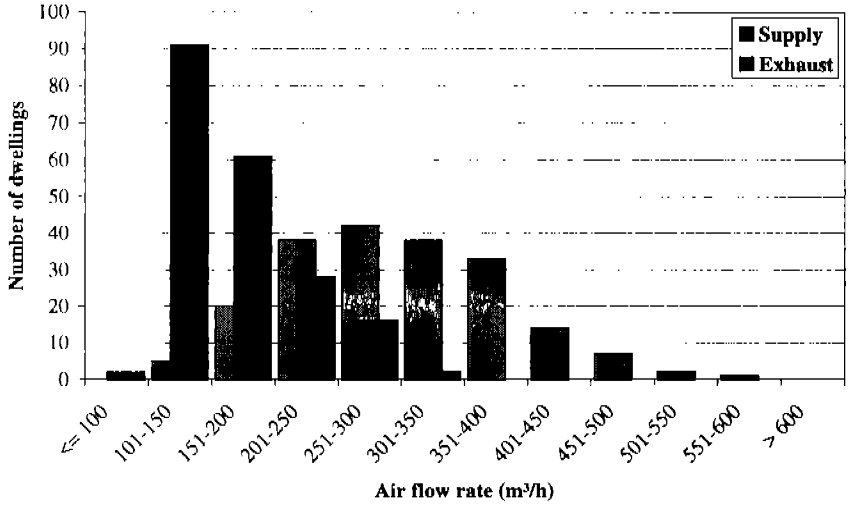

Figure 2: Histogram of the total supply and exhaust air flow rates per dwelling.

The following remarks can be made:

- ♦ Except for 5 small apartments the total required supply air flow rate is always higher than the total required exhaust air flow rate.
- ♦ The average exhaust air flow rate per dwelling is about 180 m³/h, while the average supply air flow rate is about 300 m³/h.
- ♦ On the base of the supply air flow rate an average air change rate of 0.60 h^{-1} is found. Large variations are possible: between 0.30 h^{-1} for bigger houses and 0.95 h^{-1} for small dwellings and apartments. The relation between the required air change rate and the volume of a house is illustrated clearly in Figure 3.

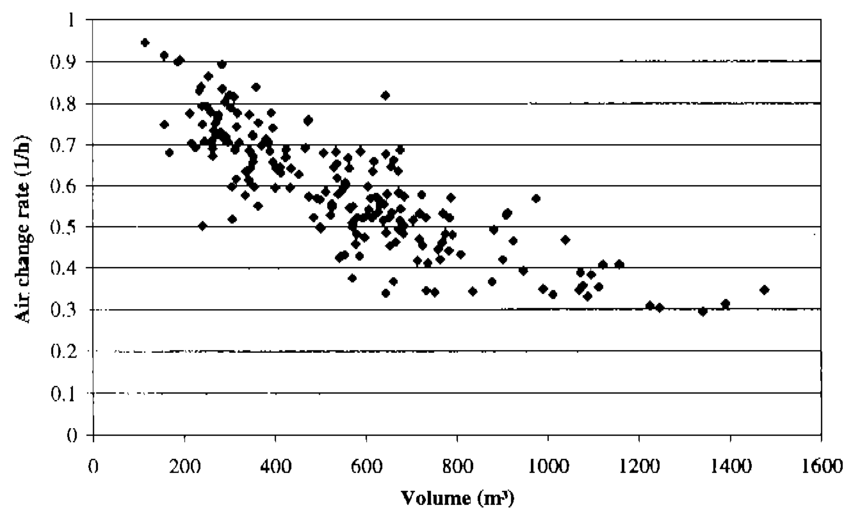

Figure 3: Relation between the air change rate of a dwelling (calculation based on the required supply air change rates according to NBN D50-001) and the volume.

3.2 Ventilation facilities for basic ventilation in the investigated dwellings

3.2.1 Presence of ventilation devices

Figure 4 shows the presence of ventilation facilities in the different room types of the investigated houses.

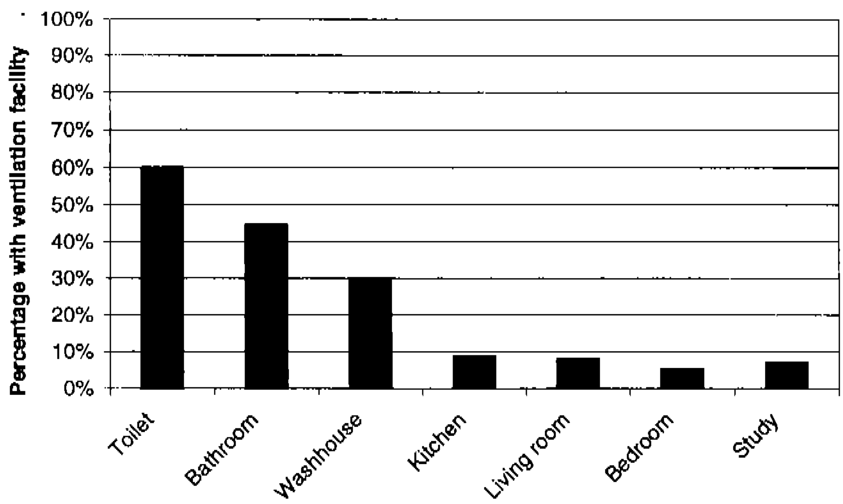

Figure 4: Presence of facilities for basic ventilation in the investigated dwellings

The following observations can be made:

- The installation of ventilation devices seems to be most common in humid rooms: toilet, bathroom and washhouse. On the contrary, dry rooms are rarely equipped with ventilation devices (less than 10%). However, measurements revealed that certain types of rooms are often very airtight (especially bedrooms and bathrooms) [3], which makes the presence of ventilation facilities essential for a good IAQ.
- In dry rooms the ventilation facilities are nearly always window registers (= natural ventilation).
- Small windows are not taken into account as ventilation device.
- It is important to mention that the devices that are taken into account don't always comply with the prescriptions from the ventilation standard because the nominal air flow rate is too high or too low (see further).
- Humid rooms are sometimes equipped with registers in the vertical wall or in the window, although this is not in agreement with the philosophy of the Belgian ventilation standard (*"supply in the dry rooms and exhaust from the wet rooms"*), because exhaust is only guaranteed by mechanical extraction or by natural ventilation through a vertical duct with an outlet on the roof.
- Most of the kitchens are equipped with a cooker hood (see further). Strictly, a cooker hood could be used for basic ventilation, but normally they will only be switched on for intensive ventilation due to high noise levels and high air flow rates. Therefore, cooker hoods are only taken into account in Figure 4 if they are connected to a duct with continuous extraction. In this case the cooker hood will usually be equipped with a flap and have an extraction in the closed position.
- About 1/3 of the exhaust ventilation devices in the humid rooms are mechanical systems.

For the exhaust systems in apartments a distinction has to be made between centralised (= several apartments connected to 1 system) and individual (= per apartment) exhaust. The distribution is represented in Figure 5. It can be seen that the most common system is the centralised natural exhaust. Only a minority of the buildings doesn't have any exhaust system.

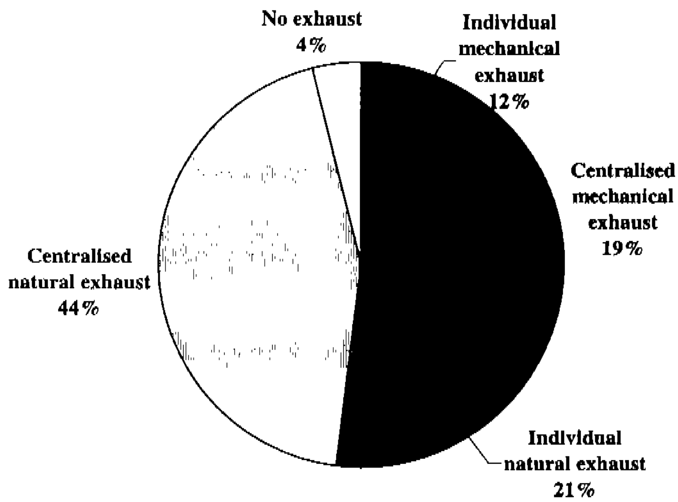

Figure 5: Exhaust in apartment buildings

To guarantee a good air transfer from the dry rooms to the humid rooms there should be transfer openings in the inner doors (or walls).

- In only 12 % of the investigated dwellings transfer registers were found, but even then only in a limited number of rooms (especially in bathroom, storage and toilet).
- The only other way to guarantee a good air transfer is by leaving a gap under the door. In all investigated dwellings the free space under the door was measured. Only in 2% of the cases the gap was higher than 1 cm, which indicates that in the majority of the rooms there are no transfer openings in compliance with the ventilation standard.

3.2.2 Nominal air flow rate

3.2.2.1 Dry rooms

In Figure 6 the nominal air flow rates of some rooms equipped with window registers are compared with the required air flow rates according to NBN D50-001.
The nominal air flow rates of the window registers are calculated on the basis of information from the manufacturer and the register length.
The ventilation standard mentions that in all rooms the nominal air flow rate of the installed natural ventilation devices has to be lower than the double of the required air flow rate.
It can be seen that an important number of rooms don't comply with this rule, especially rooms with a 'low' required air flow rate. It seems that in a majority of these rooms more than one register has been installed, which means that compliance with the requirements could be achieved by simply reducing the number of registers.
Only in a small number of cases the installed registers have a nominal air flow rate that is too small.

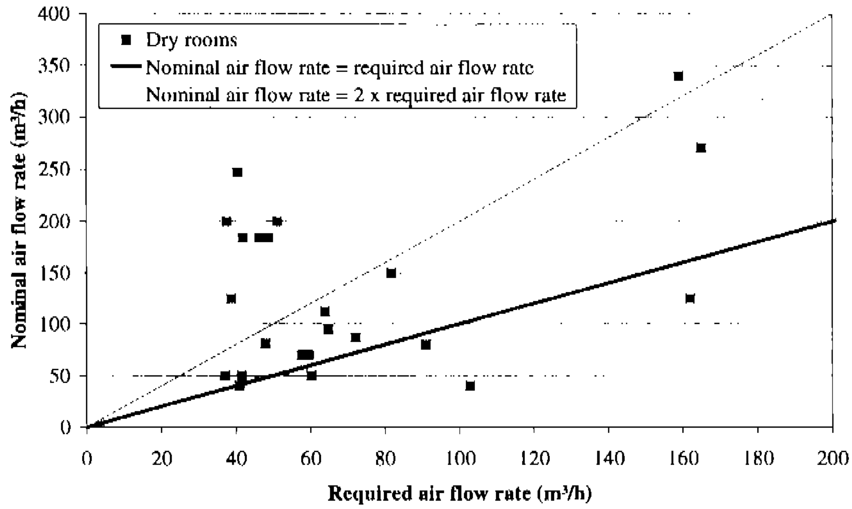

Figure 6: Comparison between installed and required nominal air flow rate.

3.2.2.2 Humid rooms

In 60 % of the rooms equipped with a mechanical extraction the real air flow rate was measured by means of a compensating flow meter. The result is shown in Figure 7.

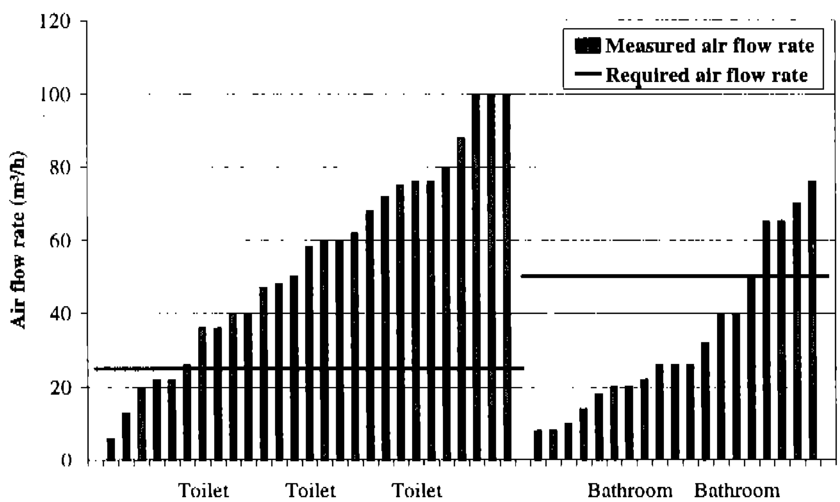

Figure 7: Measured mechanical exhaust air flow rates (the dark bars represent centralised, continually working systems).

The following observations can be made:

- The installed air flow rates often seem to differ from the prescription in the standard. In the toilets the air flow rates are usually higher than the value in the standard, while in the bathrooms the opposite is true. The average measured air flow rate is 33 m^3/h for bathrooms and 55 m^3/h for toilets.
- It is clear that in rooms with an insufficient air flow rate problems can occur with the indoor air quality. On the other hand an excessive air flow rate causes unnecessary energy losses. An important distinction has to be made between temporarily operating systems (= only during occupation and a limited period of time afterwards; often operated by the light switch) and continually working systems. In the first case it is certainly no problem that

the air flow rate is higher than the value from the standard, on the contrary, it will have a positive impact on the IAQ. In the second case the energy aspect is much more important as the additional air flow rate will be extracted continuously.

In the case of natural exhaust systems the Belgian ventilation standard not only gives prescriptions for the nominal air flow rate, but also for the section of the ductwork. The minimal section should be 70 cm² (≈ ∅ 10 cm) in toilets and 140 cm² (≈ ∅ 14 cm) in other rooms (bathroom, kitchen, washhouse,...). Figure 8 shows the existing situation in the investigated dwellings. In 75% of the toilets the section of the duct is sufficient, while only 1/3 of the bathrooms comply with the ventilation standard.

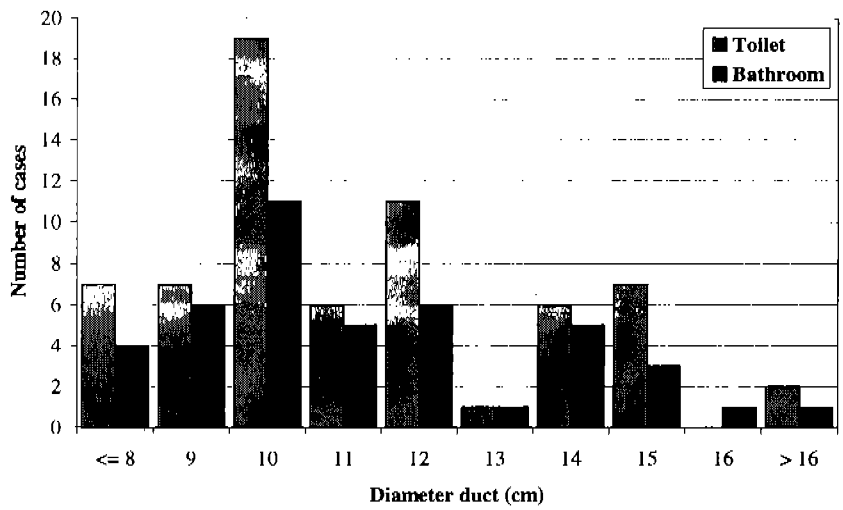

Figure 8: Diameter of the exhaust ducts for natural ventilation (behind the register).

3.3 Ventilation facilities for intensive ventilation in the investigated dwellings

In living rooms, bedrooms, studies and kitchens there must be a certain area of doors and windows that can be opened. If a kitchen doesn't have external doors or windows that can be opened a cooker hood must be installed.

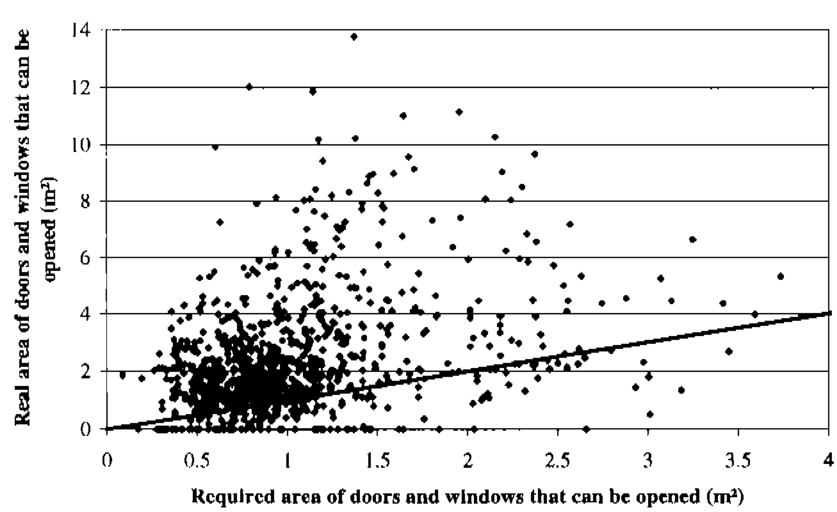

Figure 9: Area of doors and windows that can be opened.

In Figure 9 the area of doors and windows that can be opened per room is compared with the requirement. It is obvious that in most cases (more than 80%) the requirement is fulfilled. Although a cooker hood is only required in kitchens without an external door or window that can be opened, this device for intensive ventilation is found in the majority of the kitchens: more than 95% !

4 CONCLUSIONS

In spite of the existence of a standard for ventilation in dwellings, the presence of ventilation facilities is quite poor in new houses in the Flemish Region, especially in the dry rooms. Moreover, quite often the performances of the installed facilities doesn't seem to be in accordance with the prescriptions from the ventilation standard.
It was remarkable that a lot of complaints (in about 30% of the investigated dwellings) about IAQ were recorded. This indicates that there is a lack of understanding of the importance of ventilation and the way it can be achieved correctly.
The situation could probably be improved significantly by the implementation of a performance check on site.

5 ACKNOWLEDGEMENTS

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

COMPARISON OF IAQ PERFORMANCES OF FRENCH VENTILATION SYSTEMS IN RESIDENTIAL BUILDINGS

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ABSTRACT

Until now, there is no widely accepted way to express any index for this purpose and taking into account the large variety of possible pollutants. Things can be simplified is the aim is more to compare different systems and strategies than to give an absolute value of quality.

For the study of a pollutant source, the main important point for comparison is the pattern of its production, whatever this pollutant is. For human feeling and health we defined 5 main generic pollutants: constant emission related to the room area; human metabolism (using CO2 as a tracer); emission due to cooking activities; passive smoking; indoor humidity related to the dryness feeling.

The detailed data is for each inhabitant the curve of the number of hours above a pollutant level concentration Ci : Nh (Ci). A condensed one is calculated as the cumulated value above a threshold limit Cimax. This is the basis for the results presented here. Other parameters are also calculated as pressure difference between outdoor and indoor, room related parameters (humidity, condensation hazards), and energy parameters (heat needs and fan energy). This methodology was defined and used in the framework of IEA annex 27 "domestic ventilation".

The main ventilation systems used in France have been described based on the Ann27 approach, applied by using the ventilation code SIREN, developed by CSTB. The sensitivity analysis presented in the paper takes into account different climates, dwelling types, air airtightnesses, dwelling occupancies, water vapour production.

KEYWORDS

ventilation system, IAQ index, ventilation heat losses, ventilation codes,

INTRODUCTION

The IEA Ann 27 "domestic ventilation" developed a methodology making it possible to compare different ventilation systems and strategies on the same international approved basis. We first present this approach

THE IEA ANNEX 27 METHODOLOGY

In the IEA annex 27 project ventilation systems are study: nine dwellings (3 plans and 3 airthightness), three occupancy (spacious, average, crowded), three climates (cold, mild, warm). [1]

Results are given in term of indoor air quality (pollutant exposure for each inhabitant) condensation risks and energy.

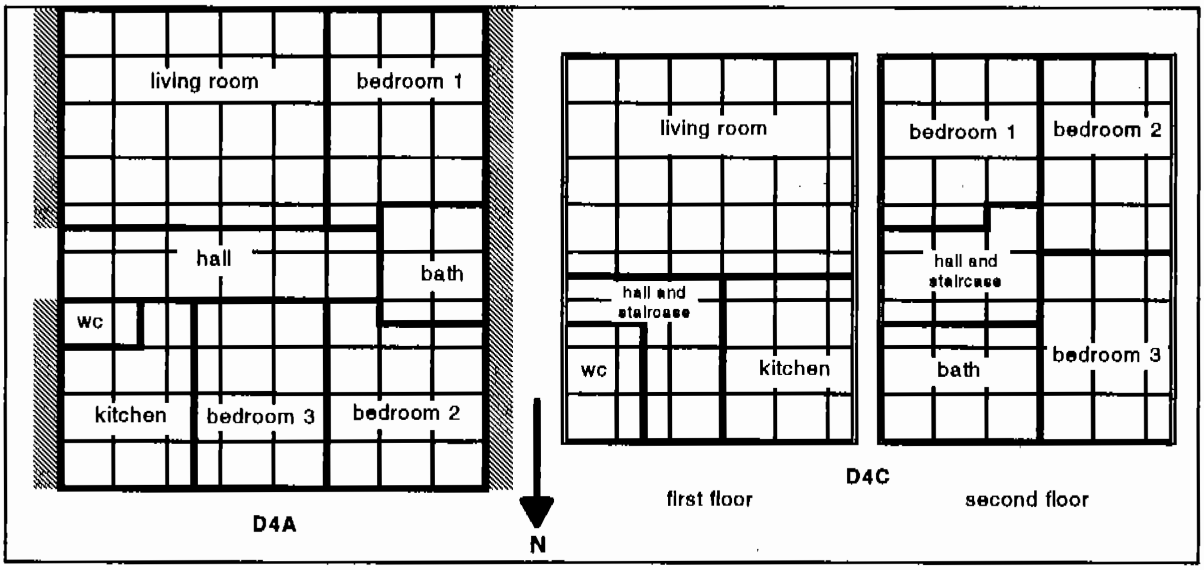
Dwellings: three dwellings have been considered:

- D4A : four rooms flat located on ground floor in a four-storey building.
- D4B : four rooms flat located on top floor in a four-storey building.
- 4C : 4 rooms single family house

Leakage values are given hereafter :

	n50 (ach)		
D4A	1	2.5	5
D4C	2.5	5	10

Half of the cracks are located at 0.625 m from the floor and the other half 1.875 m from the floor for the leakage 1 , 2.5 and 5. For leakage 10, the additional cracks are located at the floor and at the ceiling.



The climates are related to meteorological data from :

- cold : Ottawa (Canada).
- mild : London (United Kingdom).
- warm : Nice (France).

Four ventilation systems are designed :

- natural airing
- passive stack
- mechanical exhaust
- mechanical exhaust and supply

Dimensioning make use of inlets or outlets size and airflows (extract and supply)

For each system an alternative consists in additional fans in kitchen and bathroom, opening windows in bedrooms.

Indoor air quality for people

For the study of a pollutant source, it can be noticed that the main important point for comparison is the pattern of its production (level versus time and place), whatever this pollutant is. Therefore, it is possible to define some generic pollutants, which will be defined only by their pattern.

For human feeling and health we propose at first to base the comparisons on five main generic pollutants :

- Plt1 : constant emission related to the room area. It could be related to pollutant emission by the rooms themselves.
- Plt2 : human metabolism. It is based on the CO_2 production.

- Plt3 : cooking activities. It is based to the water evaporated during cooking and could be related to odours production, as to CO or NOx production in case of gas appliance.
- Plt4 : passive smoking . It is based on a production of pollutant for the hours and place when and where people are smoking (production 20 U4/h in the living when woman is present between 13 - 24 o'clock).
- Indoor humidity : this one is here only related to the dryness feeling. It is not a generic pollutant as it can be expressed directly in terms of indoor relative humidity.

A weekly schedule of the dwelling occupancy has been defined by IEA annex 27 [1].

Results :For each inhabitant we give the curve of the number of hours above a pollutant level concentration Ci : Nh (Ci). These results are also given in a condensed form based on the calculation of the cumulated value above a threshold limit. For CO2, the limits are 700 and 1400 ppm : the condensed outputs are expressed in ppm.h above these two levels.

The energy needs must be split into heat needs and electrical needs for fan.

The heat needs is calculated knowing the airflows to the outdoor and the temperature difference between outdoor and indoor. The airflows are separated into three parts :

- air exhausted by the ventilation system,
- air exfiltred through the envelope,
- airing by opening windows.

The average air flow and air change per hour is the direct averages during the heating season of the overall air dwelling airflow. It is not of direct interest because it is neither related to indoor air quality nor to heating needs. For example with the same average airflow, the heat needs will be increased if the ventilation in winter is higher (passive stack systems) and decreased if the ventilation is lower (humidity controlled systems).

We calculate **heat needs equivalent air flow rate** and air change rate which are the constant airflow (or air change rate) which would lead to the same heat needs as the ones calculated. It is simply calculated by :

$$Qave = \frac{\int 0.34 \times Q(t) \times (Ti - Te(t)) \times dt}{\int 0.34 \times (Ti - Te(t)) \times dt}$$

The electrical needs are calculated on the whole year if this corresponds to the system running: the power is considered 40 W for a dwelling : the annual consumption is 350 kWh.

THE COMPUTER CODE *SIREN95*

The computer code SIREN95 is an evolution SIREN ("SImulation du RENouvellement d'air") developed in C.S.T.B [2]. It is used to calculate the airflow throughout the entire heating season (about seven months) in a dwelling (with a maximum of 15 rooms).

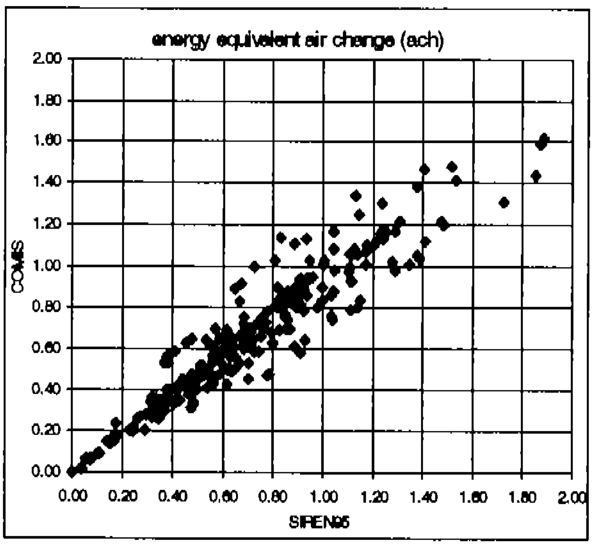
The code uses hourly meteorological data (temperature, relative humidity, wind speed and orientation) ; occupancy and pollutants production (CO_2 H_2O ...) are defined with a half an hour step ; actually four pollutant, plus water vapour can be taken into account ; for water vapour hygroscopic inertia an be adjusted. Pollutants and humidity concentrations are assumed uniform in each room.

In SIREN95 internal pressures are assumed a hydrostatic field ; the inside temperature is considered constant in a horizontal plane (only vertical gradient : stack effect is taken into account).

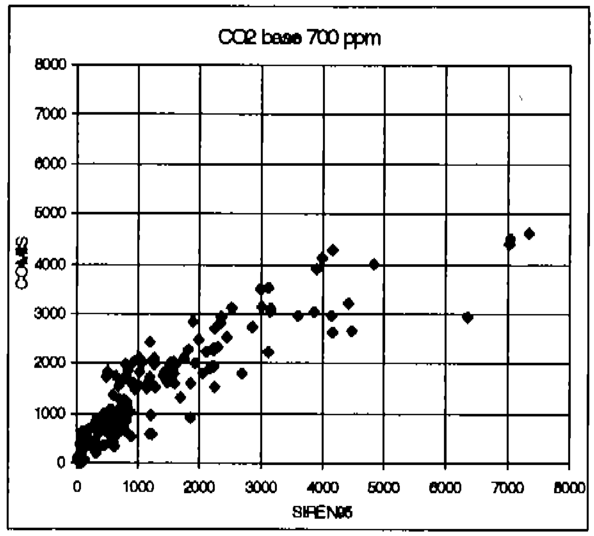
Each component (air inlets, outlets, cracks, fans, windows, ...) is characterised by its flow rate curve as a function of the pressure difference and also when relevant, of the temperature, pollutant concentration or relative humidity. The curves of components can be given by functions (not necessary smooth) or tables : a possible hysteresis can be taken into account.

SIREN95 is written in FORTRAN 77 ; it runs an entire heating season in roughly four minutes (PC Pentium 150 MHz or more)

We compared COMIS and SIREN95 on the results of 174 simulations with changes in dwellings, ventilation system, climates, occupancy, ...



Results of equivalent air change are in good accordance : for natural airing, SIREN95 (which don't take into account pressure losses due to internal doors) gives a higher level than COMIS



Four cases corresponding to natural windows airing and passive stack ventilation systems are quite different (values calculated with SIREN95 above 6000 ppm.h).
For mechanical ventilation systems (exhaust only and balanced results are in good accordance.

PERFORMANCES OF VENTILATION SYSTEMS

With SIREN95 we have run 990 simulations with changes in dwellings, ventilation system and climates.

For the most important parameters (IAQ, condensation, energy) the results are given in five classes (++ - - approach). The class limits are not directly reproducing the 20% values, as the curves are not linear (which means that doing so would not make it possible to appreciate well the - - class).

We give hereafter the classes limit for three pollutants and equivalent air change rate :

	IAQ			
	CO_2 700ppm	cooking	passive smoking	energy
unit	kppm.h	U3.h	U4.h	ach
++ to +	500	600	400	0.4
+ to 0	1000	1000	600	0.6
0 to -	2000	1500	1000	0.8
- to --	4000	4000	1600	1.0

Results are given in tables as follows :

- indoor air quality : we calculated for each case the - - to ++ classes for CO2, cooking products, and passive smoking ; the final result is the worst value.
- condensation : we calculated for each case the - - to ++ classes for the habitable rooms and the wet rooms ; the final result is the worst value.
- energy (equivalent air change rate).

APPLICATION TO VENTILATION SYSTEM IN USE IN FRANCE

Cases studied

We used the typical D4a and D4c dwellings for three typical French climate :. Nancy, Trappes and Nice. We took into account the 3 typical families with an additional sensitivity study based on different schedules of water vapour production

The different ventilation system are as follows :

Mechanical exhaust only

It is based on available products according to French regulation

	Dwelling D4A	Dwelling D4C
Kitchen outlet	45 / 120 m^3/h	45 / 120 m^3/h
WC outlet	30 m^3/h	30 m^3/h
Bathroom outlet	30 m^3/h	30 m^3/h
Living-room inlet	module 30	module 22
Bedroom inlet	module 45	module 45
Fan power	45 W	45 W

The air inlet module is the airflow under 20 Pa

Humidity controlled mechanical exhaust only system

These systems were designed and dimensioned according to the manufacturer specifications. Two kinds of system are taken into account :

- Humidity controlled type 1 : humidity controlled kitchen air outlet only
- Humidity controlled type 2 : humidity controlled air inlets and kitchen outlet

The fan power is 60 W.

Balanced ventilation system

The exhaust part is the same as the mechanical only one. The supply part is taken equal to 21 m^3/h in the bedrooms and 42 m^3/h in the living room.

The fan power is 60 W for the supply part and 55 W for the exhaust part.

Passive stack

This system is taken into account for the detached house only, based on available guidance document "Solutions techniques pour le respect du règlement thermique en maison individuelle"

- kitchen :	exhaust device ("grilles"), 400 cm² circular duct with the sizes: 200 mm diameter ; length 5.6 m cowl 200 mm diameter ζ=1,5 C=-0,5
- bathroom :	exhaust device ("grilles"), 100 cm² circular duct with the sizes: 160 mm diameter ; length 2.8 m cowl 160 mm diameter ζ=1,5 C=-0,5
- WC :	exhaust device ("grilles"), 100 cm² circular duct with the sizes: 125 mm diameter ; length 5.6 m cowl 160 mm diameter ζ=1,5 C=-0,5

Mechanical supply in the hall

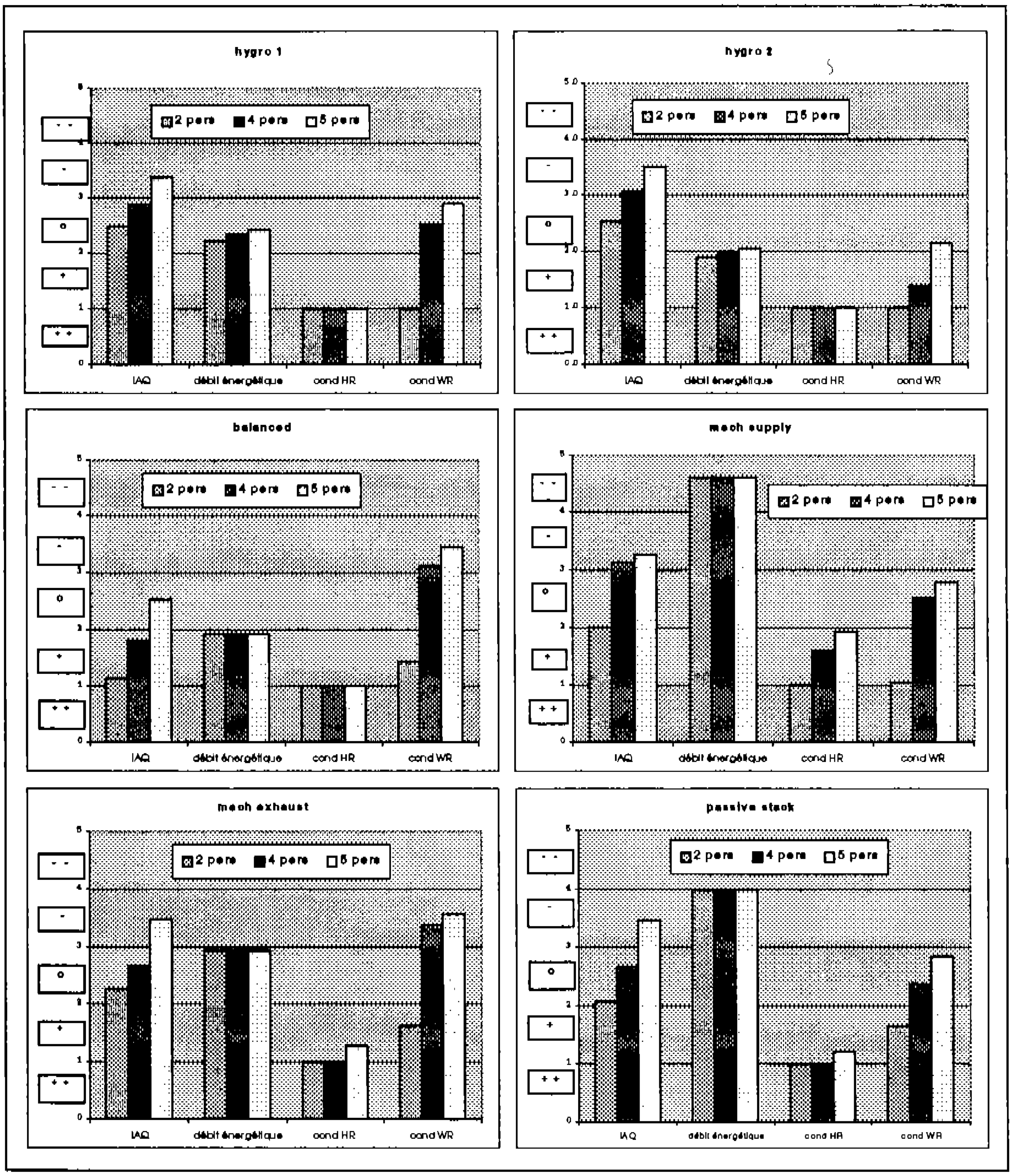
This system is available on the French market and is mainly used for retrofitting. It has been taken into account for the detached house only starting from a situation corresponding to existing buildings (grilles in the wet rooms according to sanitary regulation).

- Supply air in the upper part of the hall with an air flow of 169 m^3/h ,
- grilles in all main rooms (area 30 cm² corresponding to a module 45)
- grilles in the upper part of all wet rooms (100 cm²),
- one grille in the lower part of kitchen (100 cm²).

The fan power is 75 W.

Results

In order to facility the comparison, it is possible to present a quality profile for each system and occupation. For each case of quality index, the note is given as a 1 to 5 index corresponding to the + + to - - quoting.



The main results can be described as follows :

	IAQ		
occupants/dwelling->	2	4	5
Hum controlled 1	+	o	o
hum controlled 2	o	o	-
balanced	++	+	o
mech supp	+	o	o
mech ex	+	o	o
passive stack	+	o	o

All systems give an acceptable air quality, except for HC2 for over occupied dwellings.

	energy equiv. air change rate		
occupants/dwelling->	2	4	5
Hum controlled 1	+	+	+
hum controlled 2	+	+	+
balanced	+	+	+
mech supp	--	--	--
mech ex	o	o	o
passive stack	-	-	-

H.C. and balanced systems are the most efficient ones. PSV is less efficient due to the sensitivity of airflows to outer conditions. Central mechanical system is the less efficient one because each room is ventilated separately.

	condensation in wet rooms		
occupants/dwelling->	2	4	5
Hum controlled 1	++	++	++
hum controlled 2	++	++	++
balanced	++	++	++
mech supp	++	+	+
mech ex	++	++	++
passive stack	++	++	++

All systems are quite efficient. The Hall mechanical supply is + as the exhaust air is poorer controlled in wet rooms.

	condensation in habitable rooms		
occupants/dwelling->	2	4	5
Hum controlled 1	++	o	o
hum controlled 2	++	++	+
balanced	++	o	o
mech supp	++	o	o
mech ex	+	o	-
passive stack	+	+	o

The efficiency depends highly on the number of occupants.

CONCLUSION

The choice of a ventilation system is always based at least on a balance between indoor air quality for people, condensation risks and energy needs for heating and fan. The methodology developed within Ann 27 proved to be useable for other system than the one taken into account within this group, ending with a multiparameter comparison. This makes it possible to give advices for the choice of existing systems, and the development of new ones by quantifying the impact of the system design and dimensioning.

ACKNOWLEDGEMENTS

The CSTB thank the ADEME, EDF and GDF for her financial support.

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[1] IEA Annex 27 "Assumptions for the simulations"

[2] Villenave JG, Millet JR, Ribéron J "Theoretical basis for assessment of air quality and heat losses for domestic ventilation systems in France". 14th AIVC conference on Energy Impact of Ventilation and Air Infiltration, Copenhagen, September 1993.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19^{TH} ANNUAL AIVC CONFERENCE
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EVALUATION OF THERMAL PERFORMANCES OF RESIDENTIAL VENTILATION SYSTEMS WITH HEAT RECOVERY

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Evaluation of thermal performances of residential ventilation systems with heat recovery

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Synopsis

Ventilation systems with heat recovery offer several advantages such as, of course, energy savings but also the possibility to add acoustic and filtration treatment. This study was to evaluate the thermal performances of such systems for residential ventilation in France.
These units usually combine exhaust and supply fans, filters and a heat recovery exchanger. To test them, a special draft is being written by the CEN experts of TC 156/WG2/AH7. The study included the test of several units according to this project to evaluate their performances : temperature ratios, airflow/pressure curves, electric power, internal and casing leakages... The influence of humidity, condensation, frost and several full duct-systems simulating a standard house equipment have also been tested.

List of symbols

θextr :	exhaust air dry bulb temperature
θwet, extr :	exhaust air wet bulb temperature
θdew, extr :	exhaust air temperature at dew point
θsupply :	supply air dry bulb temperature
$\Delta\theta$supply :	difference of temperature on supply air
$\Delta\theta$inlet :	difference of temperature between both supply and exhaust air entering the unit
η :	temperature ratio (defined in EN308) in dry conditions
ηsyst :	temperature ratio defined at the limits of the system (unit + ducts)
ηx :	humid temperature ratio (defined in EN308)
Δxsupply :	difference of water-contents on supply air (inlet-outlet)
Δxinlets :	difference of water-contents on unit inlets (supply-exhaust)

1. INTRODUCTION

This study aims the evaluation of the thermal performances of balanced ventilation systems with heat recovery, used in standard single houses in France. These systems also have other advantages like acoustic and filtration in relation with outside conditions. These advantages should be studied in the future for a global performance aspect.

The study focuses on :

- thermic and aerodynamic performances of the unit
- influence of humidity and frost on efficiency
- test and simulation of a full single, family-house system with ducts, in three standard climates in France.

2. PERFORMANCES ON THE UNITS

2.1 Description of the units

Five units combining exhaust and supply fans, filters and a heat recovery exchanger were tested. Four units had double speed fans to fulfill the conditions of French regulation which imposes a standard extract airflow and an increased one in kitchen when necessary.

Unit n°	1	2	3	4
Fan position				
Exhaust	downstream	upstream	upstream	downstream
Supply	downstream	downstream	upstream	downstream
Number of Heat exchanger tested				
number	2	1	1	2
material	plastic (PVC)	plastic (PVC)	plastic (PVC)	plastic (PVC) and aluminium

Figure 1: description of the units.

2.2 Temperature ratio

For each fan speed, temperature ratio were tested according to EN308 [1] and CEN TC156/WG2/AH7 [2] project : $\eta = \frac{\Delta\theta_{supply}}{\Delta\theta_{inlets}}$

During tests, exhaust and supply airflows were balanced and then slightly unbalanced. The test conditions, without condensation, were as followed :

- exhaust air dry bulb temperature : 25°C wet bulb temperature : < 14°C
- supply air dry bulb temperature : 5°C

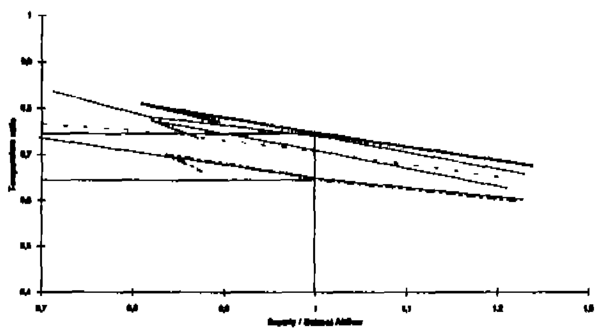

Figure 2: unit temperature ratio-standard speed-unbalanced airflows

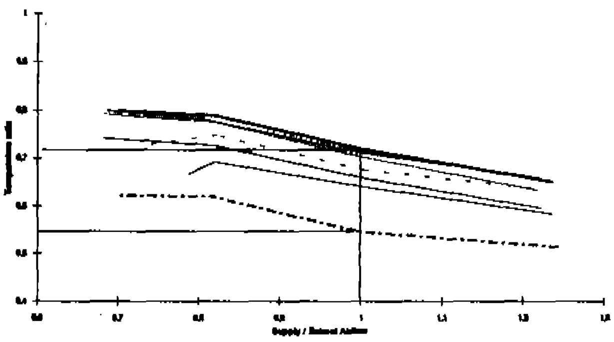

Figure 3: unit temperature ratio-maximum speed-unbalanced airflows

Results in figures 2 & 3 show that the temperature ratio of the units have varied from 64 to 74% in standard speed for balanced airflows. Unbalanced airflows modify these values.

2.3 Influence of humidity

On the same unit (# 1), tests were made for 4 test points with condensation of water vapor contained in exhaust air:

Test point	1	2	3	4
θsupply	7°C	2°C	7°C	2°C
θextr θwet, extr	20°C 12°C	20°C 12°C	20°C 17°C	20°C 17°C
Δθ = θdew, extr - θsupply	4.7 °C	9.6 °C	10.1 °C	15.1 °C
Relative Humidity	68.8%	81.3%	80.5%	84.3%

EN308 defines the humid temperature ratio as : $\eta_x = \dfrac{\Delta x_{supply}}{\Delta x_{inlets}}$

As supply air does not condensate, this ratio is equal to zero.

As condensation depends on :

- exhaust air moisture (θdew, extr)
- dry bulb temperature on supply air (θsupply).

Figure 4 shows the influence on the efficiency ratio depending on : $\Delta\theta = \theta_{dew, extr} - \theta_{supply}$
The increase of temperature ratio due to condensation is mainly linear (+ 1,5 %/°C) and was interpolated to estimate the seasonal performance (see § 3.3.4). This interpolation can be considered as valid as long as exhaust temperature is the same, which is more or less the case in the house we have considered.

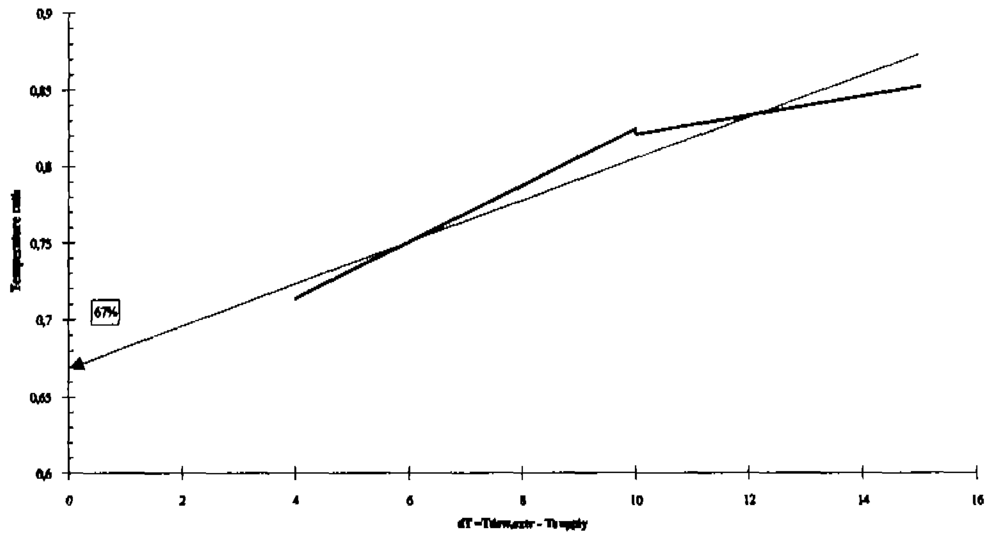

Figure 4: influence of condensation

2.4 Influence of frost

A test has been run during 4:30 hours with the following conditions :

θextr : 20°C (θdew, extr has varied between 16 and 17.4°C during the test)

θsupply : - 7°C

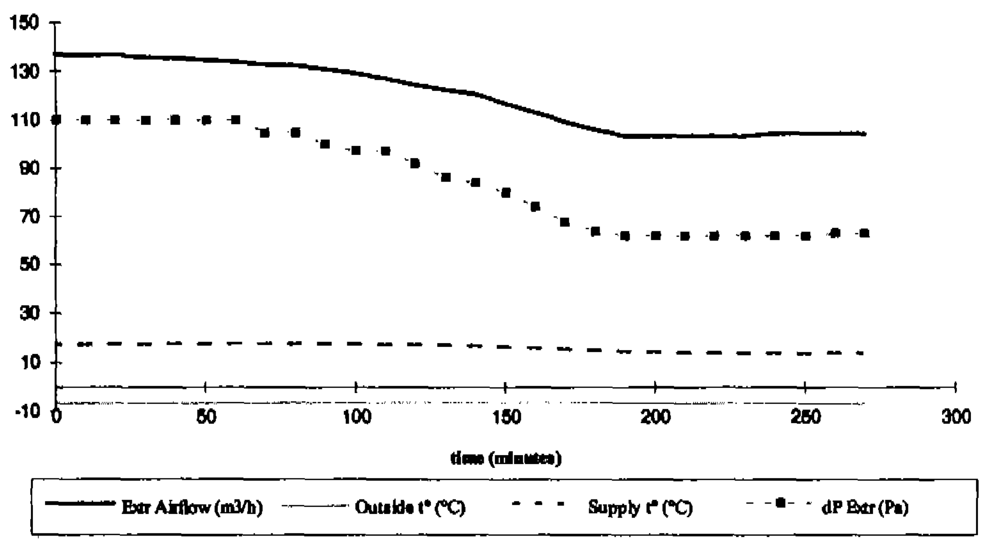

Figure 5: loss of performances due to frost

The loss of performance was stabilised after 3 hours and the airflow had decreased of 27%. The exchanger was not totally filled in by frost. We can consider that a lower supply temperature could induce a more important loss of airflow even down to zero. A variation of

the exhaust air moisture induces a variation of the time necessary to stabilise. For French climates, the risk of frost is mainly restricted to some continental areas like eastern part of France where outside temperatures are more extremes.
In this area, the use of a supplemental coil is necessary to prevent frost.

3. PERFORMANCES OF A SINGLE-HOUSE SYSTEM

3.1 Description of system

We have dimensioned two different distribution ducts for a standard house which are shown on figures 6 and 7.

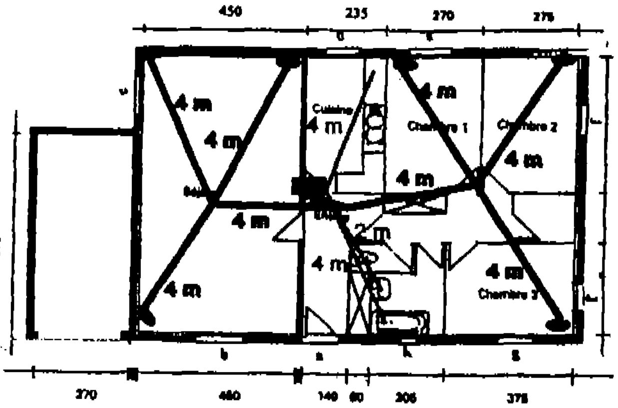

Figure 6: air distribution ducts serial configuration

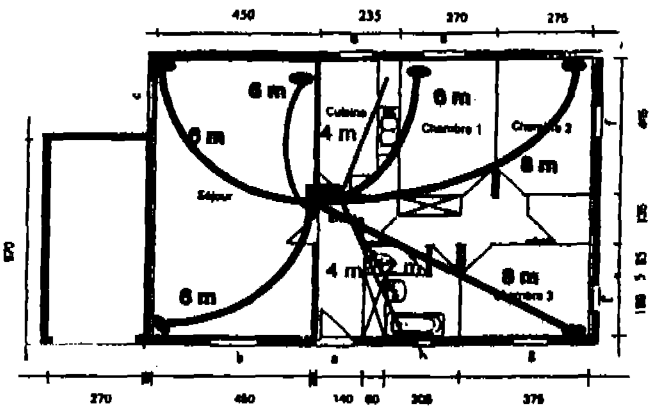

Figure 7: air distribution ducts parallel configuration

3.2 Test results

Tests have been made for two positions :

- unit and ducts situated in the attic (which was considered between fully-insulated and not-insulated). Ducts were insulated (25 mm or 50 mm) ;
- unit and ducts inside the heated volume. Ducts were not insulated.

The main results are indicated in appendix 1 and 2.

The "system temperature ratio" is defined as : $\eta_{syst} = \left(\frac{\Delta\theta_{supply\,air}}{\Delta\theta_{inlets}} \right)_{system}$

where temperatures were considered at the entrances or exits of the full system (duct + unit). When the system was inside the heated volume, we have excluded in temperature ratio calculations all exchanges with the volume itself through the unit casing or the ducts for example. When the system was in the attic, these exchanges were considered but limited to a maximum equal to heated volume results. As attics are indirectly heated by the house, when the construction techniques lead to a strong insulation directly in the attic and not between the attic and the heated volume, the system tends to act as a heated-volume system.

These results show that the system temperature ratio was between :
- 51 to 58% positionned in the attic
- 64 to 74% positionned in the heated volume.

3.3 Transient simulations

Simulations in dynamic state were run with TRNSYS/IISIBAT program. The house was totally described and 3 systems were modelised : mechanical exhaust (without heat recovery), supply and exhaust system in the attic, supply and exhaust system in the heated volume. Airflows were dimensionned according to French regulations and the system was slightly unbalanced (supply airflow ≈ 0.9 exhaust airflow).

3.3.1 Internal conditions

Room temperature was considered at 19 °C with 2 °C stratification that induced an exhaust temperature of 21 °C at minimum.
Scenarios of occupation were supposed corresponding to a family of 4 persons, 2 adults not present during the day and 2 children.
These elements induced humidity scenarios during the day as well as some heating gains.

3.3.2 Ventilation systems

The temperature ratio of the heat exchanger in "dry condition" was considered at 67 %. Influence of humidity was taken into account (according to § 2.3). A supplemental coil of 500 W to avoid frost was supposed to be installed and to start functionning when the outside temperature is below -5 °C.

3.3.3 Outside climate

Simulations have been run on three different site conditions in France :
- Nancy : continental climate with rough winter
- Trappes : near Paris, between Oceanic and continental
- Carpentras : South of France, almost mediterranean.

3.3.4 Results

The ventilation system with heat recorery compared to an exhaust ventilation system, without recovery, most common in France, leads to the following energy gains :

- system situated in the attic :	43 %
- system in the heated volume :	66 %.

Yet, when considering the infiltration airflow due to the fact that the house is at an internal pressure close to the external one, with the balanced ventilation system, and the fan absorbed power, these values decrease respectively to : 23 % and 44 %.
To obtain these values, infiltrations were calculated by a simplified method and these results should be checked with a more accurate modelisation, taking into account real wind.
Condensation increased in average the system temperature ratio of 1 to 3 %.
In France, the supplemental coil to avoid frost has an absorbed power which could have been neglected (32 kWh for one year, in Nancy).

4. CONCLUSION

Supply and exhaust systems with heat recovery allow important energy savings in residential ventilation systems :

- 43 % when the system is situated in the attic,
- 66 % when the system is in the heated volume.

Yet, these values are lower if you take into account infiltrations and fan asborbed power. A more precise study on infiltrations should be necessary.
The five heat recovery units tested have shown that temperature ratios :

- are between 61 and 74 % for dry air,
- increase proportionnally with (θdew, extr - θsupply).

The loss of performances due to frost can be important for a long period at low outdoor temperature and high indoor humidity, which is limited to the eastern part of France. A supplemental coil to avoid frost is necessary in these cases.

References

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[2] CEN TC156/WG2/AH7 N3 - January 1998 "Components/Products for residential ventilation. Performance testing, Part 7 : Performance testing of a mechanical supply and exhaust ventilation unit".

APPENDIX 1: main test results on system situated in the attic

Duct insulation	mm	50	25	25	25	25	25
Configuration		serial	serial	serial	serial	serial	serial
SUPPLY AIR							
Outside temperature	°C	-0,1	0,0	0,0	-10,2	-5,1	10,0
Supply temperature	°C	12,3	11,2	11,2	5,9	8,8	16,9
Airflow	m3/h	119	119	175	128	120	119
Recovered power	W	497	453	666	706	571	275
Accuracy	%	+- 10 %	+- 10 %	+- 10 %	+- 10 %	+- 10 %	+- 10 %
EXHAUST AIR							
Room temperature	°C	21,1	21,1	21,2	20,8	21,1	20,9
Room temperature at dew point	°C	9,8	10,0	10,4	10,6	10,8	9,6
Inside Relative Humidity	%	48,5	49,2	50,4	51,9	51,5	48,6
Exit temperature	°C	9,4	9,7	10,7	4,4	6,6	15,1
Exit temperature at dew point	°C	7,5	7,7	8,8	2,9	4,8	9,5
Outside Relative Humidity	%	87,7	87,1	87,8	89,8	88,5	69,0
Airflow	m3/h	134	135	195	140	132	140
Power	W	659	653	832	1204	976	281
AMBIANCE	(attic temperature)						
Average temperature	°C	6,1	6,0	6,0	-7,1	0,0	15,0
Atm pressure	kPa	98,4	98,8	98,6	99	99,016	98,7
CALCULATION							
ηsyst	%	58,60%	53.2%	53.1%	51.7%	53.2%	53.2%

APPENDIX 2: main test results on system situated in the heated volume

SUPPLY AIR					
Outside temperature	°C	-4,4	-4,2	2,9	3,0
T° sèche sortie	°C	17,0	17,2	18,6	18,5
Airflow	m3/h	129	218	123	123
EXHAUST AIR					
Room temperature	°C	21,1	21,0	21,1	21,0
Room temperature at dew point	°C	10,8	10,6	2,9	10,4
HR entrée	%	51,7	51,5	30,2	50,9
Exit temperature	°C	10,2	10,7	13,8	13,8
Exit temperature at dew point	°C	2,3	9,3	2,9	7,5
HR sortie	%	57,8	91,6	47,5	65,7
Airflow	m3/h	134	155	136	136
AMBIANCE	(Room temperature)				
T ambiante Moy	°C	20,6	20,5	22,7	20,4
P Atm	kPa	99,3	98,02	98,9	99
UNIT					
θentrance, exhaust air	°C	20,8	20,8	21,3	20,7
θexit, exhaust air	°C	6,1	7,4	10,5	10,5
θambiance, supply air	°C	-1,0	-1,6	6,6	6,6
θexit, supply air	°C	13,2	13,9	15,8	15,4
Temperature ratio (corrected for heated volume)					
Equivalent outside temperature	*°C*	*-2*	*-2,1*	*5,6*	*5,6*
Recovered power	*W*	*818*	*1383*	*528*	*523*
ηsyst	%	74.4%	70.3%	64.4%	64.3%

APPENDIX 3: main results of dynamic simulations.

One year simulation - Quantity of Energy (kWh)

Mechanical exhaust

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	3981	208	3621	168	5532	198	3406	168
Trappes	6770	258	4396	188	9209	253	4248	188
Nancy	7200	270	4547	188	9806	265	4401	188

Exhaust and Supply system in the attic

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	2851	711	2058	336	4356	659	1459	336
Trappes	5290	865	2485	376	7643	832	1822	376
Nancy	5603	900	2552	376	8182	867	1919	376

Exhaust and Supply system in the heated volume

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	2272	718	1290	336	4336	658	1449	336
Trappes	4497	871	1484	376	7612	832	1790	376
Nancy	4828	907	1549	376	8140	868	1859	376

VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

EVALUATING THE COMPULSORY PERFORMANCE CHECKING OF VENTILATION SYSTEMS IN SWEDEN

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Synopsis

The Swedish Parliament decided 1991 that ventilation systems in all non-industrial buildings should be regularly inspected in intervals from 2 to 9 years, shortest for schools, hospitals etc and longest for natural ventilated flats. The systems are checked to fulfil the requirements given when installed.

The goals of the evaluation were to give estimated rates for how many systems that were approved at the end of 1997 and the cause of the faults that made the system either not to be approved or to be remedied before next inspection.

The evaluation was made in three steps:

1. A questionnaire was sent out to a selected number of municipality authorities
2. A questionnaire was sent out to housing organisation representing more than 60 % of all apartments in Sweden.
3. Totally 10 300 complete inspection protocols

Results showed the estimated approved rate at the end of 1997to be: schools 85 – 90 %, day nurseries 90 – 95 %, hospitals ≈ 40 %, Offices ≈ 40 %, Dwellings 65 – 70 % (condos 85-90 %, public owned 75-80 %, private <50 %). Stack ventilation has far more faults than any other system. Repeated inspections decrease the number of faults and increase the approved rate. The most frequent fault was insufficient flow rate.

1. Background

The Swedish Parliament decided 1991 that ventilation systems in all non-industrial buildings should be regularly inspected. The exception was single family houses with mechanical exhaust and natural ventilation. The checking intervals are depending on the occupants and on the system principles. There are five classes with inspection intervals from 2 to 9 years:

1. Schools, hospitals, day-nurseries, 2 years
2. Multi family buildings, offices with mechanical supply and exhaust ventilation (**MSE**), 3 years (with or without heat exchanger)
3. Multi family buildings, offices with mechanical exhaust ventilation (**MEO**), 6 years
4. Multi family buildings, offices with passive stack ventilation (**PSV**)
5. Houses with mechanical supply and exhaust ventilation

The main reason for the mandatory inspection was the increased number of oversensitive reaction amongst people, in particular children. Indication showed also that ventilation systems were not working according to the designed intention.

2. Method

The goals with the investigation were to estimate the frequency of approved systems at the end of the year 1997 and the cause of the faults that made the system to fail. At an inspection faults can occur in many different places in a system. A fault can be of two dignities.

1. One is a fault that must be remedied before next inspection. Called Fault/Approved (**FA**).
2. The worse case is a fault, that must be corrected and followed by a new inspection before the system is approved, and a certificate is issued. Called Fault/Failed (**FF**).

To make an estimation of the percentage of ventilation in the non-industrial buildings that passed the regular inspection the approach was made from three "directions".

1. A questionnaire was sent out to a selected number of municipality authorities. Interviews.
2. The number of passed ventilation systems was asked for in three housing organisations. Two are representing most of the condominiums in Sweden and 21 % of all dwellings in multi family buildings. The third organisation is representing the municipality owned companies and 40 % of all apartments.
3. Complete inspection protocols. Totally 10 300 have been used to identify the frequency of faults for different ventilation systems, occupation use, and comparing 1995 and 1997.

3. Statistical Data

To get a brief idea of the representation of the data gathered and the situation of both dwellings in multi family buildings and for non-domestic buildings some statistical data is presented, see figure 1. It must be noted that dwellings stands for 75 % of all non-industrial built areas. The distribution of the 4.1 million dwellings between apartments and single family houses (houses) show us that 50 % of the apartments are found in the three larger city areas of Stockholm, Göteborg, and Malmö. Here are also located 40 % of the non-domestic areas, see figure 2. In Sweden are 33 % living in the three greater city areas, but also 27 % are living in small municipalities with less than 27 000 inhabitants, see table 1. The population in Sweden was 8.85 millions (1996).

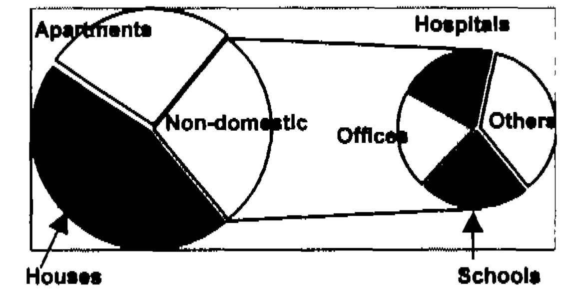


Figure 1. Area proportion of domestic and non-domestic buildings

Table 1 Inhabitants in different sizes of municipalities

Size	%
Cities (Stockholm, Göteborg, Malmö)	16
Suburbs to the above cities	17
> 90 000	15
27 000 – 90 000	25
< 27 000	27

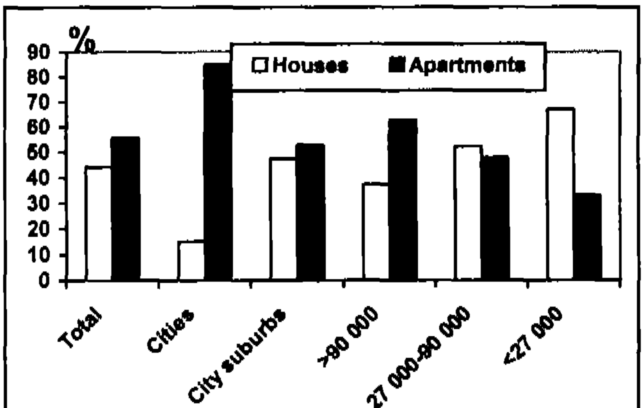


Figure 2 Houses and apartments, %, size of municipality

As we are to discuss the ventilation the main concern is what type of system the Swedish non-industrial buildings are equipped with. Most of the houses have PSV, even though nearly all have a kitchen fan. For the apartments only the old buildings have PSV and over 60 % have a mechanical ventilation. For the non-domestic buildings nearly all have mechanical ventilation or about 80 %, see also figure 3

For the three approaches the information has been gathered in the following ways:

1. Questionnaire to the municipalities. Well distributed according to the inhabitants and sizes except the three largest cities. By experience now answer was expected even though the questionnaire was sent out by the Board for housing, building and planning.
2. The collection of answers from the housing organisations have a good representation with the exception of one of the condominium organisation for which two of the city areas have large uncertainties. For the large organisation of rented apartments a very good representation is given for all sizes of municipalities and companies.
3. Most of the protocols are collected from inspections in Stockholm and Malmö city areas.

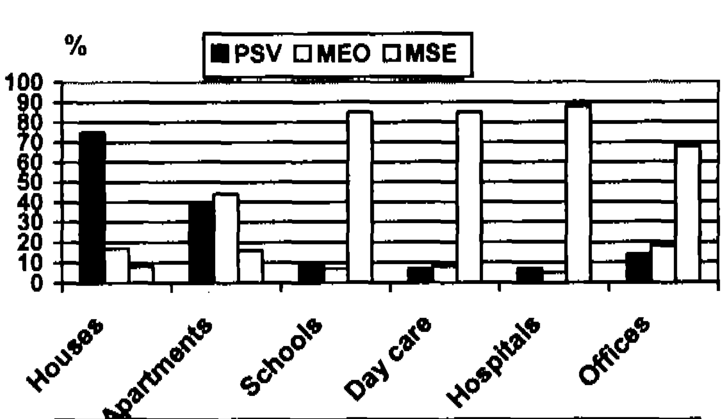


Figure 3 Ventilation systems in different buildings

In a large survey, ELIB-study, 60 municipalities were selected to represent the whole stock, ref Norlén, Andersson (1993).

4. Results

The way towards the estimation of how may ventilation systems that have been approved at the inspections is to give estimations from three different points of views.

4.1 Municipality approach

A questionnaire was sent out to 68 municipalities and answers back from 46 (68 %). It was asked for the percentage of buildings reported to the authorities to have passed the checking. As the municipalities owns most of the schools and day nurseries the reports from those are most accurate and on the other hand the information about houses are more spares or totally missing. It should be observed that all the schools and day nurseries should have been inspected at least 2 times (3 times according to the government bill but could have been postponed once) by the end of 1997. As the questionnaire was sent out in May 1997 still another 7 months of checking and renovation could be carried out. In particular the summer holiday from June to mid August is used for different measures.

Table 2 Comparison between the quest 1997 and ELIB. The estimated percentage of apartments approved at the inspection

Size	Number of quest		Representing inhabitants		Apartments, approved %	
	ELIB	Quest 1997	ELIB	Quest 1997	1995	1997
Cities, Stockholm, Göteborg, Malmö	3	0	35	0	-	
Suburbs to the above cities	13	6	14	19	80	75
> 90 000	8	4	24	35	70	42
27 000 – 90 000	19	7	20	23	75	50
< 27 000	17	29	7	23	73	65

In the distribution of questionnaires no cities were included. Many of smallest municipalities were included. Here the number of apartments are lowest. If compared to the number of inhabitants the questionnaire is giving quite a good representation of both apartments and schools and day nurseries, see table 2. However, it must be questioned the quality of the data for the apartments like: No data from the cities are given, in some municipalities only passed checking were registered, some didn't know the number of apartments.

Table 3 Percentage of buildings (bldgs) approved at the inspection. Comparison between 1995 and 1997 of the number (no) of municipalities (mnp)

Comparison	Schools		Day nurseries	
	1995, Dec	1997, May	1995, Dec	1997, May
Total, approved, %	86 %	61 %	92 %	75 %
No mnp better1997 than 1995		8		4
Equal 1995 and 1997	20		20	
All bldgs approved, no. mnp		11	14	
No. mnp 100 % bldg approved 1995 & 0 % 1997		5		4
No. mnp 0 % bldg approved 1995 & 100 % 1997		1		0

In total Sweden has 5200 schools and 15 000 day nurseries. The number that have passed the checking May 1997 is not that high as for December 1995, see table 3. However, the work continued during the whole year and as most of the buildings had been approved at inspections before half of the year had passed it was estimated that the situation was the same 1997 as 1995. But still there are buildings and systems that have not been checked at all since the bill passed the Parliament.

Interviews with responsible employees at 12 municipalities showed that most of them had the opinion that the politician was passively or actively supporting them. In particular the task is very tricky when the inspection has to take measures against the building management division within the municipality. In one case it has also been issued a penalty of fine unless a school had not been approved before a certain date. The policy is to have all the municipality owned buildings checked and passed before any measure is taken against other owners.

4.2 Questionnaires by the housing companies

In Sweden it is assumed to be about 2.3 million apartments. In general there are three categories of principal ownership, see table 4. They are condominiums, public companies (usually all shares hold by the municipality), and private companies. Most of the private companies have a much older building stock than the other. Only the organisation for condominiums and public owned companies participated in distributing the questionnaire to all the about 400 companies. For the association for the private owned real estates with about 20 000 members it was not possible to distribute a questionnaire.

As always the quality of the answers vary within a large range. Some contain very detailed information and some have given rough estimations. But astonishing most of the answers received were very detailed. Only the answers concerning condominiums had distinguished between systems, see table 6. However, not too detailed, the questionnaires give information on the matter that most of the MSE, MSEX systems have been inspected and nearly all have been approved.

As the data given for the condominiums was given in January 1997 and also information was given of apartments inspected but needed remedial action before approved if is assumed that most of those actions had been undertaken during 1997.

A slightly less number of apartments were approved in the three largest cities, see table 7. This trend was also shown for the condominiums.

Table 4 Structure of ownership for the apartment buildings

Ownership	Organisation	Apartments No	%
Condominiums	HSB	325 000	14
	Riksb.	160 000	7
	SBC	100 000	4
	Others	50 000	2
Public	SABO	900 000	40
	Others	40 000	2
Private		700 000	31

Table 5 Answered questionnaire

Organisation	Total apartments	% of apartments in Sweden	Quest answered %	Systems checked % MSE	Systems checked % MEO, PSV
HSB	300 000*	14	80	21	79
Riksb.	190 000*	8	55	26	74
SABO	900 000	40	70	-	-
Total	1 390 000	60	70	-	-

* only technical management

Table 6 Approved inspections, no. of apartments

Organisation	Approved dwell.s No.	Approved dwell.s %	Approved inspections, ventilation systems MSEX*, MSE No.	MSEX*, MSE %	MEO, PSV No.	MEO, PSV %
HSB	200 000	63	51 300	76	151 100	59
Riksb	80 000	76	23 300	88	53 600	69
SABO	470 000	75	-	-	-	-
Total	750 000	72				

*Mechanical supply and exhaust incl heat recovery

Table 7 Approved apartments (aptms), public companies (SABO). Size of municipalities

Size of municipality	Answers received aptmts	Approved aptmts %	Aptmts total
Cities, Stockholm, Göteborg, Malmö	170 000	61	225 000
Suburbs to the above cities	104 000	78	145 000
> 90 000	116 000	85	150 000
27 000 – 90 000	125 000	83	220 000
< 27 000	109 000	74	160 000
Total	625 000	75	900 000

4.3 Protocol evaluation

From the real estate owners 10 289 protocols from the inspections have been collected from 1997. The owners are in all categories mentioned above including municipalities. The main reason for the collection of the protocols were to form a basis for a computerised management of at first the ventilation checking and in a longer perspective the total management and in particular the maintenance. This means that the protocols are the ones before any remedial action has been taken. The percentage of approved systems is less than reported from the questionnaires but it must once again be observed that in the questionnaire was asked for the approved buildings after remedies and the protocols are the step before. It must be noted that in table 8 is giving results from protocols per systems (not buildings).

Table 8 Protocols split on usage and ventilation systems

Type	Protocols	Approved		MSEX		MSE		MEO		PSV	
		No.	%	No. syst.	Approved %	No. syst.	Approved %	No. syst.	Approved %	No. syst.	Approved %
Total 1997	10289	4518	44	2866		3670		3044		788	
Total 1995	8089	3013	37	2132		3205		2237		443	
Hospitals 1997	670	217	32	216		292		105		4	
Hospitals 1995	403	101	25	101		182		92		2	
Medical care 1997	194	86	44	76		91		26		7	
Medical care 1995	73	46	63	29		25		20		4	
Day nurseries 1997	645	436	68	324	66	198	64	135	80	10	40
Day nurseries 1995	332	160	48	142	51	111	40	65	70	8	38
Schools 1997	2444	1417	58	789	63	1089	53	542	64	50	44
Schools 1995	2299	791	34	562	44	1081	27	544	41	48	23
Offices	2321	893	38	788	50	1250	34	200	29	83	19
Shops	277	204	74	163	95	92	47	26	19	10	10
Apartments	2589	709	27	209	36	93	34	1737	28	550	21
Houses	78	26	33	78							
Others	857										

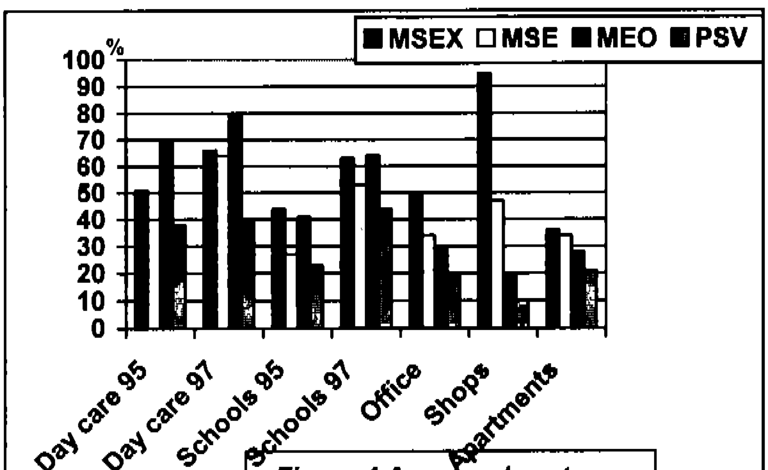


Figure 4 Approved systems

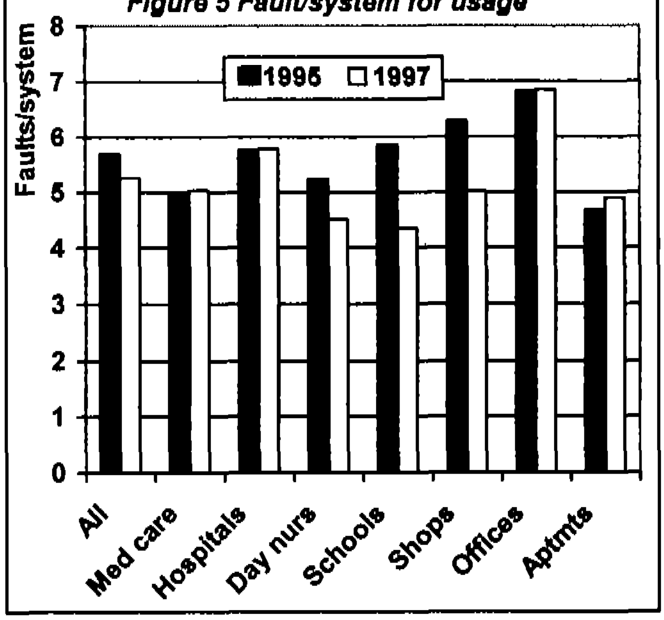
The percentage of approved systems have increased from 1995 to 1997 for the categories that have had the shortest intervals between the inspections: day nurseries, schools, hospitals. See table 8 and figure 4. The general trend is also that MSEX is the most approved system and PSV least. MSE and MEO is similar. The trend remains similar between the compared years. One explanation is that while MSE systems are designed as component delivery and installed according to drawings and also older systems, the MSEX are younger and delivered as systems with responsible producer and understanding of system approach. The PSV is old systems in old buildings. Sometimes it had been claimed that PSV suffered from lack of maintenance and could be better after remedy. But this is not shown as PSV both has least approved and most faults, see table 9. The situation for day nurseries, schools and hospitals are very relevant for the situation as most of the systems in Stockholm and Malmö is included. For offices some very big owners are included. For apartments the relevance is a little less while for shops it is a very special type of shops thus not giving the general trend. Houses are in to low number to be evaluated.

The protocol have 37 inspection points. All have been divided into building usage and system type. For each of those the percentage of approved, fault to be remedied before next inspection (FA), and fault that made the inspection to fail an approval (FF). As this is impossible to read all the material have been grouped in 5 different classes. Faults including both FA and FF for the frequency of more than 50 % or 30 % of the inspections. Faults that fail an approval FF for the frequency of more than 30 %, 20 %, 10 %. For each of the inspection points this has been noted. The sum of each of those has been noted and added. This sum is given in the last line in table 9.

Table 9 Number of inspection positions giving faults. The number of Inspection Positions that Fails are added and called "[IPF]"

	Apartments				Schools				Day nurseries						Butik		Kontor			
	MSEX	MSE	MEO	PSV	MSEX	MSE	MEO	PSV	MSEX	MSE	MEO	PSV	Hospitals	Med. care	MSEX	MSE	MSEX	MSE	MEO	PSV
Fault. Incl both fault that cause failure of approval and fault accepted to be remedied before next inspection																				
No. insp pos. >50 %	6	11	5	11	3	4	2	5	3	4	4	7	5	3	1	8	4	5	7	10
No. insp pos. >30 %	6	21	18	13	5	8	5	9	4	5	9	9	12	4	3	19	10	20	16	13
Fault, failed to give approval of systems																				
No. insp pos. >30 %	5	7	4	8	0	1	0	1	0	0	0	3	1	1	0	2	1	2	3	9
No. insp pos. >20 %	5	9	4	9	1	1	1	3	1	2	0	3	4	1	0	3	2	3	9	11
No. insp pos. >10 %	9	19	8	11	2	5	3	9	1	2	2	4	9	3	0	11	7	15	15	12
Sum of inspection position failures	31	67	39	52	11	19	11	26	9	13	15	26			4	43	24	45	50	55

Figure 5 Fault/system for usage



The number of faults FA+FF in general was 5.3 faults/system for 1997 compared to 5.7 faults/system for 1995. If a system is not approved it has in general more than 3 faults that each cause a disapproval of the system, see figure 5.

In order to judge or rank the systems the evaluation has been made from different points of views. It is a qualitative ranking based on the inspections giving the number of approved systems and the frequency of faults for each inspection point. The ranking is from "1" to "4" with the best given the rank "1". Then the rank is added and a sum given. See table 10.

The result of this sum gives a fault ranking sum in the proportion

1 : 2 : 3 = MSEX : MEO,MSE : PSV

To be read as: the ranking with regard to the number of faults for inspection positions and overall approval of systems. This gives that PSV has three times as high fault index compared to MSEX. Also that PSV can be expected to have faults in that magnitude compared to MSEX.

Table 10 Ranking ventilation systems				
Building use	**Vent. system**			
	MSEX	**MSE**	**MEO**	**PSV**
With regard to % of approved systems				
Day nurseries 1997	2	3	1	4
Day nurseries 1995	2	3	1	4
Schools 1997	2	3	1	4
Schools 1995	1	3	2	4
Offices	1	2	3	4
Shops	1	2	3	4
Apartments	1	2	3	4
With regard to the number of inspection position failures				
Day nurseries	1	2	3	4
Schools	1	3	2	4
Offices	1	2	3	4
Shops	1	2	3	4
Apartments	1	4	2	3
Sum of ranking all above. FI = fault index	15	31	27	47
Sum of ranking above except 1995. FI	12	25	24	39
Sum of ranks above except 1995 & shops. FI	10	21	18	31
Ranking total	①	③	②	④

Table 11 Most frequent faults (max possible score is 100)		
Inspection point	**Score**	**Place**
Air flow, (function)	90	1
Other functions	72	2
Handling instruction	57	3
Air intake, (supply)	30	4
Device, duct (function)	29	5
Drawings	29	6
Ducts (exhaust)	28	7
Fan (function)	26	8
Control	22	9
Duct (supply)	19	10
Device (exhaust)	19	11
Device (supply)	17	12
Filter (supply)	16	13
Fan (exhaust)	13	14
User opinion	13	15
Others (exhaust)	12	16
Fan (supply)	10	17
Filter (exhaust)	6	18

The most frequent fault that either fails to give approval of the system or can be accepted to be corrected to the next inspection has been identified. For that the frequency intervals have been used. In table 11 is given the ranking and the score. This score is given by: If the inspection point has resulted in either FA+FF in the frequency 50 %, 30 % or FF in the frequency 30 %, 20 %, 10 %. Every building usage and every system is included. If an inspection point fails for all building usage and all 4 systems the maximum number is "20". As there is 5 different frequency indication the total can be 5*20 = 100. In the table 11 is the most frequent failure "the function air flow". It is a sort of resulting fault for what can go wrong in many other positions in a system. As number 2 is "other functions". Under this heading is collected such as "dirt and disorder, bad organisation".

4.1 Costs

Estimations have been made for the cost. This is based on figures collected at the questionnaire to the housing companies (4.2), ventilation market analysis.
To upgrade the ventilation systems to the situation when it was installed an estimated cost for
Apartment buildings of 2 – 2.5 billion SEK (250 – 310 M$ or 230 – 280 MECU)
Non-domestic buildings 6 – 8 billion SEK (750 – 1000 M$ or 680 – 910 MECU)
Of this cost about half is still remaining to be undertaken.
Running maintenance cost, regular checking, and inspection is estimated to be 1.5 – 2 billion SEK (190 – 250 M$ or 170 – 230 MECU) per year to keep the achieved level after the upgrading of the ventilation systems

5. Conclusions

The estimated approved rate at the end of 1997 are:

- Schools 85 – 90 %
- Day nurseries 90 – 95 %
- Hospitals ≈ 40 %
- Medical care ≈ 40 %
- Offices ≈ 40 %
- Dwellings 65 – 70 %
  - Of which condos 85-90 %
  - public owned 75-80 %
  - private <50 %

- Fault index are in the proportion 1:2:3 = MSEX : MSE,MEO : PSV
- Stack ventilation has far more faults than any other system regardless installed in multi family buildings, schools or offices
- Repeated inspections decrease the number of faults and increase the approved rate
- The number of faults are 5.3 per system and the most frequent fault is the function of "air flow".
- The estimated remaining cost for upgrading of the ventilation systems to the standard at the installation is about 4 billion SEK and the yearly cost is 1,5 – 2 billion SEK.

6. Acknowledgement

The Swedish work was sponsored by the Swedish Board of Housing, Building, and Planning. The collection and data processing of the protocols that were made by Walter Cederholm and his colleagues at the consulting company JBS in Vällingby, Stockholm.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

AIRTIGHTNESS OF TIMBER FRAME BUILDINGS NOT HAVING A PLASTIC FILM VAPOUR BARRIER

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Airtightness of timber frame buildings not having a plastic film vapour barrier

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Synopsis

Good airtightness of a building can be achieved by the incorporation of an inner sealing layer for the exterior walls and roofs in the form of a plastic film, which also serves as a vapour barrier. However, if it is not wished to use plastic film as an inner sealing layer, then airtightness must be effected through the use of other materials or in some other way. This project has been concerned with investigation of a number of alternatives. It has been found that the arrangement most commonly used in Sweden today comprises polymer-based fibre sheets (which permit diffusion) and gypsum board, as alternatives to plastic film on the inside of the structure.

The work of the project has shown that, provided that the materials are airtight in their own right, it is possible to achieve as good airtightness with alternative materials as can be achieved with plastic film. This has been demonstrated in the laboratory and in three of five houses in the field investigation. However, the airtightness performance of the finished building can be very poor unless care is taken both in the design and in construction, and this was also revealed in the field work.

General

Good airtightness is very important for resistance to moisture (in order to avoid damage by moisture convection), thermal comfort, indoor air quality, controlled ventilation and good energy husbandry. Airtightness is very dependent on both the design detailing and the quality of workmanship. It must be possible actually to construct the features that have been designed. All the materials/layers used in the structure contribute to the final airtightness performance.

Plastic film has been used for airtightness and as a vapour barrier in stud wall structures in traditional buildings in Sweden in recent decades. With it, it is often possible to achieve an airtightness performance of 0,8 litre/(sm^2) or about 2,9 $m^3/(m^2h)$ at a differential pressure of 50 Pa, as required for residential buildings by the Swedish Building Regulations [2]. If, however, the use of plastic film as an inner sealing layer is not wanted, then airtightness must be effected through the use of other materials or in some other way.

In addition to good air quality, it is also very important when selecting materials that the degree of diffusion protection is appropriate to the requirements of the building.

The objective of this project

The objective of this project [1] has been to:

- identify, in a field investigation, airtightness problems in wooden stud wall structures not having plastic film vapour barriers
- develop modified or alternative ways of providing airtightness, and
- evaluate the proposed solutions in the laboratory.

In the project, we have investigated the following alternatives to the use of plastic film for producing airtight layers in a wooden stud wall design:

- diffusion-permitting polymer-based fibre sheets (sometimes referred to as 'windproof' sheets)
- gypsum board panels
- wind barrier paper in some laboratory trials.

Diffusion aspects are not considered in this report.

The field study

The field study investigated six buildings in southern Sweden. Three of them were detached houses, and three were schools

When selecting the buildings to be investigated, it became apparent that diffusion-permitting polymer-based sheets are often used as an alternative to plastic film. Gypsum boards, without any further additional internal sealing layer, are also often used as an alternative to plastic film. The reason given for this is because plain paper sheets have been found to be more difficult to work with than 'windproof' diffusion-permitting polymer-based sheets. In addition, the polymer-based sheets are supplied in wide rolls, which reduces the number of joints needed.

For the field study, we therefore selected buildings having either an internal sealing layer of gypsum board alone or with a diffusion-permitting polymer-based sheet layer. Table 1 indicates the types of materials and designs employed.

The design details for ensuring good airtightness at connections, penetrations and joints are not shown on the drawings of the buildings. Many of these details are decided by the construction workers and/or by the site management.

We monitored the buildings during construction, looking at such aspects as detailing of joints at ceilings, floors, intermediate floor/ceiling structures, windows, doors and ground floor structures, as well as at the way in which joints and penetrations were made. After completion, the airtightness of the buildings was measured and leaks were traced.

Table 1. Materials and airtightness sealing principles in the field study buildings. Thermal insulation in all buildings has been provided by cellulose fibre (loose fill insulation).

Building no.	Type of building	Sealing layer	Sealing principle
1	Detached house, 1,5 storeys	EF Windproof, internal gypsum board	Stapling, taping
2	Detached house, 1,5 storeys	RW Windproof, internal wooden panels or gypsum board	Stapling overlapping
3	Detached house, single-storey	1 layer of gypsum board	Joints over studs and steel angles filler, mastic
4	School, single-storey	EF Windproof, internal gypsum board	Taping
5	School, single-storey	Ceiling: RW Windproof, internal gypsum board Walls: lightweight concrete	Stapling
6	School, single-storey	Walls: 2 layers of gypsum board Ceiling: RW Windproof, internal gypsum board and wood wool sheets	Stapling, taping

Our observations in the field trial buildings showed that there are several ways in which the same detail can be made: there are good ways and there are less good ways. The main areas in which there is scope for improvement are

- penetrations
- roof truss joist joints
- intermediate floor/ceiling structure joints
- ground floor joist joints (where the building has a wooden floor).

In addition, improvements could be made by developing designs that would enable penetrations to be avoided (e.g. by running unbroken sealing layers past internal walls, glulam beams, intermediate floor/ceiling structures etc.). In particular, it is the original design that is important in this respect.

The airtightness measurements were made in the spring and summer of 1997. Leaks were traced using a thermal imaging camera and air flow velocity meters. The results are shown in Table 2, from which it can be seen that two of five buildings have an air leakage that exceeds the permissible value in the Swedish Building Regulations. The other three buildings, of which one uses gypsum boards as its sealing layer, fulfil the requirements. However, it is not completely clear whether the buildings are sufficiently airtight to provide the necessary degree of resistance to moisture.

The airtightness measurements indicate that it is possible, using the same building materials, either to produce airtight structures or, equally, to have considerable air leakage, depending on the particular technical designs and on the quality of workmanship. Building no. 4, for example, has a low air leakage, while buildings nos. 1 and 2 have high air leakage.

The airtightness measurements of building no. 3 show that good airtightness can be achieved using only gypsum boards as the internal sealing layer. This is probably further underscored by the results from building no. 5, as our observations during construction revealed shortcomings in sealing of 'windproof' paper. However, the filling and painting work was carried out carefully, and so the relatively good airtightness performance may be due to the effect of this layer.

Table 2 Measured air leakage in some of the buildings in the investigation. The results shown are the air leakage through the building envelope at a differential pressure of 50 Pa between interior and exterior. The value is a mean value of air leakage at 50 Pa negative pressure and 50 Pa positive pressure.

Building no.	Measured air leakage, $m^3/(m^2h)$	Building Reg. Requirements $m^3/(m^2h)$	No. of storeys	Sealing layer
1	5.4	about 2.9	1.5	'windproof' + plywood
2	8.5	about 2.9	1.5	'windproof' + wood panel
3	2.4	about 2.9	1	gypsum board
4	1.7	about 5.8*	1	'windproof' + gypsum board
5	3.0	about 5.8*	1	'windproof' (lightweight concrete walls)

* The Building Regulations permit twice the rate of air leakage for other types of premises. The lower rate for residential buildings is in the interests of energy conservation.

Laboratory measurements

Working with a reference group, a number of proposals for design details for connections, joints and penetrations were developed. They were regarded as being capable of providing good airtightness, having a good likelihood of being properly made (i.e. good workmanship) and having benefits for production in general. They were tried out in the laboratory on large building elements consisting of walls, ground floor structu4res, intermediate floor/ceiling structures and roof truss joists (see Figure 1). The material combinations tested for internal sealing were:

- two layers of gypsum board with staggered joints, with fibre tape over the joints
- gypsum board + 'windproof'
- gypsum board + plastic film (to provide a comparison with a sealing layer that we know from experience is capable of providing sufficient airtightness).

The air leakage rate was measured for several different pressure differences, both positive and negative, across the building elements. Figure 2 and Table 3 show the measured values of air leakage for the three elements.

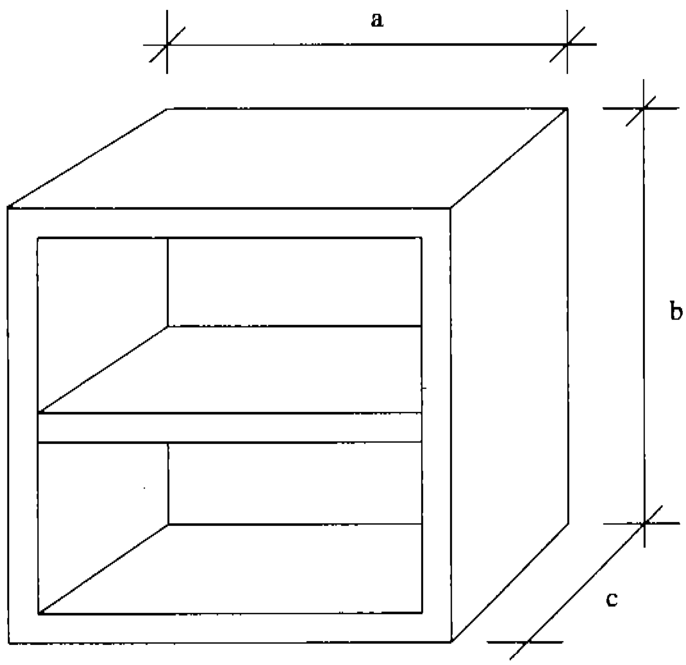

Figure 1. Test arrangement in the laboratory, consisting of walls, ground floor structure, intermediate floor/ceiling structure and a roof structure.

Table 3. Air leakage at 50 Pa differential pressure: mean values of positive and negative pressure. The upper figure indicates the measured leakage of the laboratory test elements, while the lower figure is the corresponding values for normal height walls, for comparison with the field measurements.

Material-combination	**Ground floor** Air leakage $m^3/(m^2h)$	**Intermediate** Air leakage $m^3/(m^2h)$	**Attic** Air leakage $m^3/(m^2h)$
Double gypsum boards	0.59 (0.35)	0.45 (0.26)	0.63 (0.37)
Gypsum board + 'windproof'	0.67 (0.40)	1.27 (0.75)	0.89 (0.53)
Gypsum board + plastic film	1.04 (0.62)	0.73 (0.43)	0.68 (0.40)

The measured values are quite low, and it does not seem to be particularly important as to which type of sealing layer - i.e. gypsum board, plastic film or 'windproof' - that is used. The values are also low in comparison with the field measurements, but it must be remembered that these laboratory elements contained no windows or penetrations, which could contribute to higher air leakage rates.

The table shows that it is possible to make joints with low air leakage, and that the proposed detail designs work well. Admittedly, there is some slight variation in air leakage from one to another of the different joints, but on the whole they are very similar. The element with the double gypsum boards provides the tightest connections when combined with a fibre tape stapled locally over the joints. The joints in the plastic film and 'windproof' elements are detailed in the same way. The differences noted can be due to variations in workmanship.

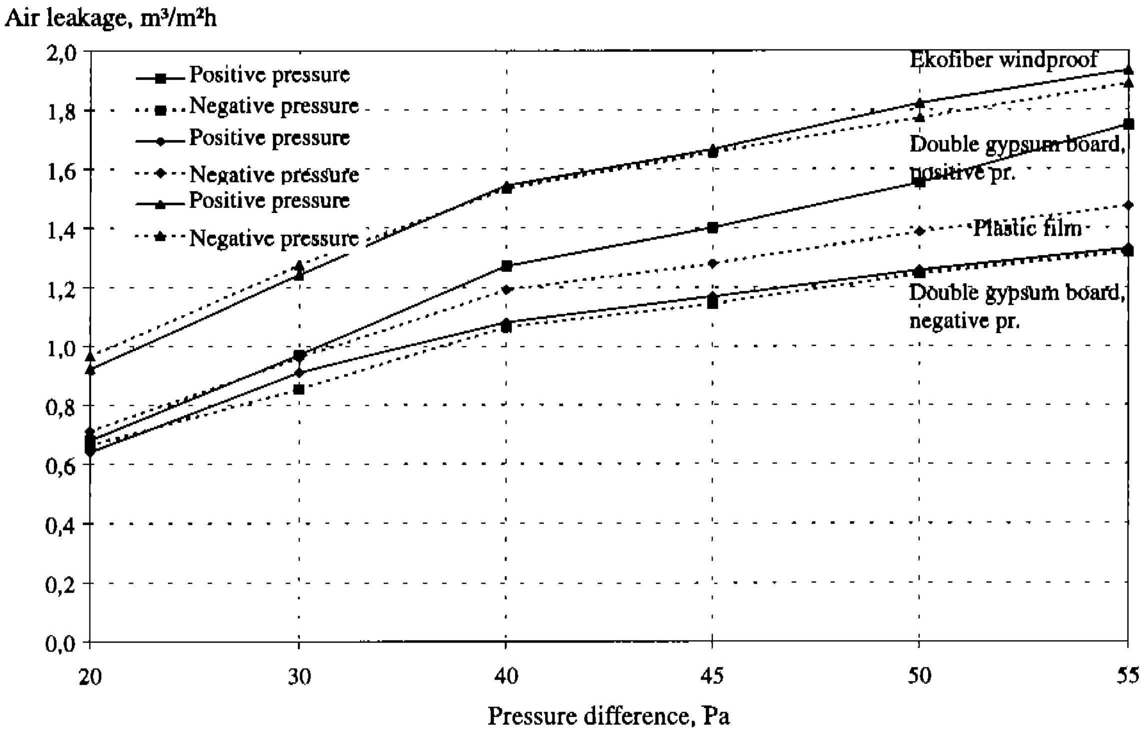

Figure 2. Example of measured results from the laboratory airtightness tests of elements having sealing layers of plastic film, fibre sheet and double gypsum boards with fibre-taped joints respectively. Test arrangement as shown in Figure 1.

The proposed connection designs consist of a number of variations of joints. These joint designs, together with a number of ordinary joints as used (for example) for joints in the middle of a wall, have been tested separately in a smaller test rig. These tests were also conducted using different types of sealing layers: plastic film, 'windproof' board in some cases and gypsum boards.

The comparisons of different materials in the same type of joint design show that, in principle, all the joints are equally airtight regardless of the material used. There are some differences, but they may be due to minor variations in the quality of workmanship. The windproof paper has the greatest leakage through the joints: this may be due to the fact that it is somewhat stiffer than the other materials, and so produces a poorer seal when overlapped.

The measurements also show that making an overlap joint, and then securing the overlap with a wood strip or gluing a sealing strip to it, produces a completely tight joint.

A single layer of gypsum board jointed over a stud produces a poorer joint than any of the other materials jointed with an overlap. However, if the gypsum board butt joint is then smoothed with filler, the seal is as good as that for the other materials. Double layers of gypsum board, with staggered joints, provide good sealing.

Comparison between the measured air leakage rates for the different types of joints and corresponding details in the large elements shown that the latter behave in approximately the same way.

Results and conclusions

The project shows that it is possible to achieve the same good airtightness with 'windproof' paper and gypsum boards as with plastic film. There is only slight air leakage through the two alternative materials ('windproof' paper and gypsum boards): this has been demonstrated in the laboratory and in three of the five buildings in the survey. However, poor design and poor workmanship can result in very poor airtightness in the finished building, as also shown in the field survey.

After interviews, investigation of the six buildings during construction and laboratory tests, the following conclusions can be drawn regarding the airtightness of buildings not having plastic film air and vapour barriers:

- If *the original design* is poor, then much of the sealing will have to be worked out and applied at the site, and the results will not always be good. However, it may be possible to work out how to deal with details, connections and joints at site planning meetings. This also applies if plastic film is used.

- *Work and time planning* (planning work at site) must recognise that good workmanship takes time.

- *Understanding of the need for airtightness* at building sites needs to be improved (although this is often properly understood when building using plastic film).

- *The airtightness should be measured, or be checked as the work progresses.* This would give greater awareness of the importance of good joint designs.

- In the buildings investigated, the main points where improvements were needed were found to be at penetrations, at ceiling joist joints, at intermediate floor/ceiling joints and at floor joints where there was a wooden floor. Good planning, with minimisation of the number of penetrations through the sealing layer, would improve airtightness.

- The laboratory measurements show that designs can be produced that, with proper workmanship, can reduce air leakage to essentially the same level as that produced by the use of plastic film. Plastic film provides satisfactory airtightness if properly applied. The materials that have been considered in this investigation as alternatives to plastic film are those that have good inherent airtightness, and so the work has been concentrated primarily on the design and quality of the joints. This conclusion cannot be applied if a material is used that permits air to permeate through it.

- Comparison of the laboratory results for designs based on plastic film, a plastic fibre fabric (having low permeability) and double gypsum boards as the airtightness layer shows that:
  - designs with a plastic fibre fabric give essentially the same results as those with plastic film. Minor differences are probably due to the quality of workmanship.
  - designs with double gypsum boards (combined with fibre tape over the joints) give approximately the same results as those with plastic film.
  - designs with single gypsum boards and with the joints sealed with filler give the same results as a similar design with plastic film.

- The long-term performance of buildings having gypsum board sealing layers needs to be investigated in a separate project, as should that of designs based on taping of joints etc. The project described in this paper has shown that alternative systems can provide good airtightness, but it is important also to show that this airtightness is long-lasting.

Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19^{TH} ANNUAL AIVC CONFERENCE
OSLO, NORWAY 28-30 SEPTEMBER 1998

A proposal for the classification of
the cleanliness of new ventilation systems

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0. SYNOPSIS

The Finnish Society of Indoor Air Quality and Climate has prepared a proposal for the classification of the cleanliness of new ventilation systems and components. The document supplements the classifications of indoor climate, construction cleanliness and material emissions published in 1995.

The classification of the cleanliness of the ventilation systems consists of two parts: a classification of the cleanliness of ventilation components and a guideline for the design and construction of clean ventilation systems.

The proposal will undergo an open review during next winter. After necessary modifications, the final version of the classification will probably be published in the spring of 1999.

1. BACKGROUND

1.1 The Finnish IAQ classification system

The final quality of indoor climate is influenced simultaneously by heating, ventilating and air-conditioning systems and equipment, by the performance of the construction and materials, and by the operation and maintenance of the building. To achieve a good indoor climate, all the guidelines presented in the classification system need to be taken into account throughout all the phases of design, construction and operation. The classification systems consist of three parts. It is intended to be used during the design and contracting of construction works and mechanical systems for buildings, and in the manufacturing of equipment and materials to build healthier and more comfortable buildings. The classification system can be applied to new buildings and for evaluation of all buildings and, when applicable, also during renovation. The classification system gives target and design values for indoor climate and supports the work of clients, designers, equipment manufacturers, contractors and operation personnel. The classifications can be referred to when writing up specifications of construction and mechanical systems. They can be used even as an attachment to such specifications. The classification system does not overrule official building codes or interpretations of them.

The selection of the categories for the Classification of Indoor Air Climate, Construction, and Finishing Materials /1/ has to be done at the beginning of a construction project. The building owner selects the categories with the design team. With the help of the Classification of Indoor Climate, the limit values for indoor climate are specified. After this, the category of cleanliness is selected according to the Classification of Construction. The Classification of Finishing Materials is used when selecting the building materials.

The Classifications of Indoor Climate and Finishing Materials have three categories, and the Classification of Construction Cleanliness two categories. Indoor Climate category S1, Construction Cleanliness category P1, and Material category M1 correspond to the best quality. Categories S3, P2 and M3 are in line with the minimum requirements set by building codes and regulations.

When the ultimate goal is good indoor climate, the best category of each part of the Classification has to be selected. a low category in part cannot be totally compensated by a high category in another part. Thus, for example, the high emissions of building materials can not be totally compensated by increasing the ventilation.

1.2 The need for the classification of the cleanliness of ventilation systems

The above discussed Classification of Indoor Climate, Construction, and Finishing Materials has been adopted for several construction projects. Construction clients and designers have used it as a tool in setting target values for indoor climate and in achieving the goals during the construction. 2,000 copies of the classification document have been distributed to various parties involved in construction. The document has also been translated into English. The principles of the classification have been used in several building projects. Some of the best known are: Nokia Headquarters Building in Espoo, Siemens Nixdorf Office Building in Vantaa, Neste Oy Headquarters Extension in Espoo, Apartment Building Puijonkartano for respiratory patients in Kuopio, Secondary School Eestinkallio in Espoo, and the University of Helsinki in Helsinki.

The Classification has been taken positively by the construction industry and the manufacturers of building materials. The first part of the classification system, which deals with the target values of indoor climate, has been used widely by designers in various building projects. The target values have also been used as reference values in the building investigations. The classification of building materials has been a success, too. At present time (August 1998) there are over 180 building materials in the best emission category (M1).

The second part of the classification system, the one dealing with construction cleanliness, has faced strong opposition from contractors. The main reasons to this are the unawareness of the nature and origin of the indoor climate problems and, on the other hand, unwillingness to change existing building practices. Now, as the major HVAC consultants have adopted the principles presented in the classification system in their design, it seems that the document is gradually changing the construction practice. It is, for example, more and more common that designers specify that the ducts have to be cleaned after the manufacturing process and handled on the construction site with capped ends. The largest project in Finland where the classification of the construction cleanliness has been used is the Nokia Headquarters building. 25 km of cleaned ducts were installed in the building, and the classification was followed in other aspects as well. The experience from this and other similar projects has pointed out the need for a separate classification for the cleanliness of the ventilation systems.

The need for improving the cleaniliness of ventilation systems stems from many Finnish and international studies. For example, the EU-funded IAQ Audit project covering over 50 office buildings in 10 countries found the ventilation systems to be the main source of odours in offices.

The surface area of a ventilation system can be 10-20 % of the total area of interior surfaces. The cleanliness of the inner surfaces of the ventilation systems has a significantly higher impact on the air quality than the cleanliness of the interior surfaces. Yet, the standard of cleanliness in the supply air system is poor compared to that of the interior surfaces.

Because the air handling systems can also be significant sources of pollution there is a great need to expand the classification to cover also components of the air handling systems such as ducts, filters etc. Finnish manufacturing industry has been very interested in developing new, cleaner components for ventilation and air-conditioning systems. A classification scheme is a good means to promote such product advancements.

2. THE PRINCIPLES OF THE CLASSIFICATION OF VENTILATION SYSTEMS AND COMPONENTS

2.1 The factors affecting the quality of suppy air

The quality of the supply air is affected by the quality of outdoor air, air handling (filtering, humidifying etc.) and the cleanliness of the ventilation system. The pollution of the outdoor air can be reduced by proper placement of the building in the site, good design of ventilation openings and efficient filtering. Handling of air usually improves the thermal climate and the quality of supply air, but reverse examples exist as well. Poorly maintained humidifiers can, for example, cause serious health and problems. In existing ventilation systems, the cleanliness is achieved by good maintenance of filters and other critical components, and regular cleaning of the ductwork. The project described here focussed on the cleanliness of new ventilation systems in new or refurbished buildings. A classification system for the maintenance procedures and duct cleaning is being planned as well.

A good classification system should have simple and easily measurable criteria. In ventilation systems, poor quality of supply air is usually best observed with sensory odour evaluation. The chemical indicators (e.g. TVOC) do not show a significant difference between good and bad quality of supply air. Therefore, the classification of emissions from building materials can not be used to evaluate the cleanliness of ventilation systems. In addition to the differences in pollutants (sensory/VOC), the air velocities and the sorption characteristics in ventilation systems are very different from those of building materials.

The knowledge about the sources of odour and other pollutants in ventilation systems is vague. The classification system has to deal with this by allowing modifications when new information is gained. Such information is currently being developed in a large Finnish project "Clean ventilation system".

2.2 The principles of the classification

The classification system gives target and design values for ventilation systems and its components. The classifications can be referred to when writing up specifications of ventilation systems. The classification system does not overrule official building codes or interpretations of them.

The classification is tool with which construction clients can formulate their wishes about cleaner ventilations systems. For component manufacturers the system is a means of declaring the cleanliness of their products (compared to competing products). The ventilation system contractors get information of new, cleaner installation procedurs.

The general target of the classification system is to ensure good quality of supply air. The supply air should not contain hazardous substances or harmful odours from the ventilation system. The following contaminants have been discussed in the classification:

- substances that are harmful to health
- microbes (mould and bacteria)
- man-made mineral fibres
- odours
- dirt, dust and particles

The classification of the cleanliness of the ventilation systems consists of two parts:

- a classification of the cleanliness of ventilation components /2/
- a guideline for the design and construction of clean ventilation systems /3/

The classification of the cleanliness of ventilation components is done according to the same procedures as the testing of the other properties (flow characteristics, noise etc.) –by third-party laboratory measurements and quality control agreements leading to official acceptance. The manufacturer declares the cleanliness properties at its own expense. There should be no need to check the cleanliness of the components on the building site. The checking of classification labels and the condition of the packaging should be enough.

The guideline for the design and construction of clean ventilation systems aims at maintaining the cleanliness during the construction process. The guideline can (wholly or partially) be referred to in the system specifications. It contains simple cleanliness criteria that can be measured in the commissioning of the ventilation system.

3. THE CLASSIFICATION OF THE CLEANLINESS OF VENTILATION COMPONENTS

3.1 General criteria

There is only one category for the cleanliness of ventilation components. A component either fulfills the criteria or not. Experience from the classification of building materials shows that manufacturers are not interested in other categories but the best. The lack of knowledge on the basic factors affecting the quality of the supply air is also behind the use of only one category. In the future it is easier to add new categories, if new infomation so requires.

The ventilation component should meet the following general criteria in order to be classified:

- The component shall as new not increase the amount hazardous substances in the supply air.
- The component shall as new not emit odorous or gaseous pollutants to the supply air, and thre shall be no visible dust on its inner surface.
- The component shall not increase in the ventilation system the growth of substances that are harmful to health or comfort.

The abovementioned criteria are deemed to fulfill if a component fulfills the criteria set for its component group. These criteria have for the time being been set only or the most common component groups and materials. Ducts, fittings, dampers and air terminal devices represent the largest surface area of the ventilation system. The other important potential pollution source in the ventilation system is the filter. The criteria for these two product groups have beed described in detail in the classification. The criteria for other components are on a more general level and they are not described in this paper.

3.2. The criteria for ducts, fittings, dampers and air terminal devices

These criteria apply for ducts and parts that have been manufactured with the conventional technology (sheet metal with mineral oil based lubricant). The general requirements described in 3.1 should be applied to other produtcs.

The ducts, fittings, dampers and air terminal devices should meet the following general criteria in order to be classified:

- Amount of oil in the inner surface of duct shall be less than 100 mg/100cm^2 ([1]
- Amount of oil in the inner surface of fittings, dampers and air terminal devices shall be less than 500 mg/100cm^2 ([1]
- Amount of man-made mineral fibres (MMVF) in the inner surface shall be less than 0,01 f/cm^3 ([2]
- Amount of dust in the inner surface of the ventilation system shall be less than 5% (tape method) or less than 0,5 g/m^2 (filter method)

For other duct materials than sheet metal manufactured with mineral oil lubricant it is possible to use an odour criterion instead of the oil criterion. According to the odour criterion, the acceptability of the supply air shall be better than 0,05 (scale -1... +1, untrained panel), or the odor intensity shall be less than 4 decipol (scale 0...20, trained panel).

In addition to the measurable criteria described above, each component group has also specific criteria dealing with the cleanablity, moisture control, tightness and use of sealants. These criteria have not been described here. In principle, the fulfillment of current Finnish and/or European standards (e.g. CEN ENV 12097) is required.

3.3. The criteria for filters

The task of the filter is very important in a clean ventilation system. The most important criterion for a filter is its removal efficiency. Therefore, a classified filter shall fulfill the efficiency requirements of its filter class throughout its lifespan.

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[1] The criteria is based on a relatively small sample of duct materials. This sample has show that the mentioned criterion is achievable in the manufacturing process and that the amount of oil does not significantly deteriorate the quality of the supply air.
[2] The general criteria for classification should be applied to other fibres than MMVF.

Air leakages in the filter and filter frame deteriorate the efficiency of the filters. A classified filter shall comply with the following criteria for leakage:

Filter class	Test pressure, Pa	Total leakage, %
Poorer than F5/EU5	200	6
F5/EU5	400	6
F6/EU6	400	4
F7/EU7	400	2
F8/EU8	400	1
F9/EU9	400	0,5

An odour criterion for new (unused) filters requires that the acceptability of the supply air shall be better than 0,05 (scale -1... +1, untrained panel), or the odor intensity shall be less than 4 decipol (scale 0...20, trained panel).

The filter shall not emit mineral fibres (MMVF) into the supply air. The total amount of MMVF fibres from a new filter shall be less than 0,01 f/cm^3.

Oils, biocides and other potentially hazardous substances may not be used in a classified filter.

The filter and its frame shall bear the pressure difference that is created over a completely clogged filter. The filter shall not be in touch with the bottom of the filter envelope or any other potentially moist surface even when the system is not running.

4. GUIDELINE FOR THE DESIGN AND CONSTRUCTION OF CLEAN VENTILATION SYSTEMS

4.1. The cleanliness categories of new ventilation systems

Two categories are described for the design and construction of clean ventilation systems. The category is selected in the design stage of the system. The general requirements for both categories of the clean ventilation systems are the same as those for the components (see 3.1).

Ventilation systems in category 1 shall meet the following requirements:

- Air supply ducts, fittings, dampers and air terminal devices are made using cleanliness classified components. Less than 20 % of the system (calculated on the inner surface area) can be made from non-classified components provided that they have been cleaned from oil and dirt at the construction site.
- The sealants used in the system should be classified in emission category M1 or M2, or their emissions should be known to be low.
- Amount of dust in the inner surface of the ventilation system should be less than 5% (tape method) or less than 0,5 g/m^2 (filter method)
- No return air shall be used execpt in systems serving only a single apartment.
- The suppy air side is equipped with a cleanliness classified filter that has a removal efficiency equivalent of at least class F7/EU 7.

Ventilation systems in category 1 shall meet the following requirements:

- Air supply ducts are made using cleanliness classified components. The fittings, dampers and air terminal devices are made mainly from cleanliness classified components. Less than 50 % of the system (calculated on the inner surface area) can be made from non-classified components provided that they have been cleaned from oil and dirt at the construction site.
- Amount of dust in the inner surface of the ventilation system should be less than 10% (tape method) or less than 1,0 g/m^2 (filter method)
- Return air from spaces with similar pollutant loads may be used. This return air shall be filtered with a cleanliness classified filter that has a removal efficiency equivalent of at least class F6/EU 6.
- The suppy air side is equipped with a cleanliness classified filter that has a removal efficiency equivalent of at least class F4/EU 4.

The guideline gives requirements and instructions for various details in design and construction of the system. These include:

- Detailed design and construction requirements for the cleanliness of various critical components
- Guidance on the storage of the components on the building site
- Guidance on the installation of the components
- Instructions on the use of the system before commissioning
- Instructions on the use and maintenance of the system

For example, the following instructions are given for the installation of the system:

- The wrappings of the components shall be removed only just before installation (Category 1)
- The ingress of dirt in the system must be prevented during the installation work
- The inner surfaces of the ductwork shall be free of scrap, strathes, screws etc that can attach dirt and dust or make the cleaning of the system more difficult.
- Exessive use of sealants should be avoided
- All open ends of the ductwork shall be sealed dust tight during breaks in the installation work (Category 1) or
- All open ends of the vertical ductwork shall be covered during breaks in the installation work (Category 2)
- The ductwork shall comply with the Finnish tightness requirements (Category 1: tightness class C, SFS 4699. Category 2: tightness class B, SFS 4699).
- Functioning of the maintenance and cleaning openings (access, openability, work area, cleaning distance) shall be checked during the installation work.

5. DISCUSSION

The proposal for the for the classification of the cleanliness of new ventilation systems was finalized in June 1998 and submitted to Finnish Ministry of the Environment. Many details of the system are still under development, for example the measurement protocols for the oil, dust, fibres and odours. These will be finalized during autumn 1998. The proposal will undergo an open review during next winter. After necessary modifications, the final version of the classification will probably be published in the spring of 1999.

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The proposal for the classification of the cleanliness of ventilation systems was developed by a workgroup Pertti Pasanen (University of Kuopio), Marko Björkroth and Antti Majanen (Helsinki University of Technology) and Jorma Säteri (VTT Building Technology and FiSIAQ).The work was coordinated by professor Olli Seppänen (Majanen (Helsinki University of Technology). The project was financed by the Finnish Ministry of the Environment, where Chief Engineer Esko Kukkonen acted as project supervisor.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

THE JOULE-TIPVENT PROJECT: TOWARDS IMPROVED PERFORMANCES OF VENTILATION SYSTEMS

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Abstract for AIVC CONFERENCE 1998

The JOULE-TIPVENT project : Towards Improved Performances of ventilation systems

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Due to the improved thermal insulation of buildings (new as well as retrofitted), the energy consumption for covering the transmission losses has been substantially reduced over the last 2 decades. The same tendency is not observed when looking to the ventilation in buildings. In the future, it is possible that, in a number of countries, the specific energy consumption for ventilation will substantially increase, because of a wider use of ventilation systems and the request for higher ventilation rates (as can be illustrated by the discussions within CEN TC 156 'Ventilation'). For modern office buildings, more than 50% of the heating demand will be related to the ventilation losses.

It is crucial to improve the performances of ventilation systems, as well with respect to the design, the execution (e.g. better control of the air flow rates,...) and their use. Optimisation should focus on better energy performance AND better indoor climate conditions.

The TIPVENT Project aims to contribute to a substantial progress and has the following objectives:

1. To achieve a better understanding of the impact of air flow rate requirements found in standards on the energy demand of buildings (residential and office type buildings) and the existing background for the specification of the ventilation requirements.
2. To evaluate, by means of monitoring, for a selection of buildings (dwellings and offices) equipped with mechanical ventilation the level of agreement between the required, design and real air flow rates, between the required, design and real sound levels, between the required, design and real draught performances, between the desirable and real fan consumption, between the desirable and real air quality of the supply air and between the expected and real performances of heat recovery in practice.
3. To analyse in the participating member countries (south, central, north of Europe) as well as in some other countries the impact of standards and building regulations on the performances of ventilation systems.
4. To develop a really performance oriented approach for mechanical ventilation including procedures for on site performance checking. This approach must allow a better market penetration for innovative technologies;
5. To apply the performance concept on a representative range of systems and produce a set of guidelines.
6. To develop a number of smart designs for improved performances with emphasis on active acoustical insulation, medium pressure air cleaning, low pressure mechanical ventilation and intelligent fan control..

In order to achieve these objectives, an industry lead partnership has been set up combining manufacturers, designers, people strongly involved in standardisation and building regulations as well as organisations involved in research. The deliverables of the project will be directly implemented by the project partners but it is clear that the project will have a substantial spin-off on the whole ventilation sector, by implementing the methodology in ongoing activities on standardisation and building regulations.

The paper will briefly present the key objectives of the project, the working method and the first project results.

Specific attention will be given to highlighting the lack of performance approach in many existing standards. Also indications of possible improvements are presented.

Finally, the TIPVENT web-site will be presented including the presentation of the content of the website describing TC 156 'Ventilation'..

Please Note: The full paper will be printed in the Supplement to the Proceedings

VENTILATION TECHNOLOGIES IN URBAN AREAS

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FILTERS FOR GASEOUS CONTAMINANTS: PERFORMANCE MEASUREMENT AND IMPACT ON VENTILATION SYSTEMS

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Synopsis
Filters for gaseous contaminants which are used inside HVAC systems are characterised by means of rated air flow rate, air motion resistance, trend of the mass efficiency versus time and holding capacity of the considered gas. The determination of the characteristics cannot do without the use of experimental activities, even though many aspects seem to be foreseeable through calculation models based on general laws. A test rig for granular media which uses toluene in small concentrations in the air is here presented.

1. Introduction

Gas filters are used in HVAC systems in order to separate the odours and gaseous contaminants that are present in the air. They are usually used inside thermal-hygrometric treatment units in order to decrease that part of the air which is renewed and taken from outdoors, while assuming that outdoor air is purer than indoor air. From this point of view, gas filtration makes it possible to decrease the energy demand associated to renewal air, but, on the other hand, they require more electrical power due to their resistance to air motion and an initial economic burden due to the investment, maintenance and replacement costs.

In order to take motivated decisions on the air-conditioning system, it is necessary to characterise these filters regarding as far as the performance is concerned: rated air flow rate, air motion resistance, separation efficiency and duration for representative values of the air flow rate, temperature and air relative humidity.

The rated air flow rate for fixed frontal dimensions depends on the face velocity: typical values are around 3 m/s and, for this reason, it is not possible to arrange the filtering media in flat layers because of the consequent excessive resistance to the air motion using the thickness that is necessary to guarantee acceptable duration of the filtering device. Gas filters surfaces are therefore extended [1] and the filtering material is arranged in such a way as to form a layer that is non-perpendicular to the axis of the duct and crossed by the air with a velocity which is lower that the frontal velocity by one order of magnitude.

Resistance to the air motion is measured by pressure drop. This does not vary in a measurable way during the technical life of a gas filter when the air flow rate is constant: there is therefore no final pressure drop to indicate the necessity of replacement.

The separation efficiency is the ratio between the mass flow rate of the gas held (separated from the air for a certain period of time) and the corresponding mass flow rate of gas entering the filter. The criterion to establish that a gas filter requires replacing is based on the minimum acceptable efficiency. The filter life duration and holding capacity of a gas derive from the above mentioned criterion and assume that the trend of efficiency versus time is known.

The data indicated above have been known for some time [2], in the case of various couplings between gases and filtering materials and are the result of research into gas absorption phenomena inside fixed beds made of porous materials. The theory is at an advanced stage of development and allows one to plan chemical plants for processing purposes. In the air-conditioning filter field, which is distinct from that of chemical plants, some gas filter manufacturers are able to correlate data in order to meet their customers' needs, and to foresee the characteristics, the mass and the arrangement of the material necessary for each specific application. Typical input data are: the gas that should be separated, its concentration and the air flow rate. The duration of life (e.g. 1 year) and the resistance to air motion are conventionally established, but hardly ever is it possible to understand the criteria according to the choices made, which is probably due to understandable reasons of discretion.

However, this situation leads to a lack of diffusion of confidence among operators in the HVAC systems field and it explains why this paper considers the point of view of those who would like to use a gas filter, but who are not able to understand the common formulations of the principles of chemical engineering or willing to accept the results of an unknown calculation program used by the reseller of a product.

It is clear that from the point of view of plant management, it would be useful to have reliable and economic indicators of the trend of gas concentration downstream to the filter. From the commercial point of view, a standardised evaluation method for gas filters or, at least, for the absorbing material used inside the filters would be desirable.

2. Assessment method

In order to evaluate a commercial gas filter, designed to be used inside HVAC systems, it is necessary to first of all know the rated value of the air flow rate which is chosen on the basis of a compromise between the contrasting requirements of the air motion resistance and the life duration.

From the best comprehension of the filter functioning point of view, it is better to distinguish between the air motion resistance due to the shape of the filter from that due to the filtering material. The first depends on the previously mentioned requirement of surface extension and generates a pressure drop that is approximately proportional to the square of the air flow rate. The second may be reduced to the case of a flat layer crossed by an air flow rate with a macroscopically one-dimensional motion. It is possible to apply the considerations concerning beds of porous materials to this layer of material, thus one can conclude that pressure drop is proportional to the length and to the air flow rate, and it increases when the dimension of the material grains decreases. Therefore, for the sake of air motion resistance and electricity use, it would be favourable to use filters with large stretched surfaces and with thin thicknesses. However, it is necessary to ensure a reasonable period of time between one replacement and another.

The interval of time in which a gas filter absorbs has an insuperable theoretical limit (t^*) constituted by the absorption capacity of the material and provided by the relationship

$$t^* = \frac{m_f \cdot \left(\frac{m_a}{m_f}\right)^*}{\dot{V} \cdot c_M}$$

where:

m_f = mass of the absorbing material of the filter;

$\left(\frac{m_a}{m_f}\right)^*$ = ratio between the mass of the absorbed gas and the mass of the absorbent material;

$\dot{V}$ = air flow rate in volume;

c_M = gas concentration in volume of the air upstream to the filter.

The ratio $(m_a/m_f)^*$ is a theoretical value in thermodynamic equilibrium conditions which expresses the maximum quantity which may be absorbed in the previously mentioned conditions. This depends on the nature of the gas, material and the thermodynamical conditions: temperature and relative humidity of air are particularly interesting for HVAC systems, while pressure is that of the atmosphere. Moreover, concentrations are much lower (e.g. 1 ppmv) than those for which literature data are available (typically 1000 ppmv).

The previously mentioned theoretical data of life (t^*) is not of immediate practical interest because the acceptable values of downstream concentrations (c_V) are differently defined for each application but nevertheless must be $c_V < 0.1\ c_M$.

In practice, a scientifically based, comprehensible and feasible criterion is necessary. For this purpose, it is necessary to know, even though it is not sufficient, the gas concentration trend downstream to the filter. On the basis of this trend and on the downstream air pureness requirements, it is possible to establish when is the right time to change the filter: for example, when the downstream concentration has reached a pre-established fraction of the upstream value.

Practical criteria for evaluating the necessary mass of an absorbing material and its life are based on such assumptions.

The trend of filter downstream concentration versus time [4] is represented by curves of an experimental origin [5] or calculated through mathematical models [6], [7]: it is however important to know its potentiality and limits.

These curves arise from the necessity of experimenting filter performance concerning a determined gas. In a laboratory, it is usual to make an air flow rate, in representative conditions, cross the filter in which the gas of interest is injected and measuring the upstream concentration (which remains constant) and the downstream one. Sooner or later, the downstream concentration begins to increase and, if one waits long enough, it reaches the upstream value. It is obvious that this trend is valid only in test conditions but, if these conditions are representative, the information obtained may be of great use in practice. When it is not possible to experiment each condition of interest, it is necessary to be able to extend the results to different conditions, that is, a model of the filter is needed. There exist models based only on physics and chemistry laws and which are also able to reliably describe cases of practical interest, moreover, there exist empirical correlations based on experience and these are vital to the current state of knowledge.

In fig.1a, a gas filter is schematically represented: a one-dimensional system in steady state condition is assumed from the air flow rate point of view. When the challenge gas is injected into the air, it is possible to obtain an upstream concentration which remains constant in time and uniform over the whole cross-section because of the one-dimension hypothesis. As soon as the gas interacts with the filtering material layer, a concentration profile is established (fig.1b) which shows the upstream value on the left while downstream the value is null. The profile then moves towards right maintaining an l length. Fig.1c represents an intermediate condition in which the absorption zone separates the saturated material zone from that of the still fresh material. The absorption zone has an l length that results from the equilibrium between the gas flow rate and the filter separating capacity. It is clear that the L length of the filter must be greater than this zone, otherwise a downstream concentration which is not null would be obtained. Fig.1d shows the condition in which the absorption zone reaches the end of the filter and, for this reason, the downstream concentration begins to be greater than the null value.

From the point of view of the utilisation of the material, it would be better if the absorption zone could be short, at the limit of null thickness, so that, when it reaches the right end of the material, every part which is on the left is saturated. The previous description makes one think that a concentration wave would propagate inside the material with a certain velocity which is certainly much lower than that of the air (e.g. 0.03 m in 2000 h)

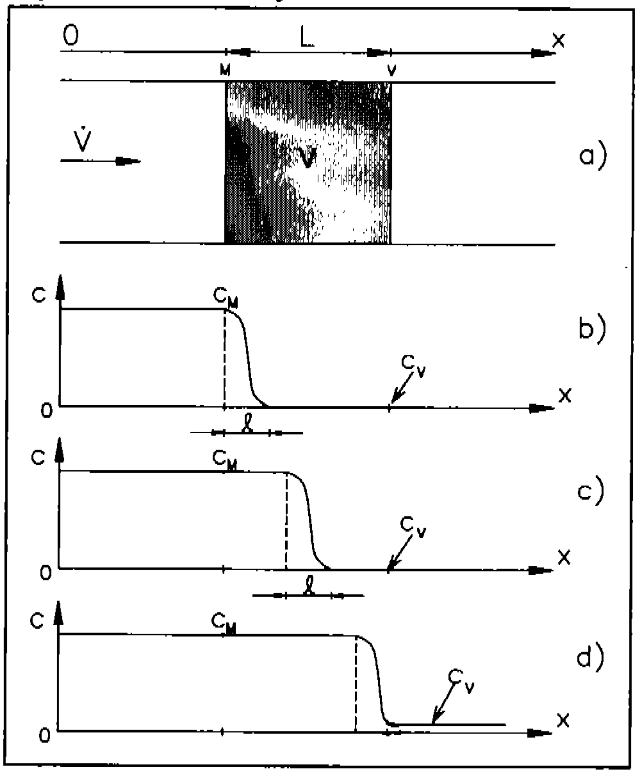

Figure 1- Concentration profiles along the filtering bed
(length L)
a) monodimensional (x) filter scheme (volume V)
b) concentrazion profile - starting point (t_0)
c) concentrazion profile - generic time instant (t)
d) concentrazion profile - ending point (t_f)

and which would determine the life time. In a generic instant, efficiency (ε) is defined as the ratio between the mass flow rate of absorbed gas ($\dot{m}_a$) and the upstream mass flow rate gas ($\dot{m}_M$) and is generally expressed by the concentrations as follows

$$\varepsilon\,(t) = \frac{\dot{m}_a(t)}{\dot{m}_M} = 1 - \frac{C_V(t)}{C_M}$$

The breakthrough (P), instead, may be written as:

$$P(t) = 1\text{-}\varepsilon(t)$$

where:
c_M = gas concentration in volume in the air upstream to the filter;
c_V = gas concentration in volume in the air downstream to the filter.

The efficiency curve versus time is therefore obtained from the concentration curves which represent typical results of laboratory measurements. An example of such experimental results is given in figure 2. The curves of the flow rate allow one to distinguish two zones that correspond to the mass of the separated gas (m_a) and the mass of the unseparated gas as this has penetrated downstream, m_p, and which is equal to the difference between the upstream mass and the separated mass m_p=$(m_M$ - $m_a)$. The check on the mass balance is simplified by using the average efficiency ($\bar{\varepsilon}$) during the considered period of time and defined by the ratio between the separated mass and the injected mass over a determined period of time, and expressible versus efficiency through the following ratio

$$\bar{\varepsilon} = \frac{m_a}{m_M} = \frac{\int_0^{t_f} \varepsilon \cdot dt}{t_f}$$

The average efficiency clearly depends on the efficiency curve trend (which expresses the attitude of the material to absorb the gas) and on the integration period (which depends on the conventional choice of the downstream gas concentration that determines, when it is reached, the technical life of the absorbing material of the filter).

As far as the efficiency trend is concerned, the most meaningful quantity is the residence time, defined referring to fig.1 as

$$t_p = \frac{V}{\dot{V}}$$

where:

V = space occupied by the filtering material (ratio between the mass and the apparent density);

$\dot{V}$ = volume air flow rate.

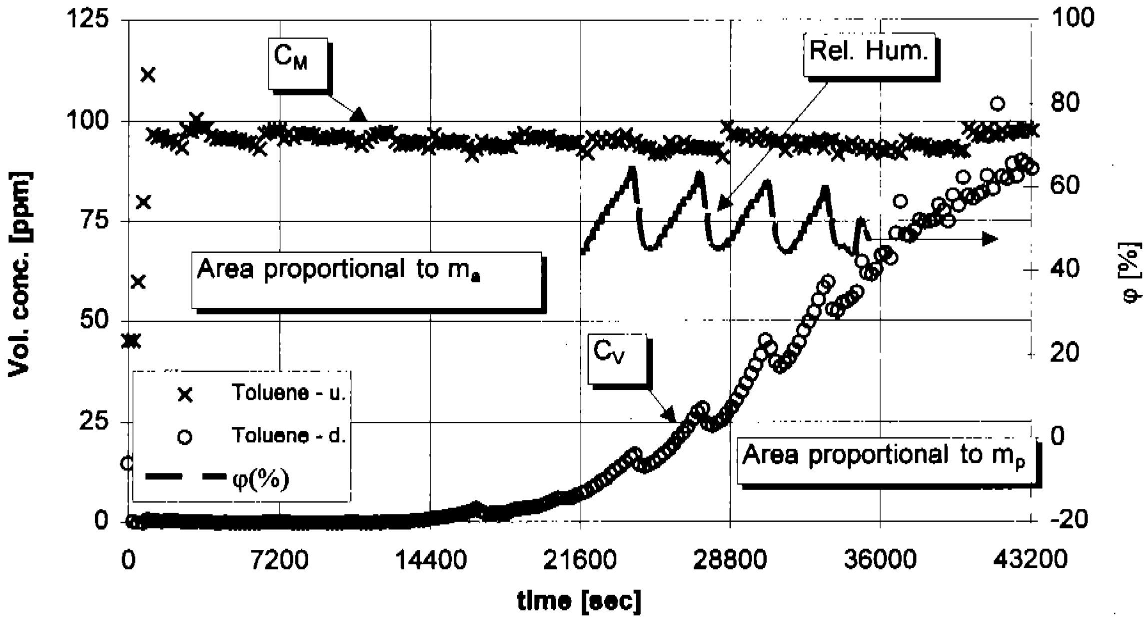

Figure 2 - Toluene volume concentration upstream (u.) and downstream (d.) the filtering material - Test conditions: T = 24 °C - φ = 50 %.

This definition, which is typical of one-dimensional systems, considers the filtering material as being homogeneous. For modelling purposes, the residence time needs to be corrected by taking the maximum and the effective apparent density into consideration, in order to be more significant [7].

The residence time t_p is also equal to the ratio between the thickness of the filtering material (L) and the average velocity of air passing through. It has been considered that the residence time t_p = 0,1 s is optimal: this data is presumably the result of experience in the HVAC systems field and would mean, for example, that if the air velocity is assumed to be 0.3 m/s, the layer of the filtering material should be L = 0.03 m thick.

Upstream concentrations in the case of filters for HVAC systems are usually low and the resistance to the transport of the mass is concentrated in the limit layer: this fact allows one to obtain simplified relationships for the case of interest.

Upstream filter concentrations are low because they are related to atmospheric and indoor air. Every study on this subject agrees on the fact that efficiency depends on the value of the upstream concentration and increases with it.

In order to explain this dependence, it is necessary to remember that absorbing materials are made of porous pellets of some millimetres in size. Inside each pellet are macropores, whose size is greater than 10^{-8} m, from which micropores, whose size is smaller than 10^{-8}, and often just a little larger than the molecular ones, spread out. The absorbing process inside the micropores causes the molecules to be so close to one another that they form a liquid which completely fills the micropores, while the molecules inside the macropores only cover the walls. It can be imagined that after the concentration increases liquids form and, therefore, the possibility of absorbing increases.

The transport of the mass from the air to the walls depends on the difference of the concentrations. With a thermal analogy, it is possible to say that the gas particles inside the air flow have to overcome two physical resistances in order to reach the wall and possibly one chemical resistance in series: the first through the limit layer, the second corresponding to the wall surface, and the third one in the chemical reaction with a reagent, if any. In this case, the first resistance usually prevails while data for the transport of the mass coefficient are unavailable.

The gas that should be separated must be representative of the actual applications: a distinction is made between two main categories (organic and inorganic contaminants) is made which corresponds to outdoor and indoor contaminants. As a limited number of gases has to be chosen, nitrogen oxide and sulphur oxide are considered, for example, for the first group, as they are particularly representative of what occurs in urban and industrial environments (applications that are more considered nowadays are those for expensive computers inside petrol refineries and those for museums). Among the organic contaminants toluene is considered as, from among the various VOC (volatile organic compounds), it has an intermediate boiling temperature. This boiling temperature affects the formation of liquid in the previously mentioned micropores.

The interaction between a gas and a certain filtering material depends on the air characteristics, in particular on its pureness, temperature and relative humidity.

In some cases it is perhaps possible to take all the above mentioned variables into account in a calculation model. However, the need for data on the transport of mass coefficients, equilibrium thermodynamic conditions at low gas concentrations and the motion inside granular beds [6] still remains. On the basis of the present situation it is necessary to proceed with an experimental method having to be satisfied with tests in only some conditions which are considered meaningful and being obliged to renounce a global vision of the whole subject.

If a standardised test method could be used it would be possible to compare different commercial products.

An experimental test rig which is suitable for characterising materials used to absorb gases is described in the following part. The small scale test allows to characterise the material but the results obtained in such way are not necessarily valid for filters made with this filtering material.

3. Experimental test rig

The complete test rig is sketched in fig.3: the filter is made of a porous material layer whose diameter is 0.05 m. The layer is crossed by an air flow with a 100 ppmv upstream concentration of toluene. The thickness of the layer of the filtering material (0.03÷0.1 m) is chosen in such a way as to obtain a residence time $t_p = 0.1$ s with an air velocity 0.3÷1 m/s.

The air for the test is taken from an air-conditioning unit where it is possible to set the values of temperature and relative humidity; this air is derived from a fan which maintains the duct below the external pressure.

The filter under test is crossed by an airflow rate in prefixed conditions during a preliminary period of time in order to "condition" it. The concentration of water vapour is measured upstream and the constancy and consistency are checked in relationship to the

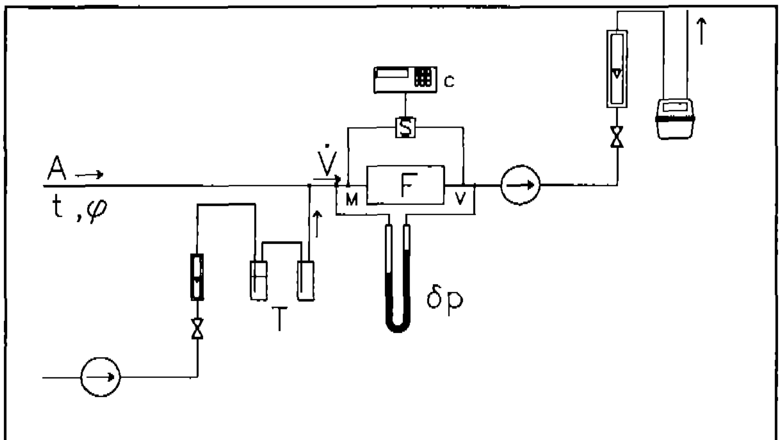

Figure 3 - Experimental apparatus
A, air -T, toluene - S, pneumatic scanner - C, gas analizer.

prefixed value of relative humidity set on the air-conditioning unit.

The trend of the concentration of water vapour is checked in order to find out when the equilibrium conditions are reached. The time necessary for this operation may be very long (24 h) and this fact implies practical problems inside a laboratory. On the other hand it should be remembered that absorbent materials have such large relative surfaces (1000 m^2/g order of magnitude) that they are very difficult to obtain and guessed at.

The sampling is made upstream and downstream to the filter under test by means of two flexible pipes made of non-reactive material which lead to a distributor. This distributor switches between the upstream and downstream line every 60 s and sends the samples to an analyser which gives the carbon dioxide, water vapour and toluene concentration values. During this phase the absence of toluene in the air is controlled. At this point it is possible to inject the challenge gas, which, in the present case, is a superheated steam.

Toluene is liquid at ambient conditions as at atmospheric pressure the temperature of saturation is 110 °C. It is contained in a glass recipient that is placed in a thermostatic bath which is kept at a temperature lower than the ambient. A small airflow rate is bubbled through the liquid so that a saturated vapour which is being superheated exits from the first recipient and passes through the second, which is at an ambient temperature. The mixture of air and toluene vapour is sucked into the main duct in which it is mixed with air. The upstream concentration is kept constant by controlling the temperature of the bath and the air flow rate, which is bubbling.

The trend of the concentrations upstream and downstream to the filter is obtained as shown in fig.2 where an example of experimental data of concentrations in conditions which can be interesting for HVAC systems is reported. In this case the temperature is constant (T = 24°C) and the relative humidity varies in a cyclic way over a 50% value. The absorbent layer is made of commercial activated carbon and its length is L = 0,06 m while the air has a residence time of t_p=0,1 s (challenge concentration of Toluene about 100 ppm).

4. Experimental results

Adopting the experimental apparatus described in the previous paragraphs, some tests have been performed. This first experimental campaign was aimed, mainly, to evaluate the performances of the measurement test rig and to investigate the capability of the system to perform sensitivity analyses on filtering media. The working activity is still under way, however, some preliminary results are already available.

Figure 2, as mentioned before, shows a typical filtering media behaviour and underline the critical factor represented by the influence of relative humidity, φ, on the filter efficiency.

The penetration of toluene downstream to the filter begins after 10000 s, that is, a

perceptible increase of concentration c_V is obtained. In particular it is possible to observe that the downstream concentration increases in correspondence to the relative humidity peaks.
This phenomenon shows that the test rig is sensible to the variation of at least one of the important quantities. Precision is required in the case of the balance of the toluene mass and in the present case a mass m_M = 21,8 g of toluene over 43000 s was injected, while maintaining an approximately constant upstream concentration of c_M =100 ppmv in the air flow. The increase of the mass of the filter corresponds to the c_V curve (whose integration in time produces a held mass of m_a = 16 g). The average efficiency in the interval of the total time considered is $\bar{\varepsilon}$ = 0,73. The trend of the downstream concentration curve is the general trend for penetration (breakthrough) curves to which, in the case of fig.2, the disturb produced by the variations of relative humidity is overlapped. This can be explained by the absorption of the water molecules by the porous surface which is therefore less available for toluene absorption thus reducing the efficiency. The phenomenon is relevant when φ>50% as in these conditions the absorption in equilibrium conditions greatly increases [7]. Examples of breakthrough curves are shown in figures 4, 5 and 6. In these charts is possible to see the influence of the filtering bed thickness, relative humidity and test gas concentration on the filter performances.

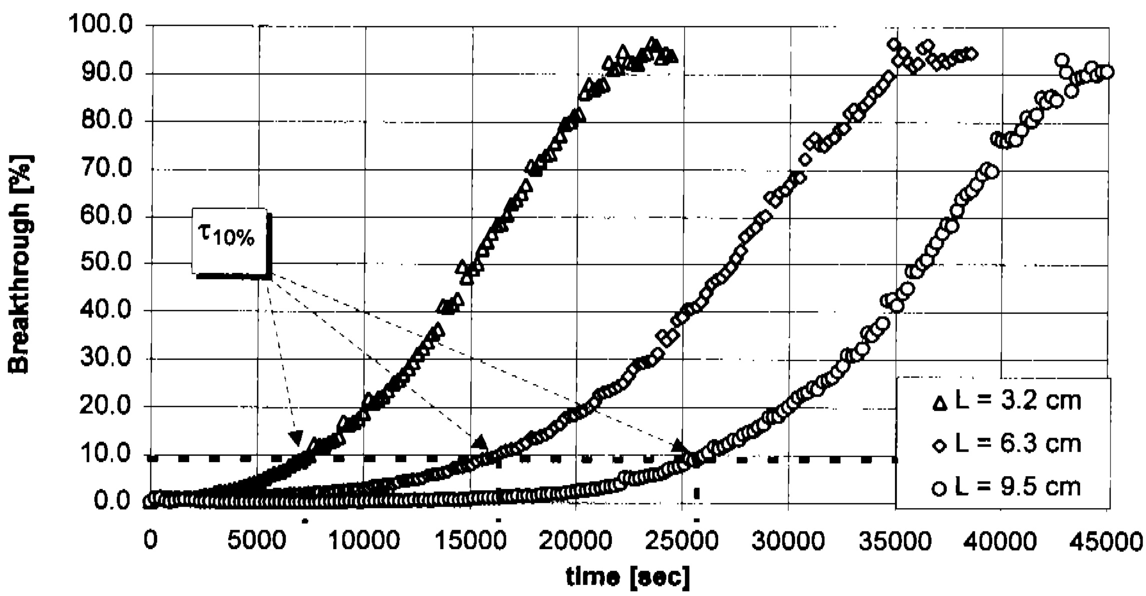

Figure 4 - Breakthrough curve - T = 23°C, φ = 50 % - Influence of bed thikness, L.
($\tau_{10\%}$ = breakthrough time corresponding to 10% of penetration)

The usefulness of these curves, which can be obtained in the laboratory, depends on the realism of the conditions. The weakest point is represented by the high upstream concentrations (100 ppmv in order to not have tests that last too long) but relationships which make possible to correct this distortion exist [4].

Figure 4 refers to constant air temperature (T = 23°C ± 1°C) and relative humidity (φ = 50% ± 3 %); it is possible to see that the filter efficiency strongly depends on parameter L and, in particular, it increases increasing the bed thickness. On the opposite, charts on figure 5, related to a fix bed thickness of 6.3 cm, point out the weak influence of relative humidity φ, on the filter behaviour. This fact may appear in contrast with fig. 2, however, it must be kept

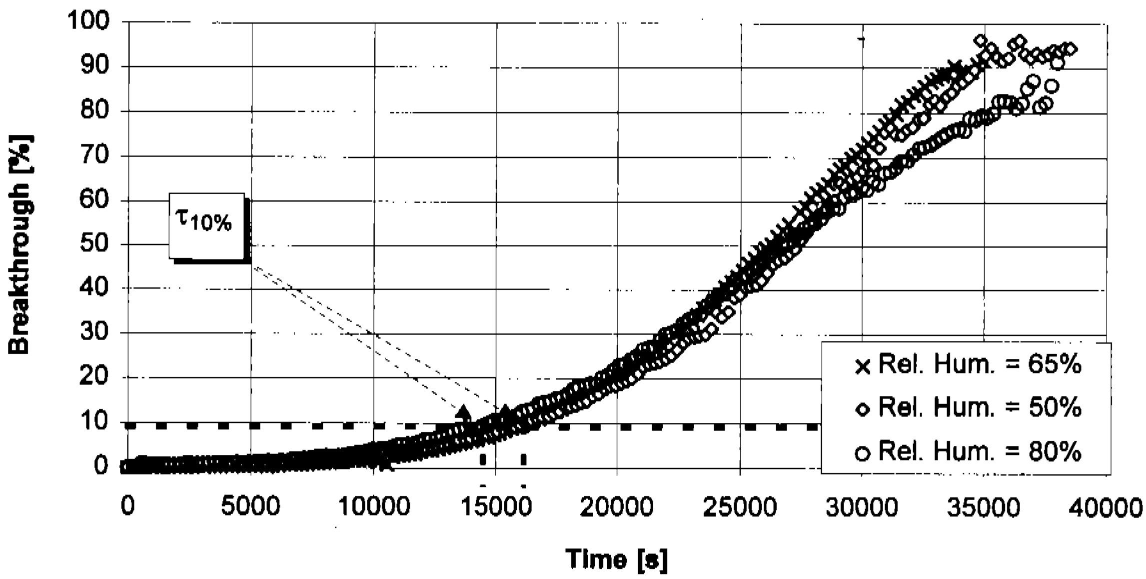

Figure 5 - Breakthrough curve - T = 23°C, L = 6.3 cm - Influence of relative humidity.
($\tau_{10\%}$ = breakthrough time corresponding to 10% of penetration)

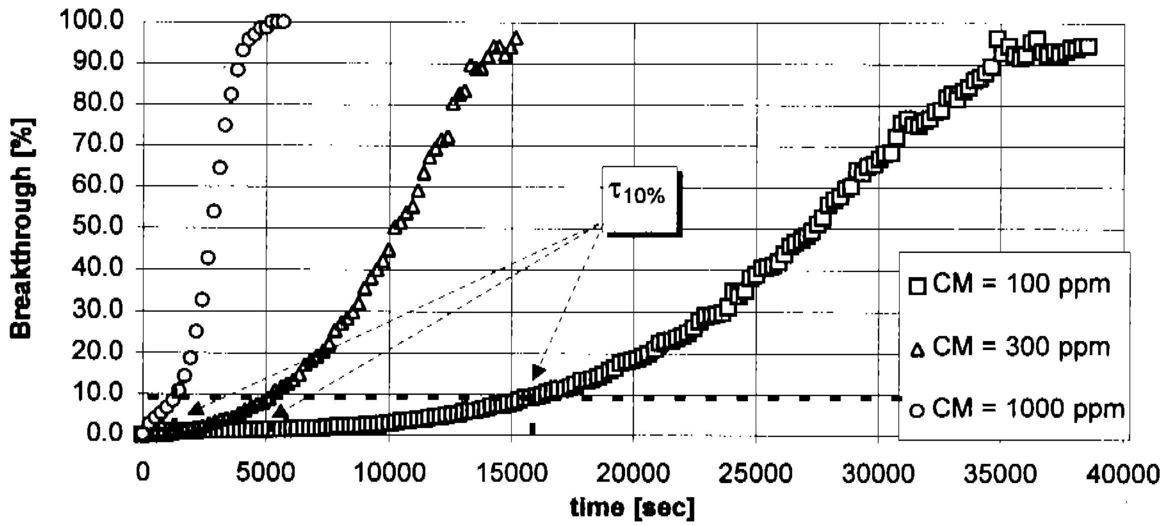

Figure 6 - Breakthrough curve - T = 23°C, L = 6.3 cm , φ = 50% - Influence of upstream test concentration.

in mind that these last tests have been performed on "pre-conditioned" filtering media and hence the water vapour has already been absorbed by the filter and is at the equilibrium during the measuring period.

For what concerns the breakthrough time, a weak point is represented by choice of the allowable downstream concentration. The data available in fact are based on very different conventions, that is, t_f is determined in each case in correspondence to c_V/c_M = 0.005 or 0.01 or 0.1. In order to reduce this confusion, which certainly does not help the operator, it is necessary to establish the time duration over which it is possible to calculate the concentrations to be matched with the acceptable threshold values [8]. Figure 7 and 8 show breakthrough times related to a C_V value equal to 1%, 5%, 10% and 25% of the C_M value.

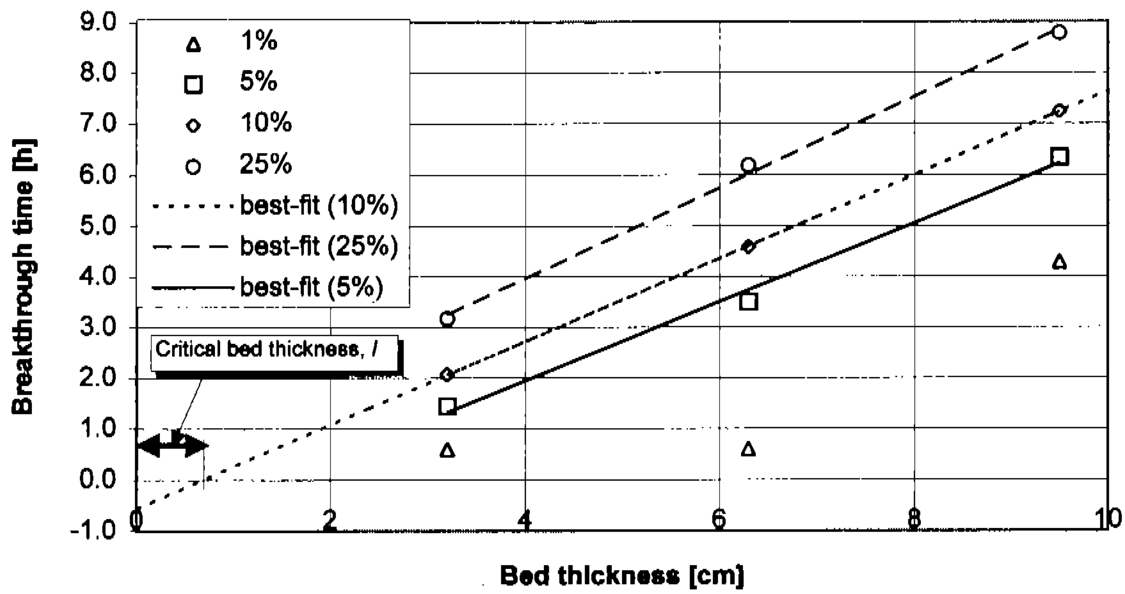

Figure 7 - Breakthrough time vs. bed thickness, L - test conditions T = 23°C, φ = 50%

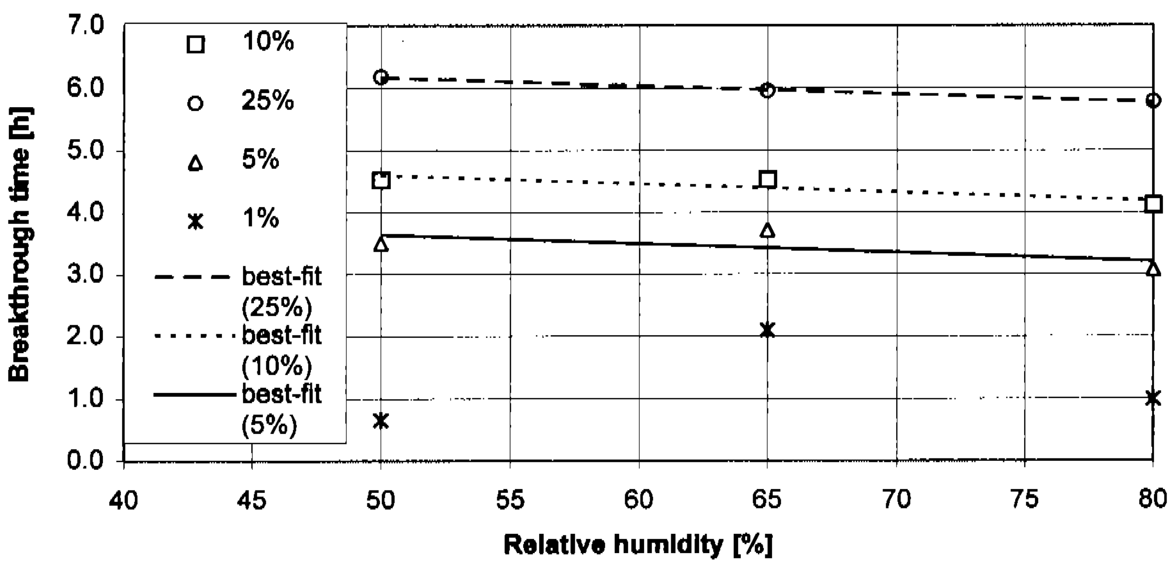

Figure 8 - Breakthrough time vs. rel. hum., φ - bed thickness L = 6.3 cm, T = 23°C

In particular figure 7 points out the profile of τ vs. the bed thickness, L, while curves of figure 8 show the same parameter vs. the relative humidity. Data related to $\tau_{1\%}$ seem to have poor reliability for both cases. This is, probably, due to the influence of the environmental background concentration that, in this case, has the same magnitude of C_V value. For other values of τ, instead, a striking linear trend is found ($R^2 > 0.9$) in the case of figure 7. This is in good agreement with theoretical models [5]. It is possible to see that the breakthrough time strongly increases with L. Furthermore, from the knowledge of the slope of the best-fit curve is possible to determine the filtering media dynamic adsorption capacity N_0. In the case of

$\tau_{10\%}$ it follows $N_0 \approx 8 \cdot 10^{-2}$ g of toluene per cm^3 of activated coal. This means 0.16 g of toluene per g of coal (the experimental value, obtained by means of weigh measurements, was 0.11 g of toluene per g of coal). Relative humidity, instead, (fig. 8) shows, once more, to have a weaker influence on the breakthrough time (at least in the field of value lower than 80%). In these case, moreover, the results are spread around the linear trend (R^2 varies from 0.43 up to 0.99), in particular for what concern the lower values of penetration (1%, 5%).
In the end, the breakthroguh time (10%) determined from figure 6 shows the filter performances as function of the upstream concentration. A comparison between experimental results of fig. 6 and the breakthrough times assessed by means of the Nelson and Correia model [4] has pointed out a sistematic over-extimation of the theoretical model (between 30% to 60%)

5. Conclusions

At the moment there is a growing commercial interest in gas filters used inside HVAC systems. In order to allow a comparison between different products it would be useful to have a standardised test method and above all a calculation method for the life time which could realistically take the situation in Europe into account. In particular, the choice of the downstream concentration (C_V) to be used in order to assess the filter life is still a critical issue. In fact, if the concentrations are instantaneous it is necessary to replace the filter as soon as the limit condition is reached and the life time could thus be too short. If the concentrations are calculated over the average life times assumed for commercial products (usually 1 year) there is a relatively long period of time in which the instantaneous concentration goes over the limit value even though the average still has not reached this value. This point is here considered to be very important in order to allow a widespread and effective use of gas filters.
In order to develop the independent knowledge inside Italy it would be useful to build a test rig to characterise materials, which could allow one to obtain experimental results in well defined conditions of temperature and humidity of the air which contains a low concentration of toluene. From the research point of view many efforts have already been made but there is some space left for the difficult subjects of mass transport, equilibrium conditions at low concentrations and motion resistance inside granular beds. In the end, experimental results obtained so far seem to be satisfactory and, even if some improvements of the test rig are still necessary, the measurement procedure allows fast and reliable assesment of filtering media performances.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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BUILDING PERFORMANCE EVALUATION FOR INDOOR AIR QUALITY USING OCCUPANT CONTAMINANT INHALATION AND ATTRIBUTION TO CONTAMINANT SOURCES

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Building Performance Evaluation for Indoor Air Quality using Occupant Contaminant Inhalation and Attribution to Contaminant Sources

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SYNOPSIS

The emissions of building materials like volatile organic compounds and indoor airborne contaminants such as environmental tobacco smoke expose occupants to hazardous substances. Although impacts of indoor air quality problems on human health, comfort, and productivity are quite large, no adequate evaluation methodology exists to assess contaminant source control techniques and building equipment systems. Even if instant indoor concentrations of many contaminants are not always high, continuous exposures to these contaminants may cause severe problems such as manifested by the sick building syndrome.

This paper proposes a method for evaluating long-term building performance in terms of indoor air quality. The approach applies exposure assessment but focuses on building performance. It employs the concept of using the total amount of substance inhaled by persons who occupy the room. This indicator is expressed by kilograms of each contaminant inhaled by persons ever present in the building during its operational life. The values include the effects of occupant rates. Concrete procedures for deriving variations of the indicators for both gaseous and particulate contaminants are described in detail. Another concept of contribution rates of contaminant sources is introduced both for instant values and on the inhalation basis. Evaluation examples of these indicators for a simple office geometry are shown for particulate matter, carbon dioxide, and formaldehyde. The results of the case studies strongly suggest the importance of indoor material selection and ventilation strategies. The contribution rate of contaminant sources makes it easier to plan a remedy for bad indoor air quality. The applicability of these indicators and future research requirements are also discussed.

LIST OF SYMBOLS

IAQ	: Indoor Air Quality
VOC	: Volatile Organic Compound
TVOC	: Total Volatile Organic Compounds
HCHO	: Formaldehyde
SBS	: Sick Building Syndrome
OCI	: Occupant airborne Contaminant Inhalation in specific spaces in absolute values
HVAC	: Heating, Ventilation, and Air-Conditioning
OCIOC	: Occupant airborne Contaminant Inhalation relative to Outside Concentration
OCIT	: Occupant airborne Contaminant Inhalation above Thresholds
SVE	: Scale for Ventilation Efficiency
CRI	: Contribution Rate for Indoor climate
CRCS	: Contribution Rate of Contaminant Sources
CFD	: Computational Fluid Dynamics
RC	: Reinforced Concrete
LCA	: Life Cycle Assessment

1. INTRODUCTION

The emissions of building materials like volatile organic compounds (VOCs) and indoor airborne contaminant such as environmental tobacco smoke expose occupants to hazardous substances. Although impacts of indoor air quality (IAQ) problems on human health, comfort, and productivity are quite large, no adequate evaluation methodology exists to assess contaminant source control techniques and building equipment systems. The potential health risk and possible injury to persons due to poor building quality must be quantitatively evaluated.

There are many indoor airborne contaminants that have impacts on human health, comfort, and productivity. A European research group has chosen 50 chemical substances as harmful inside the room [1]. Contaminants not only generated by occupants but also emitted from furniture, building materials, mechanical ventilation systems, or ducts may have strong adverse effects [2, 3]. Examples of the latter are formaldehyde (HCHO) or VOCs from building materials or furniture. Recent requirements for tighter buildings seeking energy conservation have unveiled the importance of these new IAQ problems [4]. The Healthy Residence Research Committee in Japan has recently designated 6 kinds of VOCs as harmful substances that should be urgently tackled; HCHO, toluene, xylene, wood preservatives, plastic softeners, and insecticides against white ants [5]. These tendencies support the importance of building performance evaluation in terms of IAQ.

Some airborne contaminants lead to injury in a very short time, while others have log-term adverse impacts on occupants. Even if instant indoor concentrations of many above-mentioned contaminants are not always high, continuous exposure to these contaminants may cause severe problems such as manifested by the sick building syndrome (SBS) [6]. Many health effects are not related to single exposures triggering an acute response, but are chronic, and induced either by bioaccumulation of a toxicant reaching a critical level in the target organ or tissue, or by repeated exposure causing acute episodes that ultimately lead to a chronic response [7]. Most reported TVOC concentrations in non-industrial environments are, for example, below 1 mg/m^3 and few exceed 25 mg/m^3, and at these concentration levels only sensory effects are likely to occur, but other health effects can not be excluded after long-term exposure [8]. Such investigations suggest the needs for long-term assessment indicators of IAQ.

In physiological, epidemiological, and medical science fields, peak concentration assessment is used for contaminants with acute effects, but exposure assessment is the state-of-the-art method for assessing human health risk in case of contaminants with chronic effects [9]. There are also some attempts to apply this methodology to building products. A European collaborative action for IAQ and its impact on man introduces exposure assessment after 24 hours, 3 days, and 28 days of test chamber experiments in order to evaluate adverse effects of VOC emission from solid flooring materials on occupants [7]. Taking into account this situation and the fact that more than 90% of time is spent inside buildings by people in industrial countries [10], it would be reasonable to use exposure assessment in terms of IAQ from a point of view of building performance.

The objective of this paper is to propose an approach for evaluating long-term impacts of indoor airborne contaminants on occupants, focusing mainly on performances of a particular building. This approach employs a concept of using the total amount of airborne contaminant inhaled by persons who occupy the room. Concrete procedures for deriving variations of the indicators for both gaseous and particulate contaminants are described in detail. Another concept of contribution rates of contaminant sources is introduced both for instant values and on the inhalation basis. Evaluation examples of these indicators for a simple office geometry are shown for particulate matter, carbon dioxide, and HCHO. The applicability of these indicators and future research requirements are also discussed.

2. OCCUPANT AIRBORNE CONTAMINANT INHALATION

A proposed indicator, *Occupant airborne Contaminant Inhalation in specific spaces (OCI),* is expressed by kilograms of each contaminant inhaled by persons ever present in the building during its operational life. Allergies and eye, skin, and mucous irritations are not separately accounted for at this stage. The model does not include psychological injury, perception of comfort, nor noise, illumination, and visual stress. The following are the procedures for deriving OCI *for gaseous contaminants.* The procedures *for particulate matter* are almost the same as for gaseous contaminants and are described in *APPENDIX 1.*

Occupant Airborne Contaminant Inhalation in Absolute Values (OCI)

When the perfect diffusion of the contaminant can be assumed, the concentration of each contaminant inside the room is estimated by the following differential equation:

$$V \cdot dC = W \cdot dt - Q \cdot (C - C_0) \cdot dt \qquad (1)$$

where

V : Volume of the room [m^3]
W : Contaminant generation or sink [m^3 / h]
t : Time [h]
C_0 : Outside concentration (volumetric) [-]
C : Concentration inside the room (volumetric) [-]
Q : Exchanged outside air volume [m^3 / h]

Outside air volume Q is attributed to mechanical ventilation, natural ventilation, and infiltration through cracks. The contaminant generation rate W can be written as:

$$W = W_h + \sum_i W_w(i) + W_f + W_{eq} + W_o \qquad (2)$$

where

W_h : Contaminant generation from occupants [m^3 / h]
$W_w(i)$: Contaminant generation or sink from wall surfaces made of the material i [m^3 / h]
W_f : Contaminant generation or sink from furniture and electrical equipment [m^3 / h]
W_{eq} : Contaminant generation or sink from HVAC and combustion systems [m^3 / h]
W_o : Contaminant generation or sink from other sources [m^3 / h]

Here, contaminant generation is expressed by positive values, while contaminant sink is shown by negative values. There are many kinds of contaminant generation and sink. They are caused by building materials, poor maintenance, and inappropriate operation of the building. Examples for source mechanisms are emission or release of mass, evaporation, desorption, resuspension, growth of micro-organisms, and radio activity, and the most important sink mechanisms are deposition (absorption and adsorption), sedimentation, plate out, filtering, condensation, chemical reaction, and radio active decay [11]. W_{eq} in the equation (2) includes all the effects concerning Heating, Ventilating, and Air-Conditioning (HVAC) systems such as ducts, fans, heat exchangers, filters, and humidifiers, and other processes such as combustion. The last term W_o includes the other sources such as contaminants generated by smoking or microbiological organisms.

Here, we will try to propose an approach for evaluating long-term building performance in terms of IAQ. The approach is the concept using the total amount of contaminant inhaled by persons who occupy the room. In this paper, we name it *Occupant airborne Contaminant Inhalation in specific spaces in absolute values (OCI).* The OCI value is derived by the following procedures for each contaminant. The instant amount of the inhaled contaminant is expressed as:

$$M_{ci} = k \cdot f_{res} \cdot V_{br} \cdot C \qquad (3)$$

where

M_{ci} : Instant amount of contaminant inhaled by occupants [g / (h·person)]
k : Coefficient for providing the amount of each contaminant per unit volume [g / m^3]
f_{res} : Respiratory frequency [1 / h]
V_{br} : Breathing air volume per one respiration per person [m^3 / person]

Here, M_{ci} is an instant value and a function of time. f_{res} and V_{br} are also functions of time because they depend on the level of human activities [12]. The coefficient k is an order of density and can be drawn from the gas law if the assumption of the perfect gas is appropriate *(APPENDIX 2)*.

If the equation (3) is multiplied by a weighting factor, w_{oc}, representing occupancy in the room and integrated in terms of time, we can obtain an absolute OCI value during the period concerned (hour, day, week, month, year, or even building lifetime). Here, w_{oc} is a function of time.

$$OCI = \int M_{ci} \cdot w_{oc} \cdot dt \qquad (4)$$

where

OCI : Total amount of contaminant inhaled by occupants during the period concerned [g] (Occupant Contaminant Inhalation in absolute values)
w_{oc} : Weighting factor for occupancy [person]

OCI means the total amount of contaminant inhaled by occupants in a specific indoor space, including the effects of occupant rate of the room. This means that OCI equals zero if the room is always vacant and the occupant rate is zero, and that the indicators of two rooms should be equal if one person lives in a larger room but gets the same dose as another person living in a smaller room. OCI can be thought to be a measure of the total hazards caused by IAQ problems in the building. OCI has a dimension of the number of occupants in the room times potential dose, which is normally used in physiology [9].

If we divide the resultant value by the number of regular occupancy, a normalized value for contaminants inhaled by occupants can be obtained. The obtained value means an average amount of contaminant inhaled by one person during the time concerned. It is similar to the above-mentioned potential dose.

$$OCI_{norm} = OCI / N_{ro} \qquad (5)$$

where

OCI_{norm} : Normalized OCI value during the period concerned [g / person]
N_{ro} : Number of nominal occupancy [person]

If we focus only on the average inhaled values during the occupant time, the average amount of contaminant per one hour and one person inhaled only by the people who actually occupy the room during the period concerned can be used. Substantial effects only during the occupant period are considered by this value.

$$OCI_{ave} = OCI / \int w_{oc} \cdot dt \qquad (6)$$

where

OCI_{ave} : Average amount of OCI per hour and person related only to the people who actually occupy the room during the summed occupied period [g /(hour·person)]

This value is useful with relatively slight variations in the contaminant concentrations. When there is a sharp peak in contaminant concentration or when only an acute adverse effect on human health is expected, peak values for contaminant concentrations should be used.

Occupant Airborne Contaminant Inhalation relative to Outside Concentration (OCIOC)

When the contaminant exists in nature and the indoor contaminant concentration is always higher than the outside concentration, another concept of *Occupant airborne Contaminant Inhalation relative to Outside Concentration (OCIOC)* can be more effective. The following equation can be used in place of the equation (3):

$$M_{ci} = k \cdot f_{res} \cdot V_{br} \cdot (C - C_0) \qquad (7)$$

In this equation, we subtract the outside concentration rate C_0 from the indoor concentration rate C in order to neglect the effects of contaminants contained in the outside air. This value makes it possible to concentrate only on the impacts of contaminants generated inside the building. Various values for inhaled substance such as OCIOC, $OCIOC_{norm}$, $OCIOC_{ave}$ can be calculated by the same procedures described in the previous section (equations (4) to (6)). Note again that M_{ci} in this equation should be basically positive.

The OCIOC values are useful when outside air pollution is unavoidable but indoor contaminant generation is the central problem that should be solved. These values may represent building performances in terms of IAQ.

Occupant Airborne Contaminant Inhalation above Thresholds (OCIT)

For some substances, there are thresholds below which contaminant inhalation is not perceived by occupants or does not cause any substantial problems. Bioeffluent and CO_2 concentration under 1,000 ppm would be such examples.

In such a case, we can introduce a different version of *Occupant airborne Contaminant Inhalation above Thresholds (OCIT)*. It is interpreted as the amount of contaminant that exceeds the threshold value and is inhaled by occupants. This concept can be useful when adverse effects of the substance on occupants are negligible at smaller values below the thresholds. This concept makes it possible to quantitatively evaluate the occupants' exposure level over contaminant thresholds during the time concerned.

In order to calculate this value, the equation (3) should be replaced by the following equations:

$$
\begin{aligned}
M_{ci} &= k \cdot f_{res} \cdot V_{br} \cdot (C - C_{th}) && \text{when } C > C_{th} \\
M_{ci} &= 0 && \text{when } C \le C_{th} \qquad (8)
\end{aligned}
$$

where

C_{th} : Threshold value for the gaseous contaminant [-]

Various values for inhaled substance such as OCIT, $OCIT_{norm}$, $OCIT_{ave}$ can be calculated by the same procedures described in the previous sections (equations (4) to (6)). We should note that these values refer not to perceived stress levels but to exposure levels of concentrations that exceed thresholds. Perceived stress is a very strong nonlinear system and is affected also by factors other than contaminant concentrations such as social factors, noise, the thermal and illumination environments, etc. Furthermore, there are many contaminants whose thresholds have not been established in non-industrial indoor air [8].

3. CONTRIBUTION RATE OF CONTAMINANT SOURCES (CRCS)

In the previous sections, various concepts of Occupant Contaminant Inhalation (OCI) are introduced in order to evaluate IAQ conditions using exposure assessment. Here, we introduce another concept of *Contribution Rate of Contaminant Sources (CRCS).*

The concept of CRCS is based on the idea of assessing the contribution of individual contaminant sources to indoor contaminant concentrations and OCI values. This idea makes use of the linearity of the governing equations for indoor airborne contaminant transportation, although this assumption is not valid for active contaminants. According to these useful characteristics, we can understand quantitatively the contributions of particular contaminant sources to IAQ deterioration. Therefore CRCS has a potential to make it easier to plan efficient source control methods. CRCS can also be used to evaluate the influences of non-uniform concentration distributions.

The same kinds of approaches as described here have been introduced for scale-model tracer gas tests of a large sports arena as contribution rates of each duct line to the air conditioning of each zone [13]. These values were used to design the optimal control system for room air temperatures. Making efficient use of Computational Fluid Dynamics (CFD), practical indicators were introduced for the evaluation of ventilation efficiency inside the room

(SVE: Scale for Ventilation Efficiency) [14] and the mechanisms of indoor thermal environment (CRI: Contribution Rate for Indoor climate) [15].

The following are procedures for deriving CRCS both for instant values and on the inhalation basis.

Contribution Rate of Contaminant Sources for Instant Values ($CRCS_{inst}$)

Contribution Rate of Contaminant Sources for instant values ($CRCS_{inst}$) for gaseous contaminants can be calculated by the following equation:

$$CRCS_{inst}(j) = C_p(j) / C \qquad (9)$$

where

$CRCS_{inst}(j)$: Contribution rate of contaminant source j for instant values [-]
$C_p(j)$: Partial concentration due to contaminant source j [-]

$C_p(j)$ is calculated by steady-state equations for a given contaminant source. According to the linearity of the governing equation for concentrations, the following conditions are automatically satisfied.

$$\sum_j C_p(j) = C \quad \text{and hence} \quad \sum_j CRCS_{inst}(j) = 1.0 \qquad (10)$$

Instant contribution rates of contaminant sources can be calculated not only with assumptions of perfect mixing but also when non-uniform concentration distributions are taken into account by CFD. The outside concentration can also be treated as one of the contaminant sources. $CRCS_{inst}$ can be applied to contaminant sink such as deposition or absorption by incorporating negative values. $CRCS_{inst}$ is useful to plan a remedy against *acute* effects from contaminant sources. Deriving procedures for particulate matter are similar.

Contribution Rate of Contaminant Sources on the Inhalation Basis ($CRCS_{ib}$)

Contribution Rate of Contaminant Sources on the inhalation basis ($CRCS_{ib}$) for gaseous contaminants can be calculated by the following equations:

$$CRCS_{ib}(j) = \int M_{cip}(j) \cdot w_{oc} \cdot dt / OCI \qquad (11)$$

$$M_{cip}(j) = k \cdot f_{res} \cdot V_{br} \cdot C_p(j) \qquad (12)$$

where

$CRCS_{ib}(j)$: Contribution rate of contaminant source j on the inhalation basis [-]
$M_{cip}(j)$: Amount of substance emitted by contaminant source j and inhaled by occupants [g / (h·person)] (for particulate matter, see *APPENDIX 1*)

When perfect mixing in the space is not assumed or there is time dependency of material emissions, $CRCS_{ib}(j)$ values are different from $CRCS_{inst}(j)$. The total value for $CRCS_{ib}$ is 1.0 as in the equation (10). $CRCS_{ib}$ is useful to plan a remedy against *chronic* effects from contaminant sources.

4. CASE STUDIES

Some case studies are conducted for a simple office geometry focusing on inhaled contaminants (OCI) under the conditions summarized in Table 1. Particulate matter, carbon dioxide (CO_2), and HCHO are selected as indoor airborne contaminants for OCI evaluations. HCHO is also used for CRCS evaluations. More complicated examples are left for the future.

Occupant Contaminant Inhalation relative to Outside Concentration (OCIOC)

Figure 1 shows estimated inhaled particulate matter relative to the outside concentration per one year and one person ($OCIOC_{norm}$) as a function of an air change rate and an average occupant rate. The average occupant rate is an averaged value between 8:00 and 18:00 on week days excluding the time between 12:00 and 13:00 (Table 1). Here, it is assumed that an instant occupant rate is always 0 or 100% (0 or 2 occupants) even if an average occupant rate over time is neither 0 nor 100%. This condition is selected in order to evaluate the largest OCIOC values for each average occupant rate. The inhaled particulate matter per person becomes smaller as the occupant rate becomes smaller because of the shorter occupancy time. The $OCIOC_{norm}$ values are proportional to the average occupant rate, because contaminant concentration is the same during the occupancy period. The $OCIOC_{norm}$ values are inversely-proportional to the air change rate and decrease as the air change rate increases. The $OCIOC_{norm}$ value is about 135 mg/(year·person) when the air change rate is a normal value of 2 h^{-1} and the average occupant rate is 100%. The data for particulate diameters are required to evaluate the particulate matter absorbed by the human body, because the absorption rate is dependent on the diameters.

Table 1 Calculation conditions.

- Room size: 4 m by 6 m by 3 m [H]. Room air temperature: 24 °C.
- Occupancy time: 8:00 to 18:00 including a break from 12:00 to 13:00. 2 occupants.
- Saturday and Sunday are holidays, but it is assumed that there are no national holidays.
- Generation rate of particular matter: 8.45 mg/(h·person) [16]. A value for total particular matter (TPM) is used. A pulmonary ventilation volume is dependent on the metabolic heat production rate and is estimated to be 0.49 m^3/(h·person) at desk work according to [17].
- Generation of CO_2: 200 ml/(min·person) [18].
- Generation of HCHO is assumed to be constant as follows [19].
  50 books: 1.1 μg/(h·book), 2 pairs of shoes: 1.6 μg/(h·pair), Ceiling material: 0.3 μg/(100cm^2·h)
  Office carpet: 0.2 μg/(100cm^2·h), Reinforced concrete : Assumed to be 0, Particle boards: 8.3 μg/(100cm^2·h)
  Plywood: 18.0 μg/(100cm^2·h)

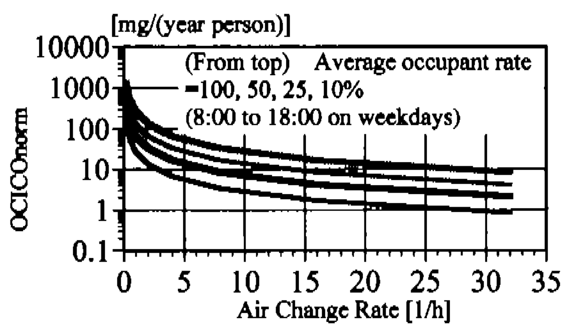

Figure 1 Estimated $OCIOC_{norm}$ values for particulate matter.

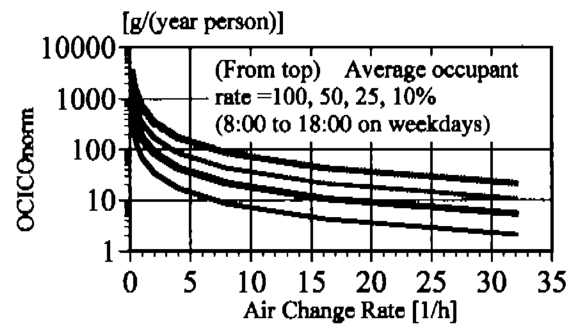

Figure 2 Estimated $OCIOC_{norm}$ values for CO_2.

Figure 2 shows estimated $OCIOC_{norm}$ values for CO_2 per one year. It depends on the air change rate and the average occupant rate in the same manner as the particulate matter. The $OCIOC_{norm}$ value for CO_2 is about 345 g/(year·person) with an air change rate of 2 h^{-1} and an occupant rate of 100%. Although the inhalation of CO_2 is not directly hazardous to human health unless it exceeds a certain level (for example 5000 ppm), this value can be an IAQ indicator for a certain period including the effect of occupant rates.

Figure 3 illustrates estimated $OCIOC_{norm}$ values for HCHO per one year. In this figure, the influences of the selection of wall materials are investigated. Three materials for side walls, reinforced concrete (RC), particle boards, and plywood are compared. The occupant rate can be interpreted also as a constant value during the whole office hours in this study. Note that the HCHO emission rate is assumed to be constant during the whole year, which is not precise because of the neglect of the emission rate reduction as time proceeds. In case of RC, the $OCIOC_{norm}$ values for HCHO are small, for example 10 mg/(year·person) with an air change rate of 2 h^{-1} and an occupant rate of 100%. This is due to the assumption of no emission on the RC walls (Table 1). However, the $OCIOC_{norm}$ values for HCHO are very high in case of particle boards with a value of about 400 mg/(year·person) in the same condition. The values for plywood are higher than that, for example about 870 mg/(year·person) under the same condition. This is almost 90 times larger than the value with RC. This result strongly suggests the importance of the indoor material selection process.

Contribution Rate for Contaminant Sources (CRCS)

Figure 4 shows CRCS values of each contaminant source with different area ratios of particle boards against RC for side walls. The contaminant is HCHO. The influence of the outside concentration is neglected in this study. In this evaluation, CRCS values for instant values and on the inhalation basis are the same because perfect mixing in the air and constant contaminant sources are assumed in the room. One can see high CRCS values from side walls when the area ratio of particle boards is high. The CRCS value from side walls becomes comparable with the other sources only when more than 31/32 or 97% of the side walls made of particle boards are replaced by RC walls. This means that a drastic change of side wall material into one with less HCHO emission can be an effective remedy against high HCHO concentration but that only a small replacement does not work very well. The CRCS value in side walls becomes zero if RC walls are fully installed. As described here, CRCS can be used to evaluate the effects of source control methods.

5. DISCUSSION

Some possible indicators for building performances in terms of IAQ, Occupant Contaminant Inhalation (OCI) and Contribution Rate of Contaminant Sources (CRCS), were introduced, and simple case studies using them were described. There are many things that should be done in the future. The following are discussions on the advantages and limitations of these indicators and required future research.

Advantages

1. The concept of OCI makes it possible to assess long-term effects of indoor contaminants on occupants. This characteristic is suitable for the conditions where the instant contaminant concentration is not high, but continuous contaminant inhalation is a main cause of chronic adverse effects. This is also effective when indoor airborne contaminant concentration is strongly time-dependent, e.g., VOC emissions from wall materials.
2. OCI has a potential to be practically used when more knowledge becomes available

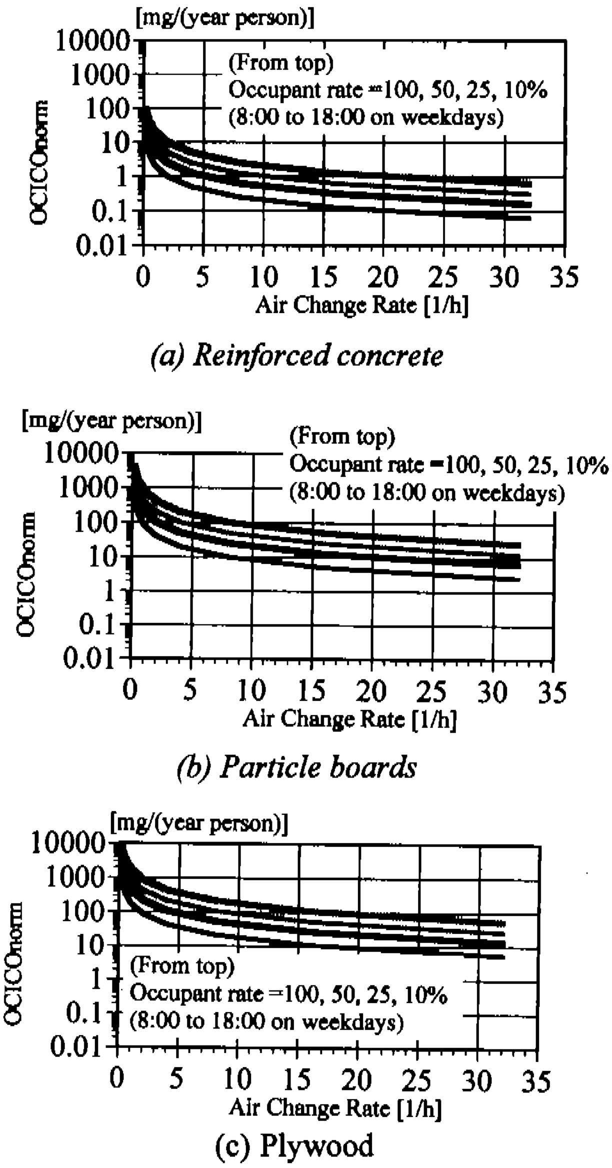

Figure 3 $OCIOC_{norm}$ values for HCHO with side walls made of various materials.

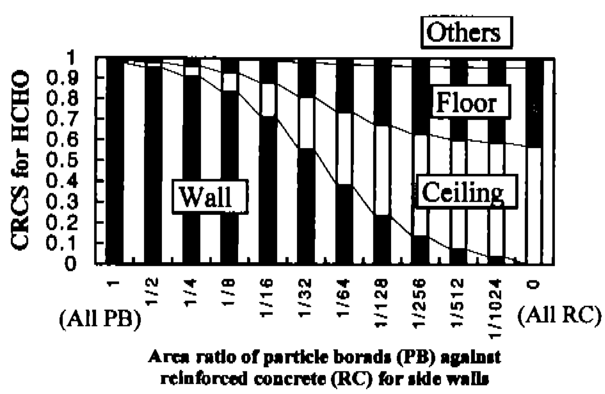

Figure 4 CRCS values of each contaminant source with different area ratios of particle boards against reinforced concrete for side walls. (Effects of outside concentration are neglected.)

on dose-response relationship investigations in physiological and medical fields.
3. OCI includes the effects of occupant rates. This means that OCI values show building performances on IAQ from the occupants' point of view and with respect to health impact.
4. Life Cycle Assessment (LCA) is useful for the evaluation of building sustainability in terms of energy consumption, CO_2 emissions, or acidification factors [20]. As IAQ is also an important factor together with the aforementioned [21], it is reasonable to employ long-term assessment indicators for IAQ such as OCI.
5. CRCS values help us analyze the mechanisms of contaminant transportation inside the room. They also make it easier to plan an efficient source control remedy for IAQ problems. Not only instant but also long-term evaluations become possible by combining the concepts of CRCS with those of OCI ($CRCS_{ib}$).

Limitations

1. Methods for evaluating indoor contaminant concentrations have not been fully established. There still remain many phenomena that cannot be modeled precisely such as emission processes from building materials and their interactions with thermal conditions.
2. It is not today possible to conclude that sensory irritation is associated with the sum of mass concentration of contaminants such as VOCs at the low exposure levels typically encountered in non-industrial indoor air.
3. Exposure-response relationships in IAQ problems have not been established yet. Further research in hygiene and physiology fields is essential to overcome this difficult question.
4. OCI values do not evaluate short-term effects on human health. Peak concentration assessment is necessary for acute effects of indoor airborne contaminants. It should be clarified for which contaminants the long-term assessment is efficient.
5. OCI assumes that the contaminants act only through respiration. This assumption may not be reasonable for the other symptoms such as eye irritations or skin dryness.
6. It is sometimes difficult to predict occupant rates appropriately during the design stage. However, this is unavoidable when one evaluates spaces from occupants' point of view.
7. It is difficult to validate OCI values by field measurements. One should use values for contaminant concentrations for this purpose.
8. CRCS proposed in this paper can be applied only to passive contaminants.

Further Research Requirement

Considering the above factors, it is suitable to evaluate IAQ by the long-term assessment or indoor contaminant concentrations, depending on the contaminant. There are still strong demands for further research concerning the following points.

1. Emission processes of indoor contaminants from building materials are affected by such factors as temperatures, air velocities, and contaminant concentrations. Emission mechanisms from building materials should be included in environmental evaluation models.
2. Indoor air quality has strong relevance to or sometimes trade-offs with other factors such as energy consumption, thermal comfort, and the light environment. Therefore, a prediction model that can simultaneously treat various factors is desired to realize sustainable buildings. The model should be able to evaluate effects of natural ventilation or thermal storage capacity of buildings, taking unsteady-state phenomena into account.
3. In this paper, indoor airborne contaminant concentration has been assumed to be uniform in a room. When this uniformity is not appropriate, for example in case of displacement ventilation, other sophisticated methods such as CFD or chamber tests should be made to estimate indoor contaminant concentrations properly [10]. However, the concepts described in this paper are also applicable to such cases. Values for ventilation efficiency obtained by CFD or chamber tests can be effectively used in the unsteady-state simulation described above to make more reasonable and precise evaluations possible.
4. Inhalation must be placed in the context of total exposure assessment, which requires consideration of all pertinent environmental media and all routes of entry into the body.

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APPENDIX 1 Occupant Contaminant Inhalation (OCI) in Case of Particulate Matter

When the contaminant concerned is particulate matter, a procedure for estimating inhaled substance is a little different from the case of gaseous contaminants. If the perfect diffusion of the contaminant can be assumed, the contaminant concentration inside the room is expressed by the following differential equation:

$$V \cdot dC' = W' \cdot dt - Q \cdot (C' - C'_0) \cdot dt \qquad \text{(A1-1)}$$

where the superscript ' means values not for gaseous contaminants but for particulate matter.

W' : Contaminant generation [g / h]
C'_0 : Outside concentration rate [g / m^3]
C' : Concentration rate inside the room [g / m^3]

The instant substance inhaled by occupants is expressed by the next equations for the equations (3), (7), (8), and (12) respectively:

$$M_{ci} = f_{res} \cdot V_{br} \cdot C' \qquad \text{(A1-2)}$$
$$M_{ci} = f_{res} \cdot V_{br} \cdot (C' - C'_0) \qquad \text{(A1-3)}$$
$$M_{ci} = f_{res} \cdot V_{br} \cdot (C' - C'_{th}) \text{ when } C' > C'_{th} \text{ and} = 0 \text{ when } C' \leq C'_{th} \qquad \text{(A1-4)}$$
$$M_{ci}(j) = f_{res} \cdot V_{br} \cdot C'_p(j) \qquad \text{(A1-5)}$$

where

C'_{th} : Threshold value for particulate matter [g / m^3]
$C'_p(j)$: Partial concentration for a contaminant source j for particulate matter [g / m^3]

Note that there is no coefficient k in the equations (A1-2) to (A1-5) as in the equation (3). Various parameters for OCI are calculated in the same way as shown in the previous explanation for gaseous contaminants.

The above-mentioned procedure can also be used to derive values for gaseous contaminants and compound mixtures usually expressed by mass concentrations, e.g., TVOC [8].

APPENDIX 2 Calculation of Coefficient k in Equation (3)

The coefficient k in the equation (3) for each gaseous contaminant can be given by the following equation, if the contaminant can be assumed to be a perfect gas:

$$k = (M \cdot P) / (R \cdot T) \qquad \text{(A2-1)}$$

where

M : Molecular weight of each gaseous contaminant [g / mol]
P : Indoor pressure [Pa]
R : Gas constant [J / (mol·K)] (= 8.3145)
T : Absolute indoor air temperature [K]

For a compound mixture, one should calculate a total k value, from the constitution of the mixture and the equation (A2-1).

VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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IMPROVEMENT OF INDOOR CLIMATE AND VENTILATION SYSTEM IN A RENOVATED MULTISTORIED RESIDENTIAL BUILDING

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Synopsis

The goal of this project was to improve the quality of indoor air in a multistoried residential building of 81 flats built in 1960. The building is located in a heavily built urban area of Helsinki. The building had a mechanical exhaust ventilation system without outdoor air inlets. A questionnaire was sent to occupants and a condition survey was made prior to renovation. The main indoor climate problem was draught with a prevalence of 60 %. Other almost as common problems were traffic noise also during nights and dust coming from the street. The ventilation system was fully unbalanced with reduced exhaust air flows partly due to uncleaned exhaust air vents. A new type of fresh air window with air filtration (EU 5) and good acoustic performance was developed. The sound insulation value measured in field was 42 dB(A). When these new windows were installed in the dwellings a new questionnaire was sent to occupants. The results showed clear improvement in all indoor climate related factors. The habitants were much more satisfied with the performance of ventilation system after the renovation measures.

1.0 Introduction

The design values for kitchen and bathroom exhaust air flows were 80 and 60 m^3/h in the 1960's, nowadays 20 and 15 l/s /1/. In small flats lower values are allow, but at least 0.5 ach is required 24 hours per day. Almost all mechanical exhaust ventilation systems in block of flats are equipped with a two-speed fan. The full speed is normally used about 6 hours per day /2/. When outdoor temperature falls under -12°C (in Southern Finland) fans are used only in half speed. In Helsinki there are about 350 such hours annually. Investigations /2/ concerning mechanical exhaust ventilation in block of flats built in the 1960's and the 1970's showed that when all the windows are closed, only one fifth of the exhaust air flows from kitchen or bathroom met the requirements of the National Building Code of Finland /1/. When exhaust fans are used in half speed, only 30 per cent of flats had an air change rate 0.5 or more per hour. In an other Finnish study the air-exchange rate over a two-week period 0.5 ach were reached in 40 % of dwellings with mechanical exhaust ventilation system /3/. The outdoor air inlets came obligate in new residential buildings without mechanical supply in the late of 1980's. There are still approximately one million dwellings in blocks of flats without outdoor air inlets.

2.0 Building and its environment

2.1 Building

The building built in 1960 is located in a heavily built urban area of Helsinki, Figure 1. The building is in the corner of two street with busy traffic. It has seven floors altogether with 110 habitants in 81 flats, most of them with only one or two bedrooms. The building is a condominium which means that most of flats are owned by the habitants. The building had mechanical exhaust ventilation system without outdoor air inlets. The exhaust air openings were placed in kitchen and bathroom. The exhaust air ducts and fan chambers were made from concrete. The exhaust air fan chambers are in the eight floor which is used also a storage room of the habitants. All flats have central heating system based on district heating. Windows were double pane type in original conditions with no outdoor air inlets.

Figure 1. Building in summer 1998 after the new windows have been installed.

No larger renovation measures have been made since 1960. In the year 1996 the condominium made a decision to renovate windows and ventilation system. The renovation costs of the new windows and ventilation system had to be kept at a level of 60 ECU/m^2.

2.2 Outdoor air quality

The nearest air quality follow up station locates only 300 m from the building in a similar urban environment. The yearly variation of inhalable particles is shown in Table 1.

Table 1. The daily means of inhalable particle concentration in each month of the year 1996.

Month	Concentration $\mu g/m^3$	Month	Concentration $\mu g/m^3$	Month	Concentration $\mu g/m^3$
January	35	May	40	September	50
February	75	June	30	October	35
March	98	July	25	November	40
April	90	August	50	December	40

The high particle concentrations are caused by accumulated sand spread on streets in order prevent slipperiness during the winter. The main outdoor air pollutants levels are; SO_2 negligible, NO_2 150 $\mu g/m^3$ (winter), 60 $\mu g/m^3$ otherwise, CO less than 5 mg/m^3.

3.0 Definition of the problem

3.1 Questionnaire

A self-administrative questionnaire was distributed to the occupants prior to renovation in January 1997. In the questionnaire visible moisture and mold damaged, odors and their sources, airing habits, use of bathrooms for baths and laundry drying were asked. Also the prevalence of the most common indoor climate related problems and symptoms were asked. The response rate was 60 %. The main indoor climate problem was draught with a prevalence of 60 %. Other almost as common problems were traffic noise also during nights and dust coming from the street. There was urgent need for improving sound insulation and outdoor air filtration. Only 10 % of the occupants were satisfied with ventilation system and one third of the occupants were satisfied with the heating system. 38 per cent of the occupants said that their dwellings are comfortable. In 30 % of flats were reported problems with odors coming either outdoors, or other flats or staircases.

3.2 Condition survey

In the beginning of the project, a new Finnish guidelines for indoor climate investigations was conducted in the building before the main renovation work /4/, /5/. The guidelines consist of the measures and methods including measurement equipment needed in the investigations. The indoor climate investigations include the following steps:

- assessing of indoor climate problems by interviewing occupants and maintenance personel of the building
- examining of the performance, condition and balance of ventilation system and comparing them to the design specification
- examining of air flows, pressure differences and air tightness of the building
- examining of the performance, condition and balance of heating system

- examining of the condition of constructions, e.g. water damages and the effect of ventilation on them
- assessing of thermal conditions, humidity and pollutants
- measuring of indoor climate parameters (temperature, humidity, pollutants of indoor air, etc.)

Visible moisture damages were recorded with the help of the check list for mould problem investigations. The condition of old exhaust ducts were inspected with video camera recording. Altogether 58 exhaust ducts were inspected. The sound insulation of outer wall was measured.

Before the conditions survey, the cleanliness of exhaust air vents were checked in ten dwellings. 40 % of the exhaust vents were so dirty (crease and dust) that it was not possible to make any air flow measurements. The mean exhaust air flow of kitchen was 8.4 l/s and of bathroom 7 l/s. No kitchen hoods were found. The exhaust vents in the dwellings were cleaned before condition survey.

During the condition survey over 40% of the dwellings were found to have moisture damages, mostly in bathrooms due to low ventilation or pipe leakage. The exhaust fans were two-speed type. The full speed was in use between 8 and 9, 11 and 13 and 17-19 daily. The exhaust air flows were still after cleaning of exhaust air vents below the target values in many dwellings. The mean exhaust air flow of kitchen was 11 l/s and of bathroom 10 l/s with full fan speed. Those values are about 60 % of the building code level. The ventilation of the underground sauna department was in very poor conditions, the exhaust air flows were less than 5 l/s.

The video film of exhaust ducts showed several local damages, holes between ducts and tools left there during construction. The ducts were so clean that duct cleaning was not necessary.

The sound insulation values of the existing windows were 26-28 dB. The noise levels inside the dwellings were 39...42 dB(A). The Finnish guideline for the equivalent noise level in living and bedrooms is 35 dB(A) during daytime and 30 dB(A) during the night (5).

The following actions were recommended based on the condition survey:
- replacement of old exhaust air vents with adjustable air vents
- tightening of exhaust air chamber and improving of sound insulation
- modernisation of exhaust air fans
- adjustment of exhaust air flows
- fresh air inlets in staircases
- separate ventilation system for underground sauna department
- cleaning of outdoor air inlets of underground spaces
- adjustment of heating system after new windows installed and ventilation system adjustment
- installation of kitchen hoods where the exhaust air cleaned before recirculated back to the kitchen
- condition survey of water supply and sewer pipelines

4. 0 Window

The main design problem is to develop a solution for the outdoor air supply through window without causing any draught in the zone of occupancy. A new type of fresh air window with air filtration (EU 5) and good acoustic performance was developed. The acoustic performance of the new window and velocity and temperature field caused by outdoor air inlet were measured in the full scale test room built in the HVAC-laboratory.

Structure

The new window, installed in dwellings located in the street side of the building, consists of two parts, a three pane window and a separate airing shutter which can be opened in order to change the outdoor air filter or when airing is needed. The thickness of the glasses from outside to inside are 8 mm, 6 mm and 6 mm. The structure of the airing shutter from outside to inside is; 1.5 mm thick steel plate, 15 mm thick stone wool with covering against air, 25 mm air flow channel, 15 mm stone wool with covering against air, 15 mm + 15 mm gypsum board and surface plate.

The outdoor air is first cleaned by filter located at the bottom of the airing shutter. The filter is changed twice per year. The air flow from the bottom to the top side of the airing shutter via rectangular channel (25 thick and 200 mm width). The air is directed to indoor through a radial diffuser which spreads the air over the wall surface. The slot of the diffuser is partially covered in order to direct the cold supply air over the warm radiator under the window. The diffuser is widely used in Finnish dwellings as outdoor air inlet. The manufacturer notified that it can be supplied 8 l/s per vent and meet the requirements of the National Building Code of Finland /1/ concerning the maximum air velocities in the occupied zone.

In the dwellings facing the yard similar windows except for the airing shutter being a three pane window were installed. The outdoor air inlet was ordinary an slot-type adjustable vent with much lower sound insulation value. The slots were installed in the upper frame of the window. The slots also have filters inside, but with lower efficiency than filters in the street side.

Sound insulation

The sound insulation value was measured in the building after some windows were installed on the street side of the building. The sound insulation value was now 42 dB. The measurements were slightly disturbed by internal noise from other dwellings and plumbing systems. That means that even better sound insulation value could have been obtained if the temporary internal noise sources could have been eliminated.

5.0 Results and discussion

New windows were installed in the dwellings between February and April 1997. The exhaust air flows were measured with full fan speed after the windows were installed. The mean exhaust air flow from kitchen was 12 l/s and from bathroom 9 l/s and from cloakroom 12 l/s. Those values are about 60 % of the building code level except cloakroom. Air exchange rates were 1.0 ach in single room flats, 0.7...0.9 ach in double room flats and 0.7 ach in three room flats. The air exchange rate was 0.5 or greater almost in all flats.

A self-administrative questionnaire was distributed to the occupants after one year of the installation of new windows in April 1998. The questionnaire was shorter than in 1997, moisture and mold damaged were no more asked. The response rate was now 72 %. In Table 2 the prevalence of indoor air related problems before and after the renovation measures are shown.

The measurements in the dwellings before and after the renovation showed that the deviation of room temperatures between single dwellings decreased remarkably. Before renovation the mean value of the room temperature was 22.4°C (20-25°C) and after renovation 22.0°C (21.0-23.4°C). The percentage of complaints concerning main indoor related problems like traffic noise and draught decreased to a great extent. Also indoor air quality were judged now much better than earlier. Now 27 % of the occupants were very satisfied with ventilation system and 43 % of the occupants were very satisfied with the heating system. 50 per cent of the occupants said that their dwelling is comfortable. The percentage of dissatisfied with ventilation system decreased from 30 % to 12 %.

Table 2. The prevalence of indoor air related problems before and after renovation measures.

Problem	Prevalence in 1997 %	Prevalence in 1998 %
Too high room temperature	4.2	6.9
Too low room temperature	42.6	31.0
Unstable room temperature	34.0	12.1
Draught	61.7	22.4
Cold floors	36.1	25.9
Dry air	38.3	24.1
Humid air	4.3	3.4
Stale air	31.9	10.3
Unpleasant odor	23.4	13.8
Insufficient ventilation (winter)	34.0	13.8
Insufficient ventilation (summer)	42.6	24.1
Dusty air	38.3	22.4
Dust in surfaces	53.2	39.7
Noise from ventilation system	4.3	3.4
Other noise (traffic etc.)	53.2	17.2

In 43 % of flats were reported problems with odors coming either outdoors, or other flats or staircases. That was only problem with increased prevalence after renovation measures. Due to better indoor air quality the habitants may detect easier temporary odors than earlier.

Airing frequency decreased after new windows were installed. Especially the percentage of habitants airing several times daily decreased from 36 % to 15 %.

Winter 1997 was slightly milder than winter 1998. The weather conditions were not reason for the decreased percentage of the habitants feeling draught after new windows were installed. It is very important to take care of the whole ventilation system when conducting a large window renovation project in a blocks of flats. Otherwise the ventilation in flats will decrease.

The poor conditions of exhaust air ducts made from brick or concrete limits often possibilities to renovate or improve existing ventilation system with reasonable costs.

Further actions

During August and September 1998 exhaust air fans will be modernized, fan chambers tightened, new exhaust air vents installed and the exhaust air flows balanced. The aim is to increase the mean value of exhaust air flows by 20 %.

Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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Title : MEASUREMENT OF HEAT AND MASS TRANSFER THROUGH TYPICAL STAIRCASES

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MEASUREMENT OF HEAT AND MASS TRANSFER THROUGH TYPICAL STAIRCASES

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Synopsis

This paper is concerned with heat and mass transfer through two typical staircases. The first staircase connects the two individual floors of a two-storey building, and the other connects the three individual floors of a three-storey building. A series of experiments have been performed in order to study the buoyancy driven flow between the floors. A single tracer gas decay technique was adopted . Temperatures at various points on each floor were constantly monitored and air velocity measurements were also provided at some specific locations. The heat and mass flow rates between the two floors , through the first staircase, were calculated from the tracer gas concentrations. The analysis of experimental data gives relations for the mass and heat flow rate as a function of temperature difference between the floors, and of the geometry of the particular staircase. Simulations of the same configurations have been carried out, using validated CFD algorithms. Airflow rates estimated by these simulations showed very good agreement with experimental values. The mass flow rates through the second staircase, are estimated using the CFD method. In addition, the paper discusses the airflow patterns in the staircases.

1. Introduction

The study of energy and mass transfer between different zones in buildings has attracted increasing international interest and research effort. Airflow through vertical openings has been widely researched. However, little information is available regarding airflow in horizontal openings, such as staircases, especially that driven by buoyancy. Mass and energy transfer through staircases can have important implications regarding energy saving, thermal comfort, control of contaminants and spread of smoke in the interior of buildings.

A number of studies related to these phenomena, have been reported. Brown (1962) has investigated air flow through small square openings in horizontal partitions. Reynolds (1986) and Zohrabian et al (1989) have performed experiments in a scale model of a typical stairwell. Reynolds et al (1988), have also developed a model for buoyancy-driven flow in a stairwell. Riffat (1989) have studied the energy and mass transfer through a staircase in a two-floor house. Other experiments were conducted by Klobut and Siren (1994), to explore the influence of several parameters on combined forced and density-driven air flows through large openings in a horizontal partition. The above studies have been mainly of experimental or analytical nature. Studies based on the application of CFD are those of Zohrabian et al

(1989) and Riffat et al (1994), who used CFD modeling and compared predictions with experimental data. The objectives of this work are to study the buoyancy-driven air movement through two typical staircases of full-scale buildings , to compute the heat and mass transfer between floors, to compare the CFD predictions and measurements and consequently to improve the existing predictive methods of such processes.

2. Description of Experiments

A series of experiments were carried out in a two-storey building and in a three-storey building, in order to investigate the buoyancy-driven airflow through staircases that connect the floors of these buildings.The first staircase extends to a height of 6.3 m, while the lower and upper floor have an effective volume of 29.1 m^3 and 35.8 m^3 respectively. The second stairwell extends to a height of 13.0 m. The lower, central and upper floor have an effective volume of 41.5 m^3 , 24 m^3 and 40.0 m^3 respectively. Since the geometry is rather complex, figures can describe the buildings in a more effective way. Figure 1 shows schematic diagrams of these buildings with the main dimensions and the locations of the instruments.

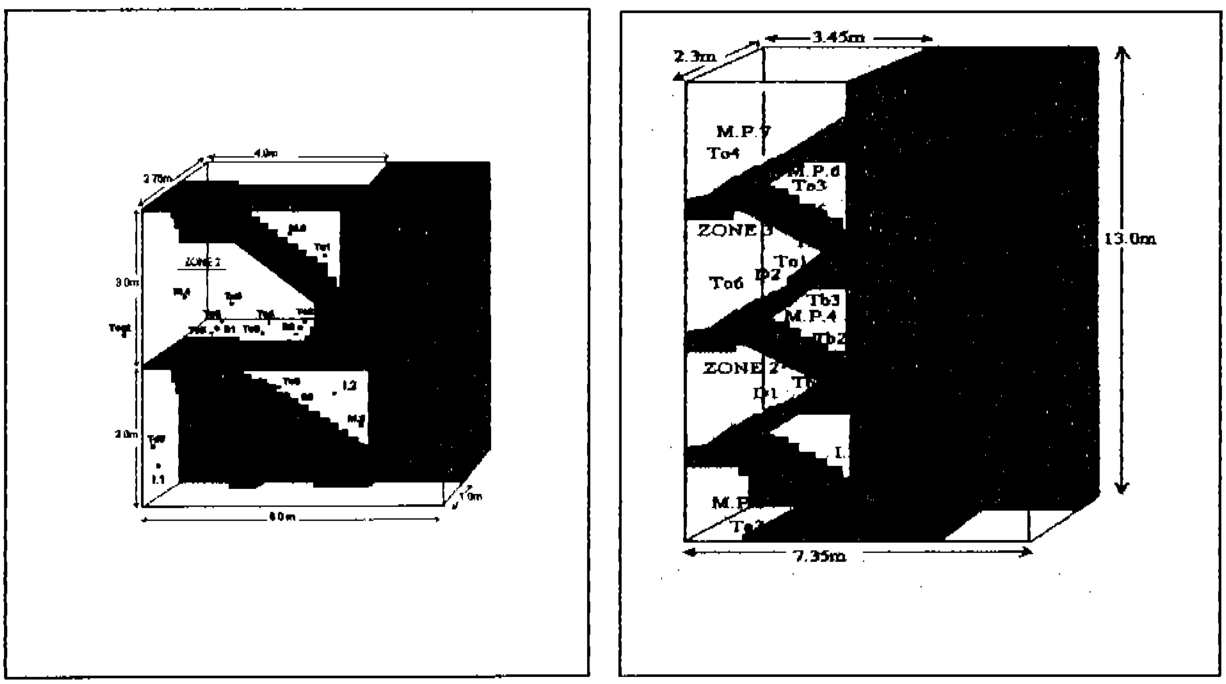

Figure 1: Schematic diagrams of the two staircases and instrumentation.
(T : temperature sensor,D : air velocity sensor,
M:measuring point and I : injection point).

All the openings, such as the main door, the doors connecting the staircases with the appartments and the windows were kept closed and sealed during all the experiments. Some small openings and cracks were also sealed in order to reduce the infiltration as much as possible. The mass and heat flow rates between the floors are mainly determined by the size and geometry of the openings connecting the floors. These horizontal openings are defined by the staircases geometries. The opening of the first staircase has dimensions of 1.5 m by 2.7 m while the two openings connecting the three floors of the second staircase have dimensions of 2.15 m by 2.30 m (Fig.1).Furthermore, an additional obstacle was placed suitably across the opening of the first staircase, decreasing its size. This modified opening had dimensions of 1.5m by 2.15 m.Under this configuration, additional experiments were carried out in order to study the effect of size of the opening on the values of the rates mentioned above. The experiments characterized by the original configuration, will be referred as runs with opening A, while the others will be referred as runs with opening B. The mass and heat flow rates between the floors are also affected by the temperature differences of these floors. The temperature differences between the floors were defined as the differences of the average storey temperatures. In order to investigate this dependence, the lower zone was heated while the other floors were unheated. This was done by using thermostatically controlled heaters. The airflow rates between the floors were measured using a single tracer gas decay technique. Several tracer gases are available , but $N_2 0$ was chosen for this work since it has desirable characteristics in terms of detectability, safety and cost and it has been used successfully in previous air movement studies. The concentration of gas was measured using an infrared gas analyzer (accuracy:± 1%). At the beginning of each experiment the opening between the two floors at the first building and similarly the opening between the lower and the central floor at the second experimental building, were closed by PVC sheets and every gap between these sheets and the adjacent surfaces was sealed. Tracer gas was released in the lower floor (zone 1), where it was mixed with air. This was accelerated by using small fans near the injection points of gas. After uniformity had been achieved in zone 1, the PVC sheets were removed and the evolution of tracer gas concentration in all zones, was monitored.

Applying the tracer material balances in the two zones of the first experimental building , the rate of change of tracer concentration in zone 1 and in zone 2 at time t are given respectively by :

$$V_1 \, dC_1/dt = - C_1 (Q_{10}+Q_{12}) + C_2 Q_{21} \qquad (1)$$

$$V_2 \, dC_2/dt = C_1 Q_{12} - C_2 (Q_{21}+Q_{20}) \qquad (2)$$

where V_1 and V_2 are the effective volumes of each zone, Q_{10} and Q_{20} are the volumetric flow rates of air that exfiltrate from each zone to the outside,Q_{01} and Q_{02} are the volumetric flow rates of air that infiltrate from outside into each zone and Q_{12} and Q_{21} are the volumetric flow rates of air that exchanges between the two zones through the stairwell in both directions. C_1 is the concentration of the tracer at time t in zone 1 and similarly C_2 is the concentration of the tracer at time t in zone 2. The other two flow volumetric rates can be determined using the continuity equations:

$$Q_{01} = Q_{10} + Q_{12} - Q_{21} \qquad (3)$$

$$Q_{02} = Q_{20} + Q_{21} - Q_{12} \qquad (4)$$

These equations are valid, provided that a steady state exists and that the concentration of tracer gas in the outside air is negligible. These volumetric-balance equations can be solved using the theoretical technique based on the Sinden (1978) method. A similar method was adopted by Afonso and Maldonado (1986). According to this method, a multizone system may be represented by a series of cells of known and constant volume that are all connected to a cell of infinitely large volume (outside). Since the unknown flow rates involved in these equations are six, it was suggested by Sinden that equations (1) and (2) could be integrated from two different intervals from the concentration decay curves for each zone. This technique would yield the necessary number of equations to solve for all the unknowns. Equations were integrated using Simpson's rule and the final system of simultaneous equations was solved numerically by Gauss elimination. This method was employed only for the first staircase, since it's implementation results to significant errors when it is applied in buildings with more than two zones. In these situations multiple tracer-gas procedures are recommended (Roulet and Vandaele (1991)). For this reason , the airflow rates through the second staircase, are estimated using the CFD method.

The air temperature was constantly monitored by thirteen thermocouples (accuracy: ± 0.2 ° C) which had already been calibrated (Fig. 1). Surface temperatures on almost all the internal surfaces, were measured by an infrared thermometer (accuracy: ± 0.1%). Limited air velocity measurements were also provided by three air velocity sensors (accuracy:± 0.01 m/sec ± 5%) (Fig. 1).

3. Measurements and Results

Eleven experiments were performed in the two-storey building and five in the three-storey building described above, under various temperature differences between the floors. The lower floor was heated for a long time before the beginning of monitoring so as to reach thermal equilibrium. The measurements verified the initial assumption that every floor can be considered as a different zone at least for flows driven by buoyancy as those investigated here. Figure 2 shows the tracer gas concentration level in zones 1 and 2 against time during the second run as it was monitored in the first building and similarly, Figure 3 shows the tracer gas concentration level in zones 1, 2 and 3 against time during the first experiment in the three-storey building.

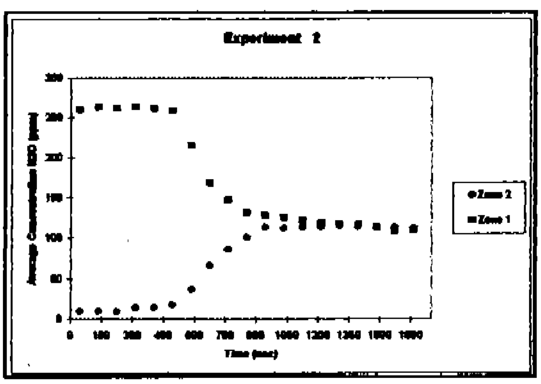

Figure 2: Variation of concentration of N_20 with time (First Building)

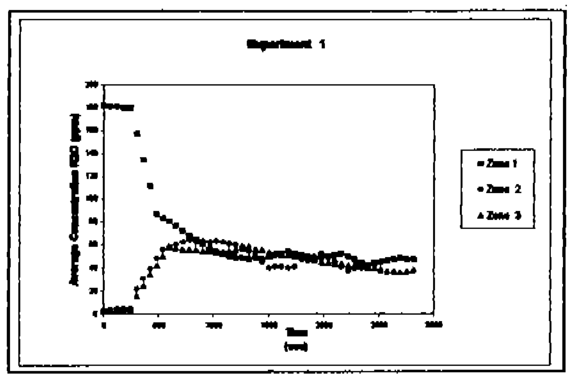

Figure 3: Variation of concentration of N_20 with time (Second Building)

The average temperature difference between the zones for the duration of each run, the ambient temperature and the wind speed for all the experiments carried out in the first building and in the second one are given in Table I and II respectively.

The airflow volumetric rates between the two zones of the first building, were estimated from these tracer gas concentration data using the method described above. Since infiltration and exfiltration of air, due to the temperature difference between the inside and outside of the building and the wind speed can affect the interzonal airflow, the induced flow was subtracted from the total airflow between the two floors. The results verified that the airflow rate between the two zones is a function of the temperature difference between the zones and of the size of the opening.

Table I
Experimental Conditions (First Building)

Run	*Average Temperature Difference between zones 1 and 2 (° C)*	*Ambient Temperature (° C)*	*Wind Speed (m/sec)*
1	6.2	14.8	1.8
2	0.5	15.6	1.6
3	0.2	16.7	1.5
4	3.3	17.5	1.4
5	1.1	17.5	1.3
6	4.9	19.5	1.4
7	3.4	19.6	1.7
8	0.7	18.1	1.3
9	2.4	19.6	1.7
10	0.1	20.4	1.2
11	2.7	20.0	1.3

Table II
Experimental Conditions (Second Building)

Run	*Average Temperature Difference between zones 1 and 2 (° C)*	*Average Temperature Difference between zones 2 and 3 (° C)*	*Ambient Temperature (° C)*	*Wind Speed (m/sec)*
1	3.1	0.2	18.1	2.5
2	4.0	0.6	17.7	2.8
3	4.8	0.5	17.2	1.9
4	0.9	0.2	19.0	2.1
5	2.3	0.1	19.1	2.4

Applying Bernoulli's equation [1], the volumetric flow rate and the mass flow rate through a horizontal opening separating two zones can be given approximately by :

$$Q = A\ C\ \sqrt{\Delta T\, g\, H/T} \tag{5}$$

and

$$M = \rho\, A\ C\ \sqrt{\Delta T\, g\, H/T} \tag{6}$$

where A is the cross-sectional area of the opening, C is the coefficient of discharge, ΔT is the average temperature difference between the zones, T is the mean absolute temperature of the two zones, ρ is the average air density and and H is the thickness of the partition separating the zones. This equation was employed despite the complex geometry of the specific building. To evaluate the coefficient of discharge for this opening, the measured volumetric airflow rate was divided by the theoretical one, given by equation (5). Since two different opening sizes (opening A and B) were investigated as mentioned before, it was found that the coefficient of discharge was rather independent of the opening size.

It was also found to decrease from about 0.76 to 0.34 as the temperature difference between the two floors increased from 0.2 to 6.2 ° C (Figure 4). This decrease in the coefficient of discharge may be due to an increase in interfacial mixing as a result of the direct transfer of some air from the upper zone into the inflowing warmer air from the lower zone. It also appeared to remain constant near 0.34 for high temperature differences. Brown (1962) and Riffat (1989) investigated similar phenomena and suggested comparable values for this coefficient. The coefficient of discharge C and the dimensionless ratio $\Delta T/T$ were correlated very well (with r-squared value equal to 0.96) :

$$C = 0.1469\ (\Delta T/T)^{-0.2} \tag{7}$$

From equations (6) and (5), the mass flow rate between the two zones can be given by :

$$M = 0.1469\ \rho\ A\ \sqrt{gH}\ \ (\Delta T/T)^{0.3} \tag{8}$$

and similarly the heat flow rate between the two zones can be given in the form :

$$Q = 0.1469\ \rho\ c_p\ A\ \sqrt{gH}\ \ (\Delta T)^{1.3} / T^{0.3} \tag{9}$$

where c_p is the specific heat of air. These equations indicates that the mass flow rate through the first staircase increase linearly with $(\Delta T/T)^{0.3}$ for each opening configuration. A plot of the mass flow rate against $(\Delta T/T)^{0.3}$ for both opening sizes of the first building is shown in Figure 5. The airflow rates through the second staircase, are going to be estimated by using the CFD method for the reasons mentioned before.

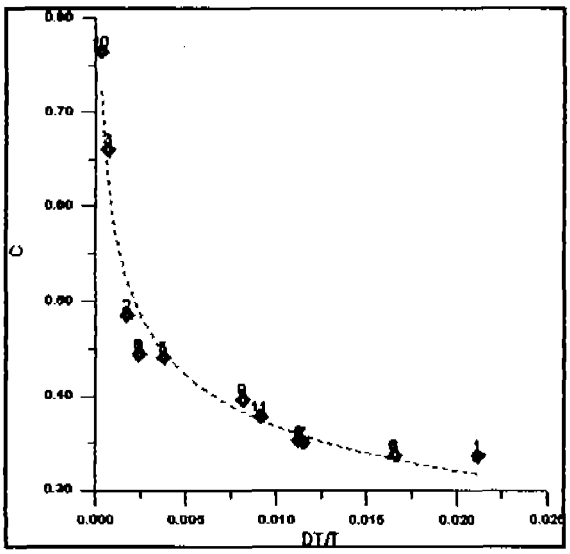

Figure 4: Variation of coefficient of discharge with $\Delta T/T$ (First Building)

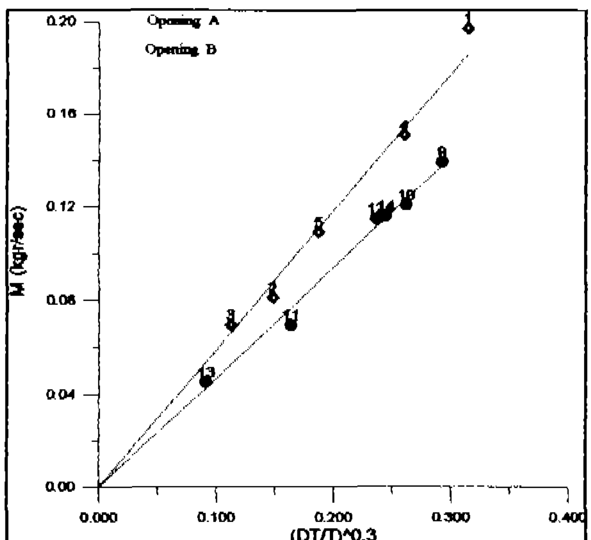

Figure 5: Mass flow rate against $(\Delta T/T)^{0.3}$ (First Building)

4. CFD Simulations

Simulations were performed, applying CFD algorithms which had already been developed and extensively validated. These algorithms generate approximate solutions to the Navier-Stokes equations, i.e. the conservation equations of mass, momentum, thermal energy and concentration species. In particular, these algorithms used the finite volume method, based on a Cartesian grid and the power–law interpolation scheme. To derive the pressure the SIMPLE algorithm was employed. After descretization these equations were solved numerically using the Tri-Diagonal Matrix algorithm. These algorithms incorporated the more recent RNG k-ε model based on Renormilized Group theory established by Yakhot and Orszag (1992). The advantage of this model is that it is valid for a very wide range of flow types including both high and low Reynolds number flows. To simulate the highly transient mass and energy transfer between zones, time dependent versions of the above equations were used. Small time steps of around 1/10 of the characteristic time scale were used in order to improve the accuracy of results. The computations were performed over the whole period of each run. Computations were also three-dimensional since the flow had been expected to be highly assymetrical due to complex geometry of the staircases. For simplicity, an orthogonal, equally spaced grid system was used to cover the domain.

First, the CFD algorithms were employed to simulate the cases corresponding to the eleven runs of the first building described in Table I. Five different grid sizes were investigated to determine the necessary resolution for grid-independent solutions. These grid sizes ranged from 2160 cells (15x9x16) to 35280 cells (42x20x42). Comparison of simulated volumetric flow rate between the zones showed that a grid-independent solution was achieved by using a 30x14x30 mesh (12600 cells). Since our main goal was only to model the general flow patterns and to predict the volumetric flow rate between the zones, further refinement of the grid would not produce any appreciable gains in accuracy. The criterion for convergence was to achieve a small value (~ 10^{-4} kgr sec^{-1}) for the sum of the mass residuals at the end of each time step. However, the convergence becomes quite problematic in low-Reynolds number flows. Tight control with variable relaxation factors were applied during simulations in order to achieve convergence. Care was taken to set the boundary conditions and the initial conditions of each zone, for the concentration level ,air and surface temperature which are necessary for the simulations. These values were provided by the experiments.

As the simulation proceeded, mass and energy transfer between the two zones caused variations in air temperature and velocity. These were recorded and analysed to reveal flow patterns , airflow rates and their variation with time.The volumetric flow rate between the two zones is obtained using :

$$Q = \sum_{i=1}^{n} | Wi | Ai/2 \qquad (10)$$

where n is the number of cells within the span of the horizontal opening connecting the two floors,W_i is the vertical component of air velocity at individual grid points within the opening and A_i is the area perpedicular to W_i of individual cells within the opening. These rates were calculated at the end of each time step and averaged over the whole duration of each experiment. The final values were compared with experimental measurements. The relative difference between the simulated predictions and experimental estimations ranged from about 2.0 % to 11.6 % , with an average value equal to 5.5 % for all the experiments. This is a very good agreement, considering the many factors - such as the turbulence model, experimental errors and boundary conditions- which affected the accuracy of results.

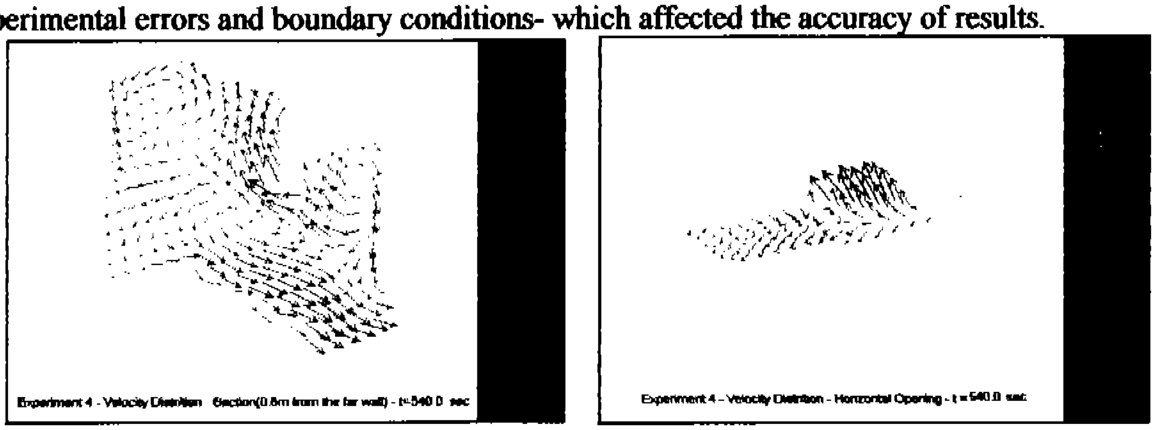

Figure 6: Air flow pattern through the first staircase and in the opening at t =540 sec during experiment 4.

These simulations revealed also the airflow patterns in the zones as well as in the opening connecting the two zones. It was found that these patterns are affected mainly by the temperature difference between the floors and by the geometry. The flow patterns in the zones were dominated by many vortices which promoted heat transfer and uniformity. The higher the temperature difference was, the more intense the eddies are. These flow patterns varied with time. Figure 6 shows examples of the predicted airflow through the first staircase and in the opening, at a moment 540 sec into the fourth experiment.

In order to estimate the airflow rates between the three zones of the second building, simulations of the cases corresponding to the five runs are carried out. From the first simulations, the convergence appears more problematic than before. So care must be taken for the appropriate relaxation factors. This work is still in progress.

5. Conclusions

The mass and heat flow rate through the opening connecting the two zones was a function of the interzonal average temperature difference and of the size of the opening. The mass and heat flow rate increased significantly with increasing temperature difference. In particular the mass flow rate increased linearly with $(\Delta T/T)^{0.3}$. Further experimental work is required to investigate the possible effect of staircase geometry and size of the opening on the value of these rates.

CFD simulations of buoyancy-driven flow through two typical staircases, were carried out. These simulations revealed the general airflow patterns. Comparison of the simulated volumetric airflow rates between the two zones of the first building and those based on experimental measurements, showed very good agreement, despite the difficulties of CFD method to model these phenomena. Investigation is required so as to improve the effectiveness of these algorithms to model these airflow phenomena.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19th AIVC Conference, Oslo, NORWAY
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VENTILATION STRATEGIES FOR THERMAL PERFORMANCE IMPROVEMENT OF AN ATTACHED SUNSPACE

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Ventilation Strategies for Thermal Performance Improvement of an Attached Sunspace

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SYNOPSIS

In this paper ventilation strategies are examined in order to improve the thermal performance of an attached sunspace of a two-storey semi-detached house in the area of Athens Greece. The ventilation strategies examined are cross and single-sided ventilation through the vertical windows of the sunspace. Simulations were conducted implementing multizone ventilation model COMIS coupled with the thermal simulation model Suncode. Wind pressure distribution is estimated using a wind pressure parametrical model and results of wind tunnel experiments.

It was concluded among others that ventilation strategies appear to be important for the energy control and for the formation of the temperature variations in the attached sunspace. Incorrect use of the windows could turn over the benefits of the bioclimatic design. Keeping the windows closed during winter makes the sunspace energy efficient and energy independent in most hours of the day for almost all of the examined mild climates (Greek and USA cities). In some of the examined climates the risk of overheating is likely to happen even in winter. The ventilation strategies during the summer period affect temperature variations significantly less, mainly because of the shading devices used to block direct incoming solar radiation. As its was derived from the thermal balance diagrams in most cases the attached sunspace contributes significantly to the heating demands of the house.

Keywords: ventilation strategies, air flow, thermal coupling, sunspace, bioclimatic design.

1. INTRODUCTION

In buildings located in mild climates, ventilation strategies utilise mainly natural ventilation techniques in order to reduce energy demand and to meet the specifications of relevant Building Regulations or Codes of Practice. The user's behaviour relating to the use of external openings and ventilation systems should also be taken in to consideration as a part of the overall ventilation design. To justify such complex strategies, it is necessary to examine all the interrelated thermal and ventilation phenomena. Design ultimately rests with factors as outdoor climate, building environment, thermal and airtightness characteristics of the building's envelope, as well as bioclimatic design.

Attached sunspaces used as passive solar systems are especially effective in mild climates. The design criteria of the attached sunspace should be seen by two points of view: during the heating period solar gains should be maximised and ventilation and conduction losses should be reduced, while during the cooling period solar gains should be minimised and ventilation strategies should be used to remove the excessive heat.

The above mentioned design criteria should always be combined with the prior mentioned factors each one consisting of various parameters. For example building environment which is mainly depended on the nearby buildings affects the wind pressure coefficients on building envelopes reducing the ventilation rates and also reduces the incoming solar radiation. It is therefore essential to simulate all thermal and ventilation phenomena with the same degree of detail, in order to examine ventilation strategies for thermal performance improvement of the attached sunspace.

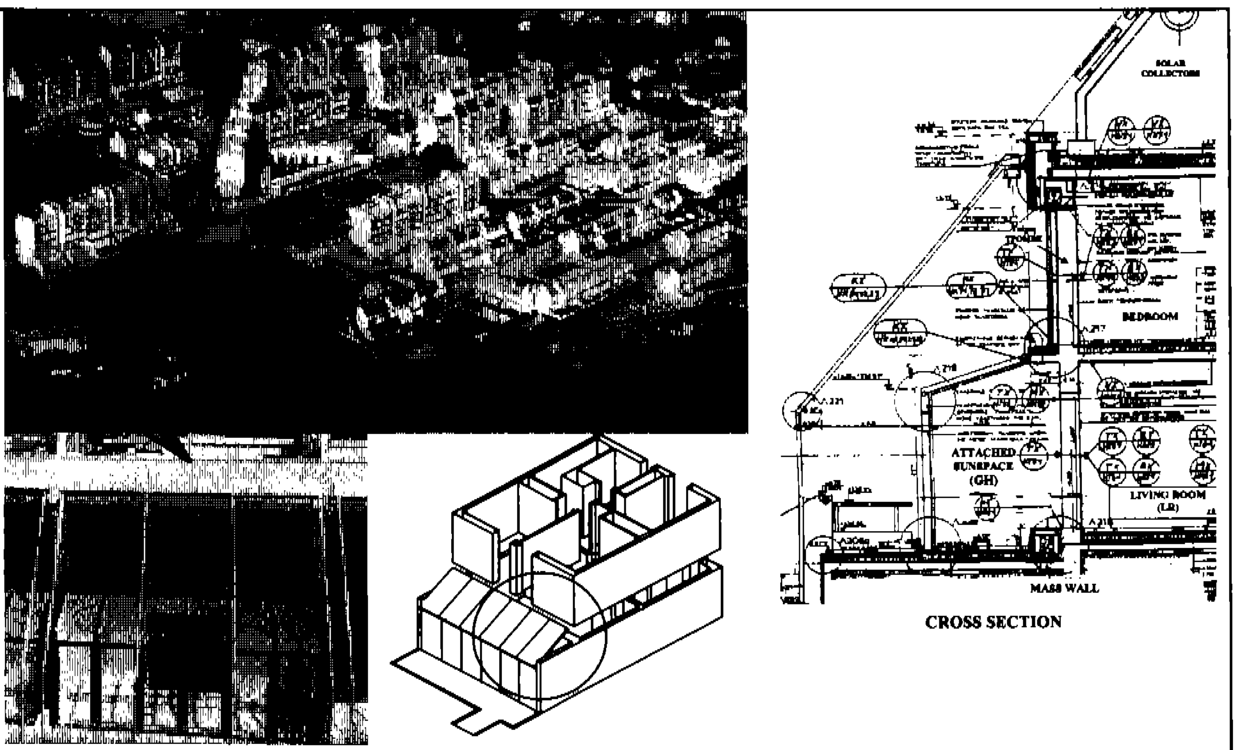
2. DESCRIPTION

The sunspace is attached to a two-storey semi-detached house (maisonette) placed at a block of five similar semi-detached maisonettes. It is located at the Solar Village, a housing project of Greek Workers' Housing Organisation (OEK) and completed in the early 90's in Pefki Athens. It consists of 435 apartments distributed in 30 buildings, an energy centre, a solar information centre and a commercial and community centre (figure 1).

The maisonette consists of a living room, a kitchen and a WC at the ground floor and three bedrooms and a bathroom at the floor, at a total area of 105 m^2. The attached sunspace covers an additional area of 24 m^2 approximately at the ground floor (figure 1 right). There are also two Trombe walls located at the south-facing bedrooms of the first floor.

The examined maisonette is heavily insulated with 10 cm of mineral wool in the external walls and with 10 cm polystyrene boards, on the roof. The inner walls are of 10 cm thick brick, covered with 2 cm plaster on each side. The outer envelope consists of 20 cm thick brick walls, 20 cm thick concrete frame and 15 cm thick concrete slabs. Increased thermal mass is placed at the sunspace by means of concrete mass walls, floor and sidefins.

The maisonette was used to validate the air flow model and the thermal model as well as the coupling of the two models (Koinakis 1998). This prior work has been implemented in this paper to study the thermal impact of ventilation strategies at the examined attached sunspace.



AERIAL AND CLOSE UP VIEWS OF THE HOUSE BLOCK, THE MAISONETTE AND THE ATTACHED SUNSPACE

Figure 1: Views and cross section of the examined attached sunspace

3. SIMULATIONS

The basis of the study of the ventilation strategies is the modelling of the air flow phenomena including infiltration and flow through large openings. Thermal and ventilation phenomena appear to be remarkably intense in the attached sunspace due to the following reasons:

- the glazed area is very increased compared to the rest of the house,
- there are various possible combinations of closed and opened windows and
- the thermal phenomena are very intense mainly due to incoming solar radiation and the risk of overheating.

The attached sunspace is used as a part of the bioclimatic design, in order to contribute to the energy demand and to be used as an additional living area attached to the living room. It is therefore important to fulfil the following demands:

- to ensure incoming interzonal thermal flows to the main house during the heating period
- to achieve air temperatures inside the greenhouse closer to the comfort temperature (e.g. 18 ^{0}C) during the heating period
- to retain air temperature inside the greenhouse between reasonable limits (e.g. between 18^{0}C and 25.6^{0}C), implementing solar shading and night cooling to avoid overheating.

In order to examine whether the above demands were fulfilled it was decided to study the sunspace during 6 day heating and the cooling periods, covering from 19 to 24 of January and from 19 to 24 of July respectively. Six indicative mild climatic regions were selected, corresponding to three main Greek cities: Athens, Thessaloniki and Herakleion, as well as to three USA cities: Los Angeles CA, New York City NY and Phoenix AR. The climatic data was derived from reference years. The Greek climatic files were based on 10 year period data (Koinakis 1998) and the USA climatic files were based on the TMY2 data (Marion & Urban, NREL 1995).

Three indicative ventilation strategies were examined during the simulations: all external windows closed, all half-opened and only two windows at the two corners of the sunspace half-opened. No auxiliary heating systems and no ventilators were used. Solar shading is the only control measure implemented during the cooling period in order to reduce the risk of summer overheating.

As previously mentioned, air flow was simulated implementing the nodal airflow model COMIS (Feustel 1991). Pressure distribution on building's envelope was calculated using the parametrical model cp-calc (Grosso 1992). The results for the Cp values found to be similar to those of wind tunnel experiments (Wiren 1987), for building identical to the block of the maisonettes. The modelling of infiltration as well as the flow through large openings is described in COMIS Fundamentals (Feustel 1991).

The nodal airflow model was coupled with the nodal thermal simulation model Suncode, modifying the source codes of the two models where necessary. The procedure of the thermal coupling is presented in figure 2. The thermodynamic integrity for each time step is guarded by the sequential coupling method, in which the thermal and the air flow model run in sequence: each model uses the results of the other in the previous time step. In the present simulations the air flow calculations use air temperatures calculated in the previous time step. The above mentioned technique was validated as part of another task (Koinakis 1998), using real case experimental data and found to be sufficient and accurate, giving the potential to simulate a wide range of thermal and ventilation phenomena and ventilation strategies.

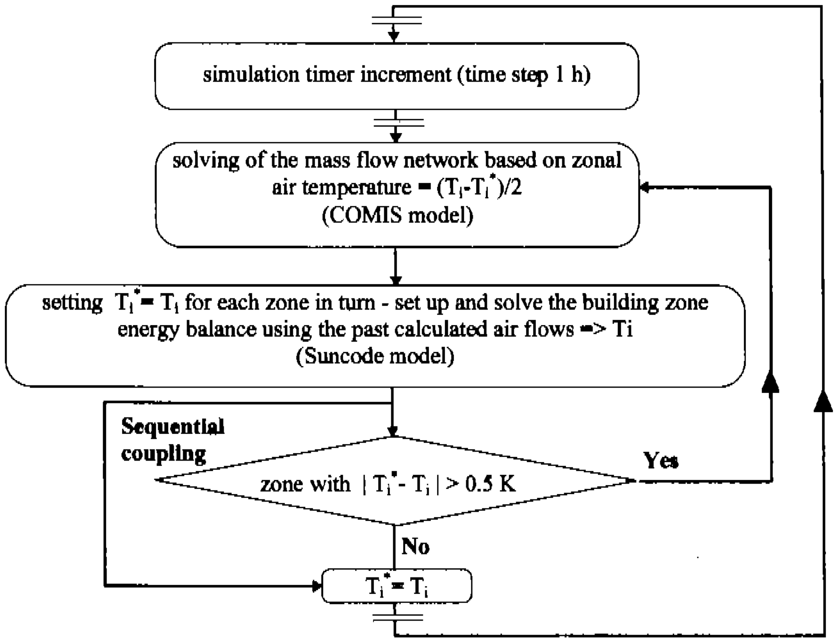


Figure 2: Schematic flow diagram showing the implementation of sequential coupling of air flow and energy balance calculations.

The energy balance equation defining the air temperature, T, of the attached sunspace can be written as:

$Qwall + Qwindows + Qinf\&vent + Qsolzon = 0$

where:
$Qwall$ is the energy flow between the sunspace and all the enclosing mass walls (=floor, mass wall of the living room)
$Qwindows$ is the energy flow through all kinds of windows
$Qinf\&vent$ is the energy flow due to air infiltration and ventilation and is given by the equation: $Qinf\&vent = UAinf\&vent*(Tamb - T)$

where:

$Tamb$ is the ambient air temperature
$UAinf\&vent$ is the infiltration/ventilation equivalent conductance value and is given by the equation: $UAinf\&vent = VOL*Cair*Pair*e^{a*elev}*AC$
where:

VOL is the zone air volume, $Cair$ is the air specific heat, $Pair$ is the air density at sea level, a is a coefficient derived fron exponential curve fit and, AC is the air change rate derived fron the air flow model.

More detailed analysis is given at the bibliography (Weeling & Palmiter 1985, Koinakis 1998).

None heating venting or cooling system is used in the sunspace, while the rest of the house is heated and cooled all year using steady heating and cooling setpoints (20^{0}C and 25.6^{0}C respectively.

4. RESULTS

Figures 3.1 and 3.2 present the hourly variations of the air temperatures in the attached sunspace for the three indicative ventilation strategies. The ambient air temperature and the total horizontal solar radiation hourly variations are also presented, in order to allow direct comparison of thermal performance. January and July simulations are presented at the left and right columns respectively.

The daily energy balances of the sunspace are presented in figures 4.1 and 4.2. The energy balance presented in these figures, consist of five terms: i) the solar gains through the glazed area, ii) the heat flows through the ground floor slab, iii) the flows through the windows due to conduction, iv) the interzonal flows to/from the adjacent living room, v) the energy flows due to infiltration and ventilation.

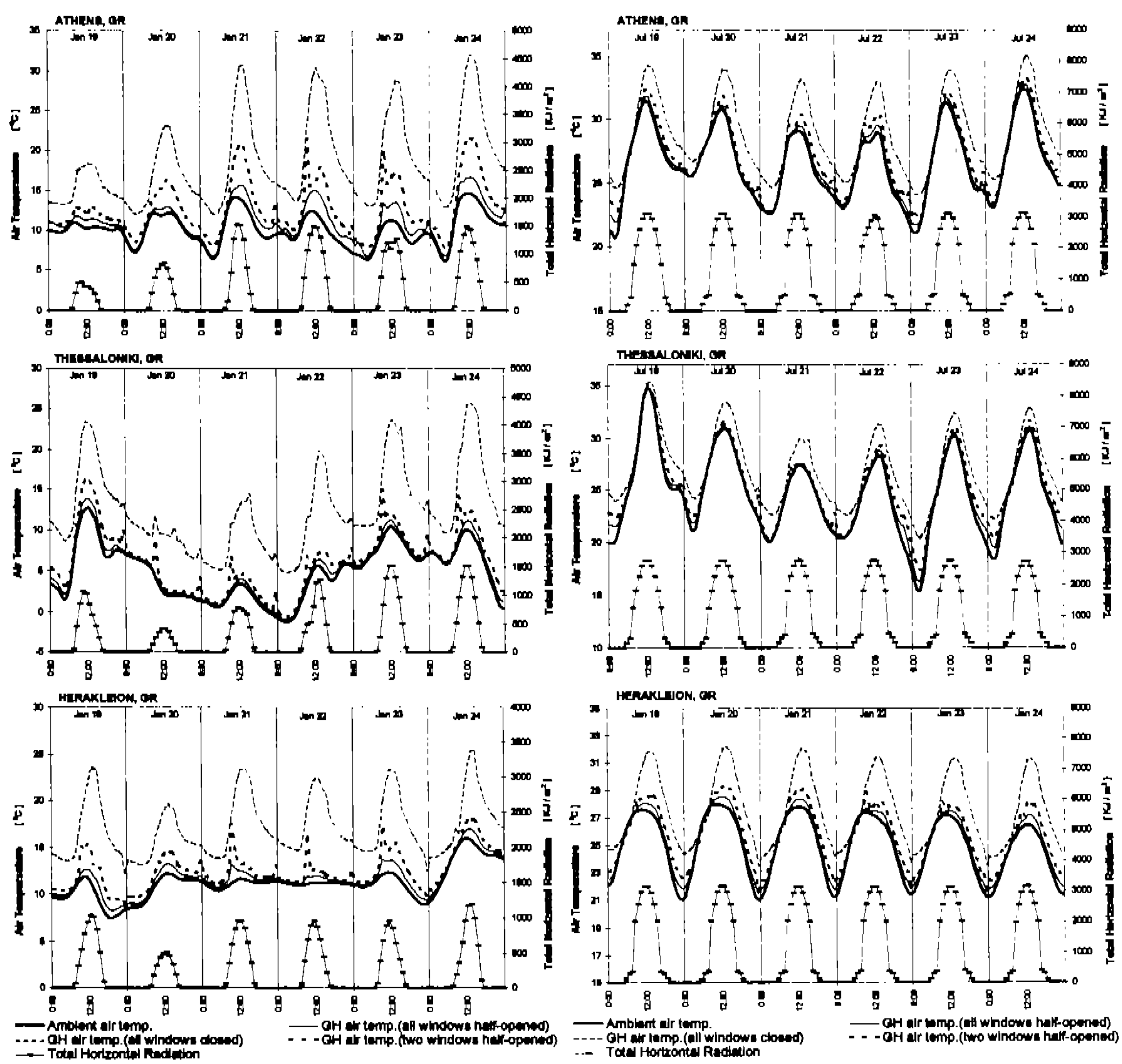


Figure 3.1: Hourly variations of the air temperatures in the attached sunspace for the selected ventilation strategies. Corresponding ambient temperatures and solar radiation. Greek climatic files.

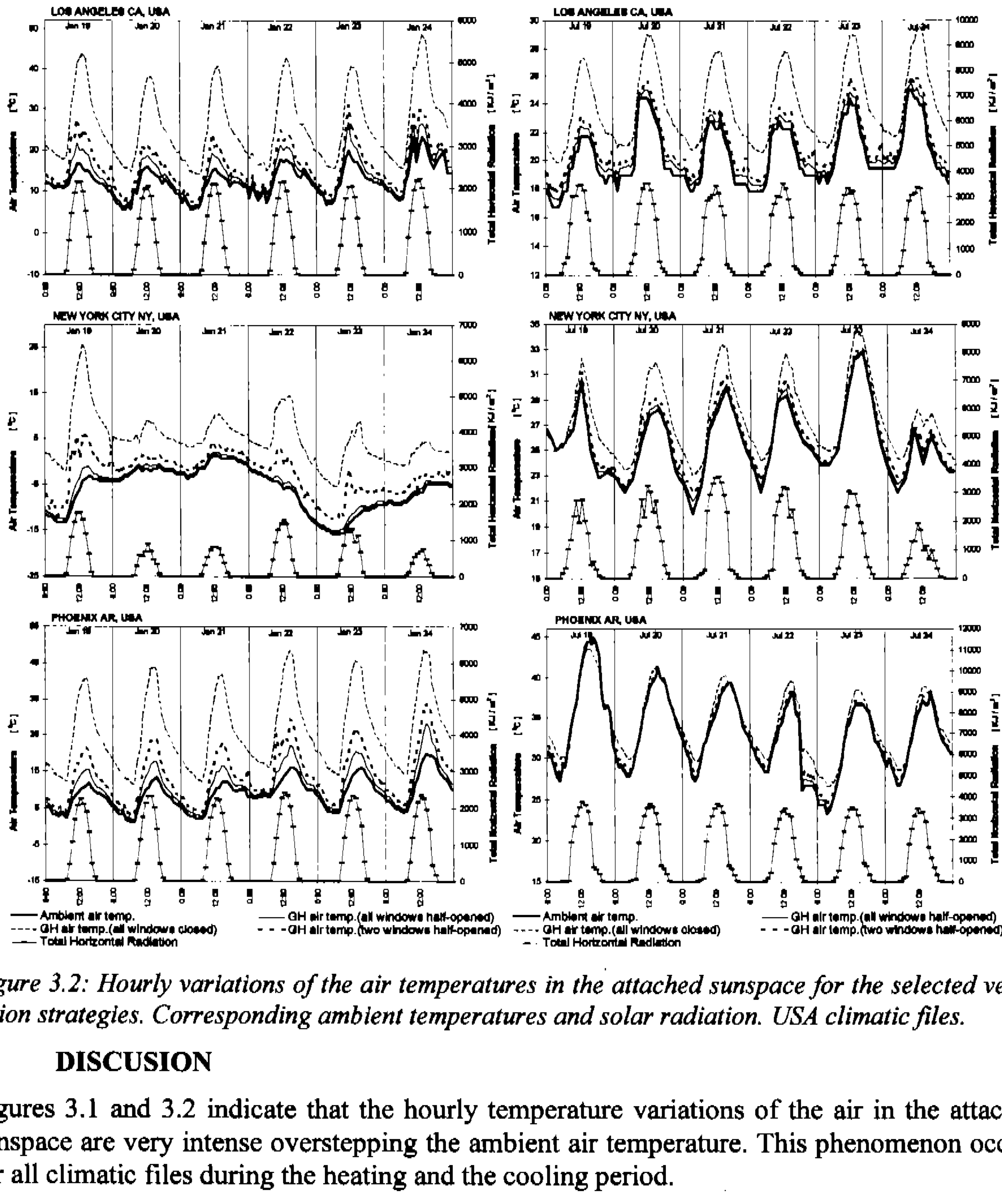


Figure 3.2: Hourly variations of the air temperatures in the attached sunspace for the selected ventilation strategies. Corresponding ambient temperatures and solar radiation. USA climatic files.

5. DISCUSION

Figures 3.1 and 3.2 indicate that the hourly temperature variations of the air in the attached sunspace are very intense overstepping the ambient air temperature. This phenomenon occurs for all climatic files during the heating and the cooling period.

The ventilation strategies affect these variations very significantly especially during winter noon. Keeping the windows closed during winter makes the sunspace energy efficient and energy independent in most hours of the day in almost all the examined mild climates (except New York City). In addition the risk of winter overheating is always present in Los Angeles and Phoenix, while it is almost avoided in the Greek climatic files. Even the slightest opening of the windows affects temperature variations dramatically.

The ventilation strategies during the summer period affect temperature variations significantly less, mainly because of the shading devices used to block direct incoming solar radiation (the shading coefficient used equals 0.7). Temperature variations were also dramatically affected by even the slightest opening of the windows.

Comparing solar radiation and air temperature variations leads to the conclusion that solar radiation could affect indoor air temperatures significantly only in the case of completely closed windows.

The effect of the examined ventilation strategies on indoor temperatures is expressed in heating and cooling degree-hours and compared with ambient temperatures at the tables 1 and 2. It is derived among others that keeping all the windows closed during the winder could reduce the heating degree-hours from 7% to 55% of the ambient temperature degree-hours, depending on the climatic file examined. Keeping all the windows open during the summer period and using the appropriate shading devices increases the cooling-hours from 102% to 125% only, depending on the climatic file. The mild ocean climate of Los Angeles helps the sunspace to be used with zero thermal surcharge during summer.

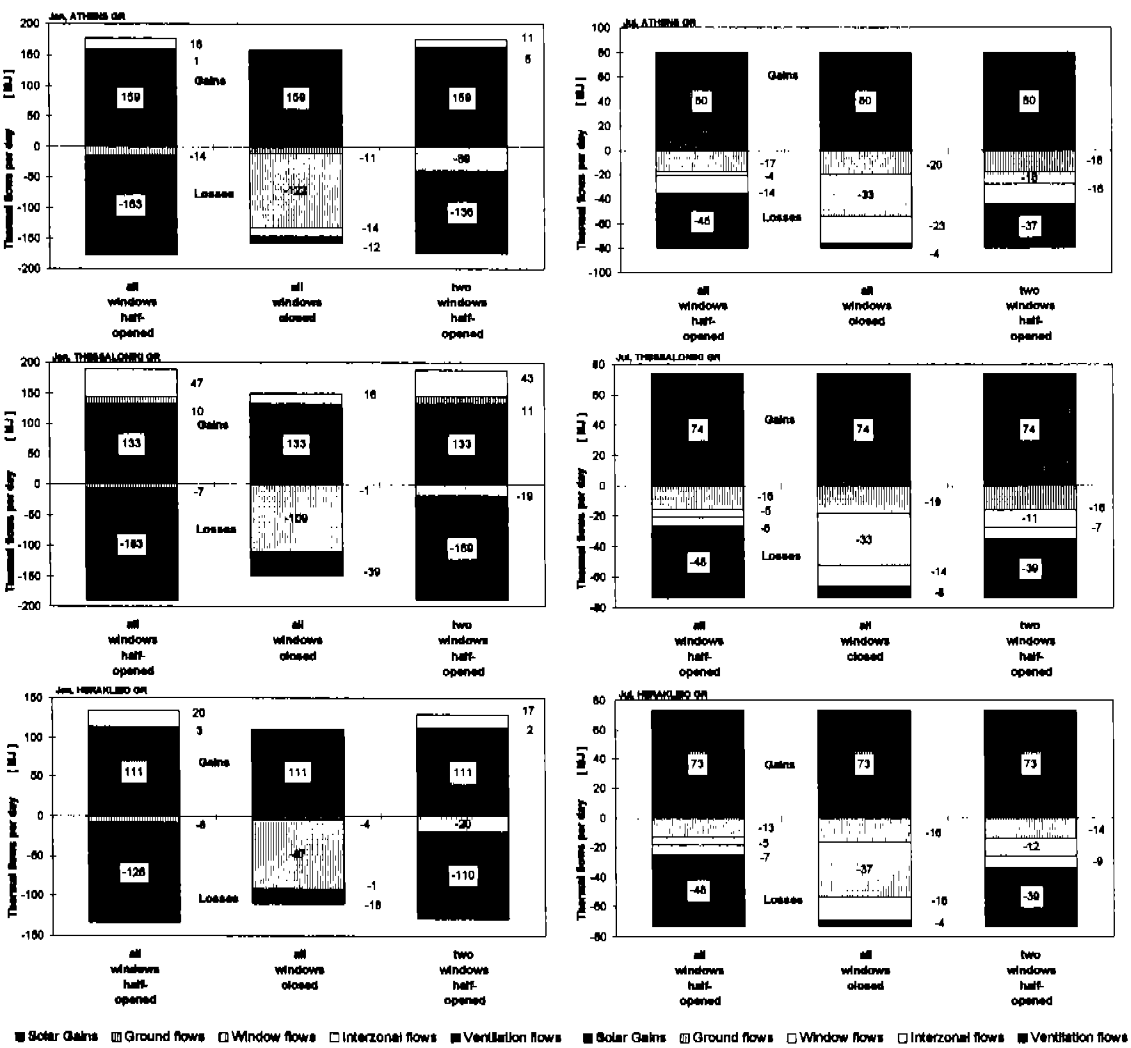


Figure 4.1: Daily thermal balance of the attached sunspace for the selected ventilation strategies. Greek climatic files.

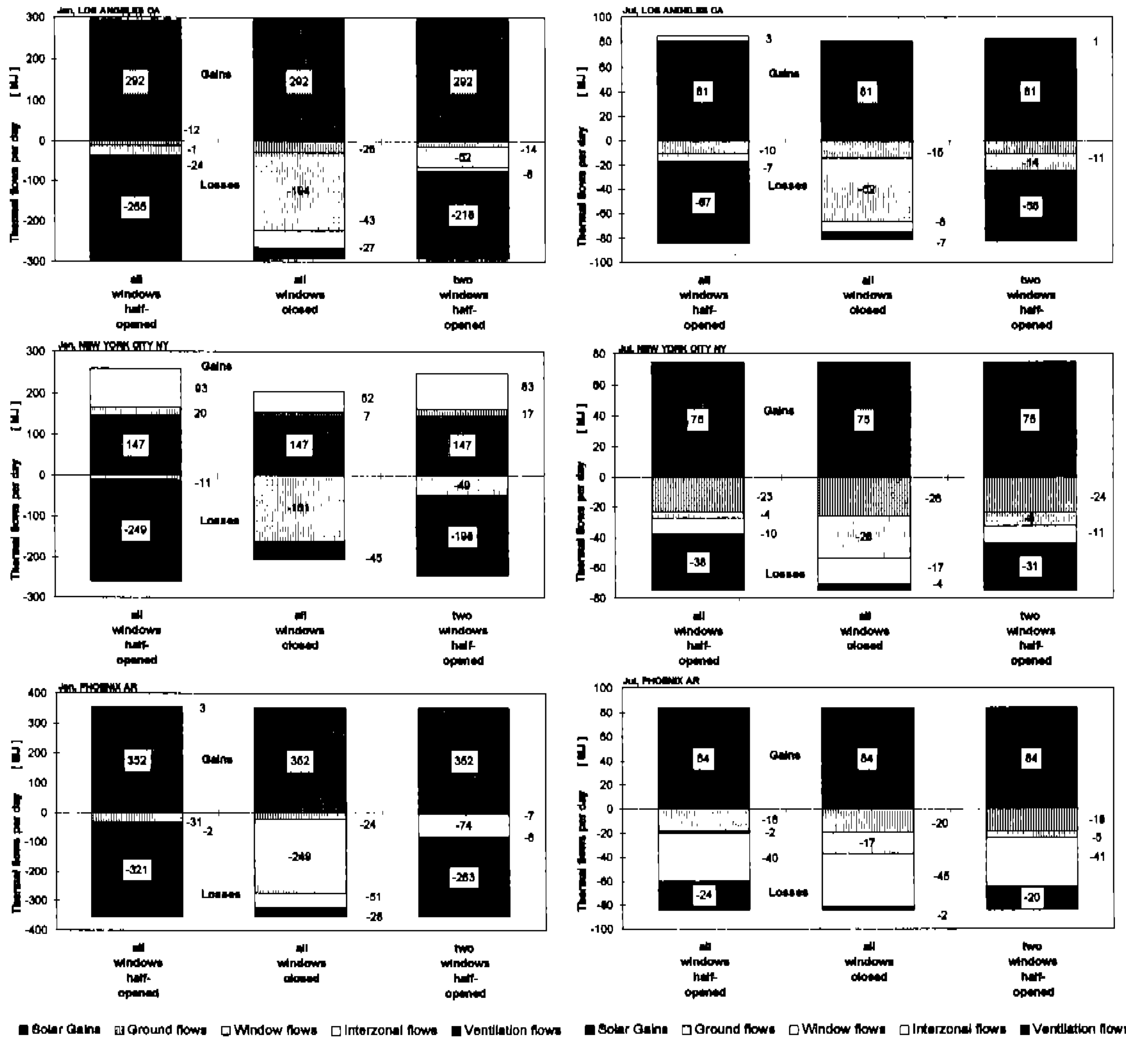


Figure 4.2: Daily thermal balance of the attached sunspace for the selected ventilation strategies. USA climatic files.

Figures 4.1 and 4.2 present among others the increased importance of the energy flows due to ventilation and infiltration in winter as well as in summer. The other important term of the thermal balance is solar gains. In most cases the attached sunspace contributes significantly to the heating demand and does not increase cooling loads significantly.

Table 1: Heating degree-hours. Mean daily values (base temperature 18 ^{0}C) (% - compared to ambient air temperature)

Climatic file	Ambient air temp.	Air temperature in the attached sunspace		
		all windows half-opened	all windows closed	two windows half-opened
Athens GR	189.9	166.5 (88%)	45.9 (24%)	131.4 (69%)
Thessaloniki GR	315.2	303.9 (96%)	150.5 (48%)	284 (90%)
Herakleion GR	163.4	150 (92%)	44.8 (27%)	130.6 (80%)
Los Angeles CA	136.7	109.1 (80%)	9.5 (7%)	83.3 (61%)
New York City NY	580.5	562.1 (97%)	317.8 (55%)	499.2 (86%)
Phoenix AR	220.1	177.5 (81%)	21 (10%)	133 (60%)

Table 2: Cooling degree-hours. Mean daily values (base temperature 25.6 ^0C)
(% - compared to ambient air temperature)

Climatic file	Ambient air temp.	Air temperature in the attached sunspace		
		all windows half-opened	all windows closed	two windows half-opened
Athens GR	42	45.8 (109%)	81.3 (194%)	51.9 (123%)
Thessaloniki GR	29.3	31.2 (107%)	52.5 (179%)	34.5 (118%)
Herakleion GR	15.3	19.1 (125%)	52 (340%)	24.6 (161%)
Los Angeles CA	0	0	13	0
New York City NY	25.9	28.6 (111%)	53.4 (206%)	32.7 (126%)
Phoenix AR	181.1	184.2 (102%)	208.1 (115%)	188.5 (104%)

6. CONCLUSIONS

Ventilation strategies appear to be important for the energy control of the attached sunspace. The incorrect use of the windows could turn over the benefits of the passive solar design. Even the slightest opening of the windows affects the air temperature in the sunspace. Keeping the windows closed during winter makes the sunspace energy efficient and energy independent in most hours of the day for almost all of the examined mild climates.

The ventilation strategies during the summer period affect temperature variations significantly less, mainly because of the shading devices used to block direct incoming solar radiation (the shading coefficient used equals to 0.7). Comparing solar radiation and air temperature variations leads to the conclusion that solar radiation could affect significantly indoor air temperature only in the case of completely closed windows.

Keeping all the windows closed during the winder could reduce the heating degree-hours from 7% to 55% of the ambient temperature degree-hours, depending on the climatic file examined. If all the windows kept half-opened the heating degree-hours are dramatically increased reaching 80% % to 97% of the ambient temperature degree-hours. Keeping all the windows open during the summer period and using the appropriate shading devices increases the cooling degree -hours from 102% to 125% only.

As it was derived from figures 4.1 and 4.2 in most cases the attached sunspace contributes significantly to the heating demand. The use of solar shading is absolutely necessary in order to keep down the cooling loads.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

DISPLACEMENT VENTILATION IN A CLASSROOM – INFLUENCE OF CONTAMINANT POSITION AND PHYSICAL ACTIVITY

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Displacement ventilation in a classroom – influence of contaminant position and physical activity

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Synopsis

This study describes how the air quality in a displacement ventilated classroom can be influenced by the position of a contaminating person, and by the activity of a person who walks around in the room. Tracer gas measurements have been performed in a full scale mock-up of a classroom, with person simulators at the student's desks.

The spreading of contaminants from a person seems to be strongly dependent on the position of the person. The closer the contaminating person sits to the outlet terminal(s), the less of his/hers contaminants are spread in the room. Paradoxically, people sitting furthest away from the air supply were found to be provided with the least contaminated air. Physical activity, produced by a walking person, tends to increase the concentration of contaminants emitted from people in the room, whereas the air exchange efficiency actually can benefit from it. At all levels of activity tested in this study the displacement ventilation system provided significantly better air quality than a mixing system would.

Introduction

The question of how to ventilate classrooms has gained increased attention in recent years in developed countries, where the problem of allergies has become acute. The classroom contribution to the allergy problem might to some extent lie in harmful emissions from building and furnishing materials, but probably even more in the spreading of allergens from one student to the other; all possibly in combination with poor ventilation flow rates.

When new ventilation systems are to be installed in schools, the choice nowadays often falls on the displacement system – at least this is the case in northern Europe. Its reputation of providing good ventilation effectiveness is probably the main reason for this. Often a two-zone model is used to describe the functioning of the displacement system: The air supply at a low level creates a lower zone of fresh air, whereas the contaminants are to be found in an upper zone, which is well mixed by thermal plumes from heat sources in the room.

However, an interesting question is whether the supply air reaches all people in a densely populated room like a classroom. One may suspect that the supply air, as it flows along the floor, is "stolen" by the convection flow along the bodies of the people sitting closest to the inlet terminal(s), leaving no fresh air left for the people sitting far way from the supply. Furthermore, the often good ventilation effectiveness has been shown in some previous studies (see [1]&[2]) to be sensitive to physical disturbances, like people's movements.
This study aims to describe how the air quality in a large, densely populated and displacement ventilated room is influenced by the position of a contaminating person, and also by the activity

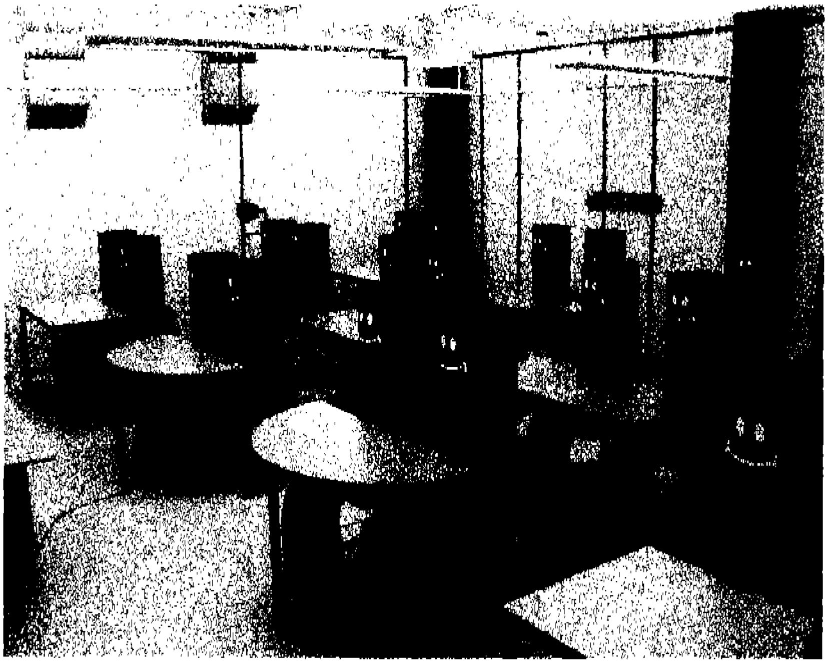

Figure 1. The classroom with the person simulators.

of a person who walks around in the room. To investigate this, tracer gas measurements have been performed in a classroom where students were represented by person simulators.

Method

The measurements were performed in a full scale mock-up of a classroom; size: 7.20x8.40 m, 3.00 m high. The well insulated classroom was situated in a laboratory hall, having an air temperature of about 22 °C. In the classroom, 24 students and a teacher were represented by person simulators (PS). Each PS was composed of one straight circular cylinder, constituting the main part of the body, and another, bent cylinder imitating the legs. The total area of a PS was 1.7 m^2, which is about the average of grown up (Caucasian) men and women. Each PS had an internal (electrical) heat production of 100 W, simulating human metabolism. The creation was covered by a cotton based textile fabric in order to get a relevant radiation emissivity of its surface. A quite even surface temperature distribution was attained.

A picture of the classroom with the person simulators is shown in figure 1, and a sketch of the set-up in figure 2. The doors to the outdoor climate simulation room were open all the time in order to have about the same temperature around the whole classroom. The two circular inlet terminals (diameter 320 mm) diffused the air all 360° between heights 435 and 830 mm above floor. The total air flow rate was 200 l/s, providing 8 l/s per person, which is in line with the recommendations for schools in Sweden. This gives an air exchange rate of 4.0 room volumes per hour. The inlet air temperature was kept at 14.5 °C; this low value was needed in order to keep the room air temperature at a reasonably low level. The two outlet terminals extracted the air at the uppermost 165 mm on the "right" wall, according to figure 2. Fluorescent lamps, developing a total power of 525 W, were hanging 510 mm below the ceiling.

In order to simulate the spreading of contaminants from a person, like allergens and bioeffluents, one of the person simulators was chosen to be the source of contamination.

Tracer gas was thus distributed along the whole length of the Contaminating Person Simulator (CPS). This was done through a tube, mounted on the CPS, and perforated in such a way that the gas flow was proportional to the local relative body area.

When studying the influence of physical activity, the "teacher" simulator was replaced by a real person (male, length 1.80 m, weight 70 kg, wearing a long-armed shirt and long trousers). This person performed intermittent walks in the class room, following the path in figure 2. Two different walking speeds were tested: "slow walk", at about 0.6 m/s , and "normal walk", at about 1.3 m/s. The "slow walk" was executed at 3 min intervals, each walk taking about 30 s to perform. The "normal walk", taking about 15 s to perform, was executed at the shorter 1 min intervals, as well as at the 3min intervals. Walk direction was changed after each walk.

As tracer gas, Sulphur heaxaflouride (SF_6) was used, 10 times diluted with air before released at the CPS. Measurements of gas concentrations were done in the outlet terminals and at eight different heights in the classroom in both positions C1 and C2, according to figure 2. To get an idea of the actual exposure of the "students" to the contaminant, gas concentrations were also measured in the "breathing" zone (1.15 m above floor level) of some of the person simulators.

In order to get reliable mean values of the concentrations, measurements were performed during about 1 h 15 min for each test case. When walking was performed, the walking procedure started about 1 hour before measurement start. At the end of each test the room air was mixed by means of two powerful fans, whereupon the mean gas concentration in the air was measured. The vertical air temperature gradient was measured in positions T1, T2 and T3 according to figure 2. In test cases involving physical activity, only position T2 was used.

Results and discussion

Figure 3 shows how the position of the Contaminating Person Simulator, CPS, influenced the Normalized Mean Concentration in the Room, NMCR; "normalized" meaning that the concentration is relative to the outlet concentration. (I.e. NMCR is the reciprocal to the often used "Contaminant Removal Effectiveness", or "Ventilation Effectiveness".) Obviously there was a strong influence of the CPS position, NMCR being lower the closer the CPS was sitting to the outlet terminals. With the CPS in position A – furthest away from the outlet – NMCR actually got higher than 1.0, which is the value one would get in a perfectly mixed system. This indicates that the contaminant had collected into high concentrations somewhere in the room, and some local measurements (not presented here) showed that it was in the upper half of the left hand side of the room (position C1) that the high concentrations could be found. Smoke visualization revealed that the major part of the thermal convection flow along the person simulators (PS) reached the ceiling, beneath which a net air flow towards the outlet was observed. Thus it seems that the upper zone was not at all well mixed, but that there were short cuts for the contaminants to the outlet, as well as contaminant collection in some areas, all depending on the CPS position.

Now to the influence of physical activity. Figure 4 shows how NMCR depended on the level of activity of the walking person. In all test cases involving activity the CPS was sitting in position C. All activity cases showed higher mean concentrations than when there was no

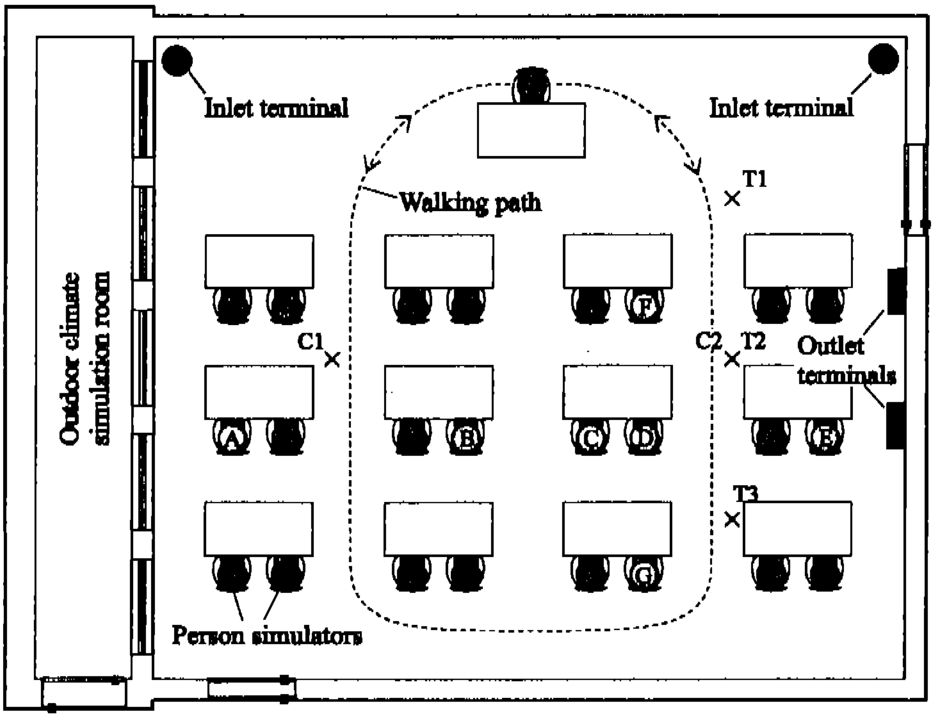

Figure 2. Sketch of the set-up in the classroom, drawn to scale.

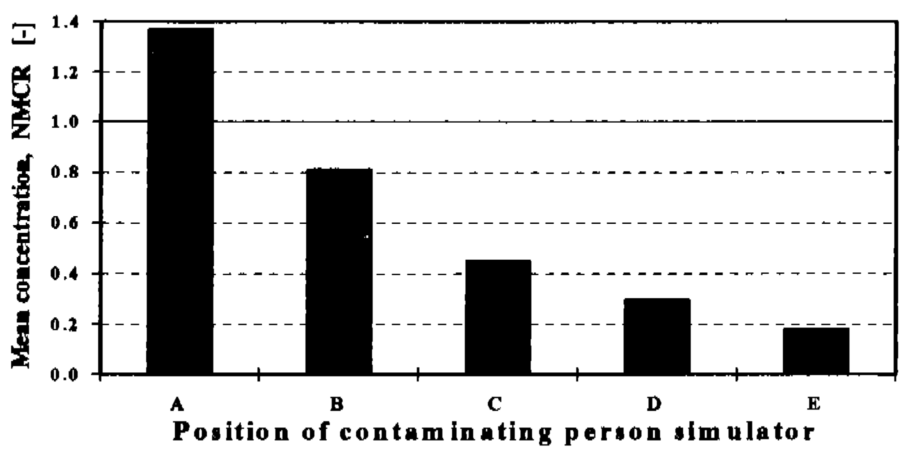

Figure 3. Normalized Mean Concentration in the Room (NMCR) for different positions of the contaminating person simulator.

activity in the room. The shorter walking intervals, 1 min, caused higher concentration, whereas the speed to perform a walk doesn't seem to mean so much for the NMCR. It is noticeable that even at the relatively high activity level of one walk around the room every minute, NMCR stayed well below 1.0, indicating a still existing displacement effect.

The vertical concentration profiles are shown in figure 5. Again the values are normalized with the outlet concentration. The reference case "perfectly mixed room air" would thus cause a 1.0-line-profile. Clearly there was a significant difference in concentration distribution between the two measuring positions, with the highest values found in position C2. This indicates substantial horizontal variances in concentration profiles within the room, again showing that the two-zone model doesn't apply very well to cases like this. The higher the activity, the more the contaminants were spread to the otherwise quite clean zone around C1, whereas around

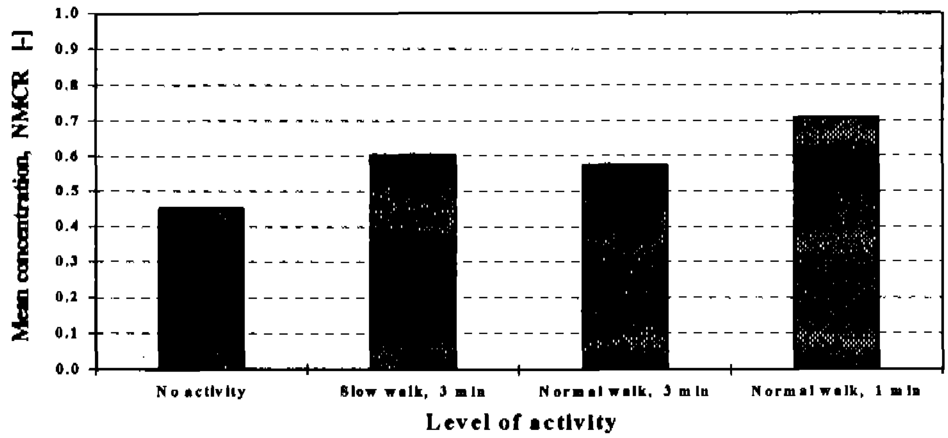

Figure 4. Normalized Mean Concentration in the Room (NMCR) for different levels of activity. Contaminant in position C. Concentrations relative to outlet.

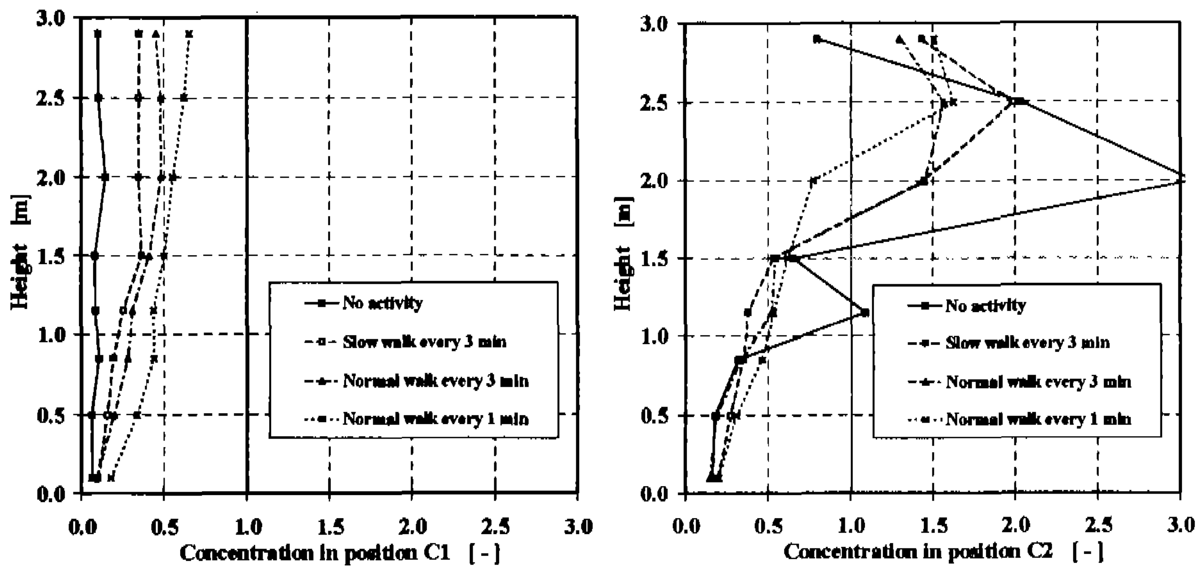

Figure 5. Vertical concentration profiles in position C1 (left) and position C2 (right) at different levels of activity. Contaminant in position C. Concentrations relative to outlet.

C2 the high concentrations in the upper part of the room decreased with increased activity.

Figure 6 presents the influence of the activity on contaminant exposure in the breathing zone of four of the person simulators. Lowest exposure was found in position A, which is in accordance with the findings in figure 5, showing low concentrations in this part of the room. All PSs were exposed to higher concentrations the more frequent the walks through the room were, but it is only in position D that we can see a marked impact of walking *speed.* All values are however well below 1.0, implying that all PSs are exposed to cleaner air than they would if the room had been equipped with mixing ventilation. That holds in fact even for the PS in position D, which was sitting next to the CPS.

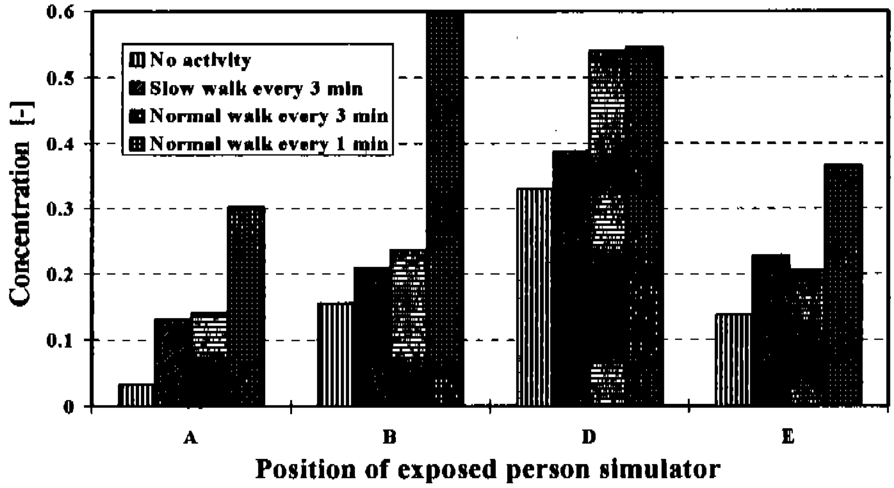

Figure 6. Exposure of person simulators to contaminant from position C at different levels of activity. Concentrations relative to outlet.

Does the supply air then reach also the persons at the back row? In figure 7 the exposure of three PSs, sitting in the same column, to contaminants from position C are given. In these positions also the mean age of air was measured, using the concentration decay method. In this case there was no physical activity in the room. Contrary of what one might expect, the PS at the back row (G) didn't get worse air than the others – it actually got better! The mean age of air differs only marginally between the three positions, meaning that it took about the same time for the supply air to reach each of the PS. However, the exposure to the contaminant was higher the closer to the inlet terminal the PS was sitting. This phenomenon can be explained by studying the air movements in the lower part of the room in more detail. The arrows in figure 7 show the flow pattern which was observed using smoke. The supply air was seen to flow rather quickly along the floor to all parts of the room. Not so much seemed to be dragged up by convection currents at the feet of the PSs, especially not at the front row, where the speed of the air was highest. The supply air then bounced against the rear wall, and flowed back towards the front wall, forming a pretty thick counter-flow. Almost all the air in this counter-flow then seemed to be captured by the convection currents along the person simulators. Most of these convection currents along the PS bodies seemed to reach the head of the PS, and then follow the plume towards the ceiling. But parts of the flow along the legs were obstructed by the table and instead seemed to join the counter-flow, as sketched in figure 7.

Thus, much – maybe the main part – of the air that reaches the breathing zone comes from the counter-flow from the rear part of the room. On its way the counter-flow collects possible contaminants from the lower part of the "student's" bodies. This line of reasoning explains the air quality values presented in figure 7. Similar concentration measurements were performed also in the column of position E, giving a similar result, although the differences were smaller.

In a separate test case, where 32 person simulators were used and provided with air corresponding to 5 l/s per person through only one displacement terminal, smoke visualization revealed that also in this case the supply air was spread over the whole floor, and bouncing at the back wall. The study by [3] also confirms that fresh air is found over the whole floor area in a displacement ventilated classroom, and that the efficiency of the system is good.

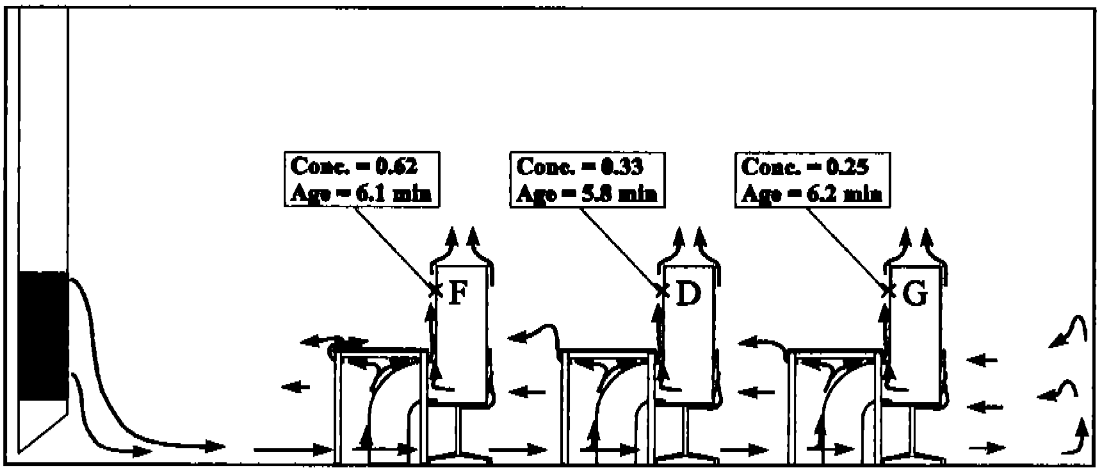

Figure 7. Sketch of the air flow pattern in the lower part of the room. Exposure in position F,D and G to contaminants from position C are given (concentrations relative to outlet). Also the mean age of air is indicated. Nominal time constant: 15.1 min.

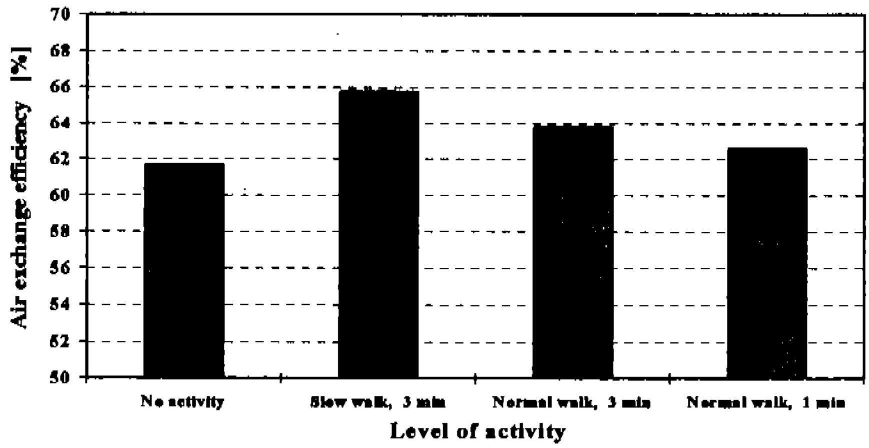

Figure 8. Air exchange efficiency for different levels of activity.

Also the air exchange efficiency in the classroom was measured (using the concentration decay method) for the four different levels of activity. The result is shown in figure8. All cases show values higher than 50 %, indicating a displacement effect. It may be a bit surprising to see that all levels of activity caused somewhat better efficiency than when there was no activity. This phenomenon, that some activity in a displacement ventilated room can cause better efficiency values, have earlier been reported in [1] and [2]. It seems like movements in the lower part of the room disturb the upwards directed convection currents along heat sources, such that those currents – of relatively "young" air – are distributed in the lower zone of the room, instead of contributing to the strong thermal plumes which mix the air in the upper zone.

The temperature gradient in the occupied zone was quite high, see figure 9. In all test cases the temperature difference between heights 0.1 and 1.1 m exceeded the ISO7730-standard value of 3.0 °C [4], warning us about thermal comfort problems. The activity tended to decrease the gradient, but only the highest activity level caused a change of any note. One should bear in mind that all measurements presented here were executed under practically steady-state conditions, and it is certainly doubtful whether this condition ever is reached in a populated classroom in practice, due to heat accumulation in materials. In reality the temperature gradient

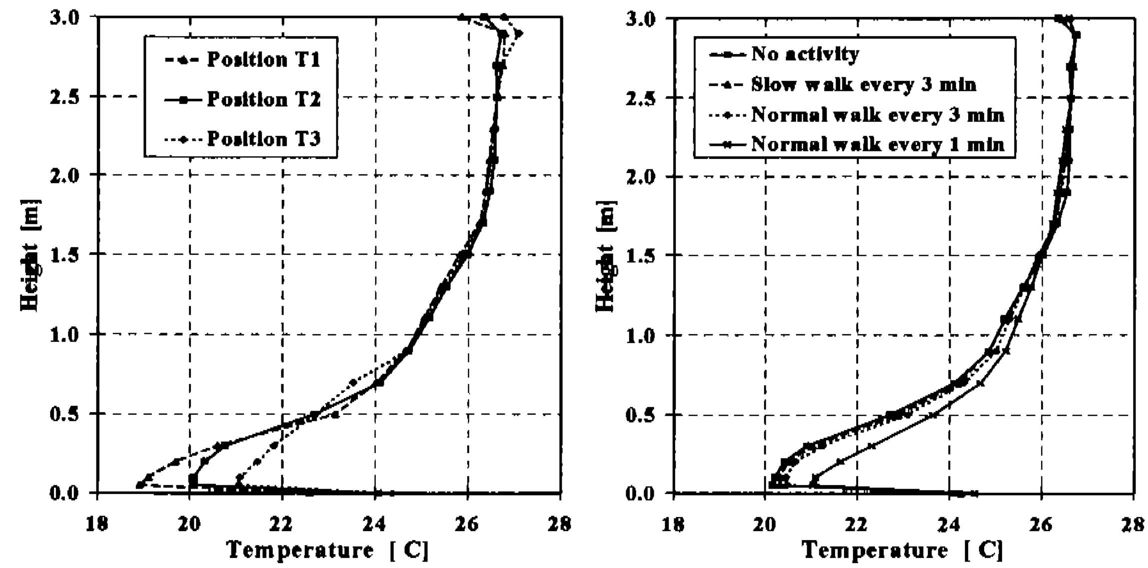

Figure 9. Temperature profiles at three positions (left) and at different activity levels (right)

might be lower than presented in figure 9, making the thermal comfort higher, but maybe also making the efficiency values of the displacement ventilation system more affected by physical activity. A dynamic case is definitely more difficult to investigate, but is an advisable subject for future research.

Conclusions

In a large, displacement ventilated and densely populated room, the spreading of contaminants from a person seems to be strongly dependent on the position of the person. The closer the contaminating person sits to the outlet terminal(s), the less of his/hers contaminants are spread in the room. People sitting furthest away from the air supply might in fact get the best air because a large portion of the air that reaches the breathing zone has first been bouncing against the back wall.

Physical activity, produced by a walking person, tends to increase the concentration of contaminants emitted from people in the room, whereas the air exchange efficiency actually can benefit from it. At all levels of activity tested in this study the displacement ventilation system provided significantly better air quality than a mixing system would.

The temperature gradient in the occupied zone was rather high during these steady-state experiments, and it was only marginally affected by the movements of a person.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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MODELLING INDOOR AIR POLLUTANT CONCENTRATIONS CONSIDERING AIR MIXING CONDITIONS

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MODELLING INDOOR AIR POLLUTANT CONCENTRATIONS CONSIDERING AIR MIXING CONDITIONS

SYNOPSIS

Modelling of indoor pollutant concentrations that varies in time can be a useful tool for estimation of the strength of internal sources and sinks. Usually the modelling has been carried out using one zone, i.e. with the assumption that the air is well mixed [1,2,3]. The present paper demonstrates that the methodology may be modified to fit multizone situations.

By studying the decay of a tracer gas, a correct model can be obtained for a specific volume in a building. For each zone in the volume, the model will include one exponential function. Once the correct model is obtained it can be used to determine the strength and the origin of a pollutant source or sink. This method will increase the agreement between the measured and calculated indoor concentration of the pollutant studied.

As pointed out in this paper, the well mixed assumption with one zone can not be used in all buildings. In a specific case, which was created in a test chamber, the total volume had to be divided into three zones, with different airchange rates to achieve a satisfactory agreement with measured data. A two zone model was sufficient to study fast variations in concentration, but to study slow changes a three zone model was needed. A similar situation, like the one created in the test chamber, may arise in buildings in operation when the supply air devices and the exhaust air devices are incorrectly installed and as a result there is a short circuit, which divides the volume into two or more zones with different airchange rates.

LIST OF SYMBOLS

$\dot{V}_{source}$	internal source strength	cm^3/s
$\dot{V}_{sink}$	internal sink effect	cm^3/s
$\dot{V}$	airflow rate	m^3/s
V	volume	m^3
$\dot{V}_i$	airflow rate in zone i	m^3/s
$\dot{V}_{tot}$	total airflow rate	m^3/s
C_E^n	exhaust air concentration at current timestep	ppm
C_E^{n+1}	exhaust air concentration at next timestep	ppm
C_S	supply air concentration	ppm
C_E	exhaust air concentration	ppm
C_{Ei}	concentration in zone i	ppm
C_{Ew}	weighted exhaust air concentration	ppm
Δt	timestep	s
j	number of zones	

1 INTRODUCTION

Analysing concentration measurements of indoor air pollutants is sometimes difficult. Even if the instruments used are correctly calibrated and the measurement locations are properly selected, the analysis may lead to erroneous conclusions. To facilitate proper analysis, with respect to quantification and location of sources and sinks of various pollutants, the measured concentrations can be interpreted by means of a mathematical model. By inserting continuously monitored outdoor air pollutant concentrations data into a model the indoor concentration vs. time can be calculated, under various assumptions about source and sink effects indoors. In this context, the air mixing within the room or building studied can be expected to be important with respect to the accuracy of the calculation.

Determination of the time constant for the ventilation of a room or a building can be done by performing an airchange rate measurement with a tracer gas. By studying the decay of the tracer gas the airchange rate and time constant can be determined for the volume. One way to determine if the building or room is divided into more than one zone with respect to airchange rate is to supply the tracer gas at a constant rate until the volume has a uniform concentration. When a constant concentration level of tracer gas is reached in the exhaust air, the supply of tracer gas is stopped and the decay will start. The shape of the decay concentration curve in a diagram will reveal whether or not there are several zones in the studied volume with different airchange rates.

Figure (1) shows an airchange rate measurement carried out in an office building using the measurement principle described above. The building is three stories high and has several rooms which are supplied by the same ventilation system.

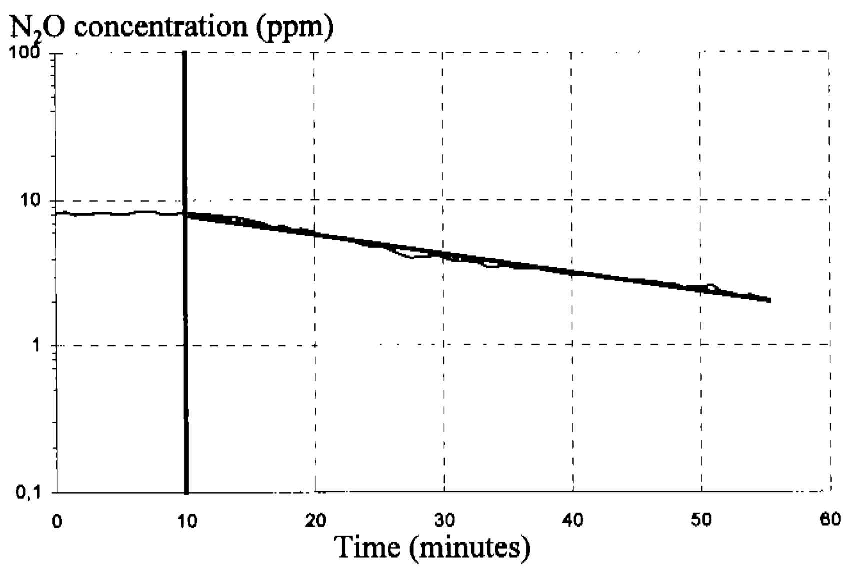

Figure 1 The decay of a tracer gas as a function of time in an office building with a total volume of 4500m^3. The tracer gas (dinitrogen oxide) was released in the central supply air duct and the concentration of tracer gas was measured in the central exhaust air duct. The supply of tracer gas ends at 10 minutes which starts the decay. The straight line in this line-log diagram indicates that the whole building has almost the same airchange rate. According to this measurement the airchange rate is 1.8 h^{-1}.

The tracer gas was released in the central supply air duct of the building and the concentration was measured in the central exhaust air duct. The supply of tracer gas was stopped when the concentration had reached a constant value. The decay is a straight line in the line-log diagram which indicates that the rooms in the office have almost the same airchange rate.

The model used here works under the well mixed assumption, but even when this is not true, the model can be modified to handle volumes not well mixed. The present paper deals with systematic errors that can occur if a multizone situation is not considered. The influence of random errors with respect to airchange rate measurements has previously been presented [4,5,6].

2 METHOD

2.1 Concentration model

The model is based on equation (1) which is a balance equation. What is supplied to the volume is equal to what is removed.

$$\dot{V} \cdot C_S + \dot{V}_{source} = \dot{V} \cdot C_E + \dot{V}_{sink} + V \cdot \frac{dC_E}{dt} \tag{1}$$

Equation (1) is valid under the assumption that the air is well mixed. The analytical solution to equation (1) is then:

$$C_E(t) = C_S + \frac{\dot{V}_{source} - \dot{V}_{sink}}{\dot{V}} + \left(C_0 - \frac{\dot{V}_{source} - \dot{V}_{sink}}{\dot{V}} - C_S \right) \cdot e^{-\frac{\dot{V}}{V} t} \tag{2}$$

where: $\quad C_0 = C_E(t = 0)$

When the air in a volume is not well mixed, the model is modified to handle as many zones as needed to make the well mixed assumption true for each of the zones. In this paper the assumption is that the air is well mixed in each zone.

To enable multizone modelling equation (3) can be used. The concentrations for all the zones are added, each weighted with respect to the airflow in each specific zone and the total airflow ratio.

$$C_{Ew} = \sum_{i=1}^{j} \frac{\dot{V}_i}{\dot{V}_{tot}} \cdot C_{Ei} \tag{3}$$

where: $\quad \dot{V}_{tot} = \sum_{i=1}^{j} \dot{V}_i$

In the context of this paper only one tracer gas is used as a pollutant and released in the supply air. The sink and source effects are never considered which simplifies the calculation.

2.2 Experimental procedure

Tracer gas is supplied at a constant rate in the supply air of the volume to be studied until a constant concentration level is reached in the exhaust air of the same volume. A constant concentration level in the exhaust air means that the concentration in the studied volume, for example a room or a building, is uniform. The supply of tracer gas is stopped and the decay starts. If present, a zone with higher airchange rate will reach a lower concentration faster than the rest of the volume. Figure (2) shows a calculation of a decay in a volume with two zones.

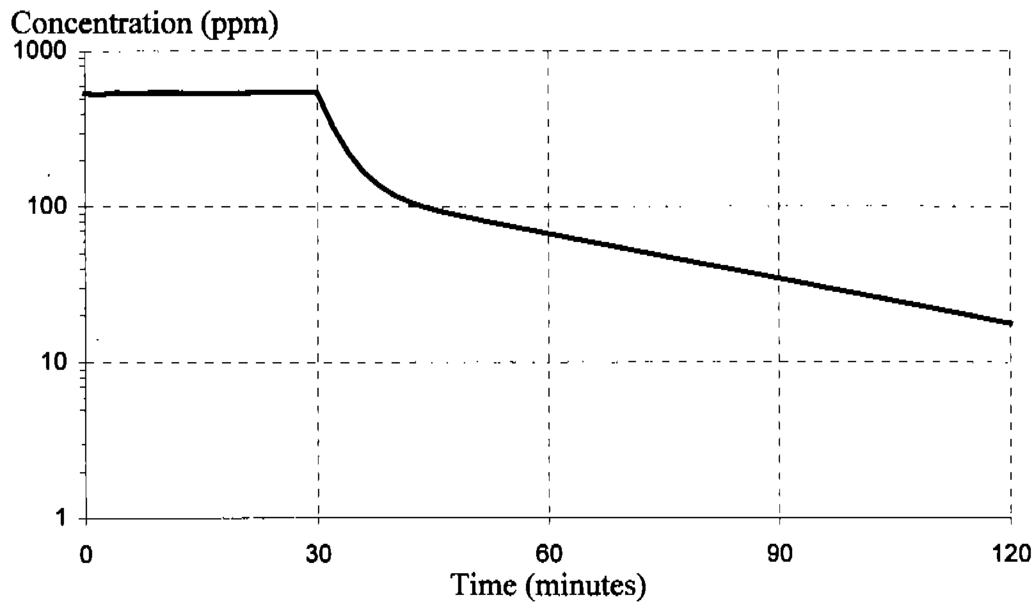

Figure 2 Calculated decay of a tracer gas as a function of time. The decay which starts at 30 minutes is calculated with two zones with different airchange rates. The smaller of the two zones has a higher airchange rate, which can be seen as the concentration of the tracer gas decreases fast for a short time in the beginning of the decay. The larger of the two zones has a lower airchange rate and the influence of that zone is dominant in the end of the decay.

A smaller part of the total volume has a higher airchange rate and the concentration decreases faster in that part, and as a result this can be seen as a short but more steep slope in the beginning of the decay curve. The end of the curve is dominated by the larger part of the volume with the lower airchange rate and as a consequence the curve is more flat towards the end. This could be the situation in a ventilated volume with a short circuit in the ventilation system.

In Figure (3) the situation is the opposite with the major part of the volume with a higher airchange rate and a smaller part with a lower airchange rate. The characteristics of the decay curve will also be different with a long steep slope in the beginning and then flattening out just in the end. This could be the situation in a volume which has a zone where the air has stagnated. With these characteristics in mind it is possible to make an estimation of the situation in a volume by just studying the decay curve.

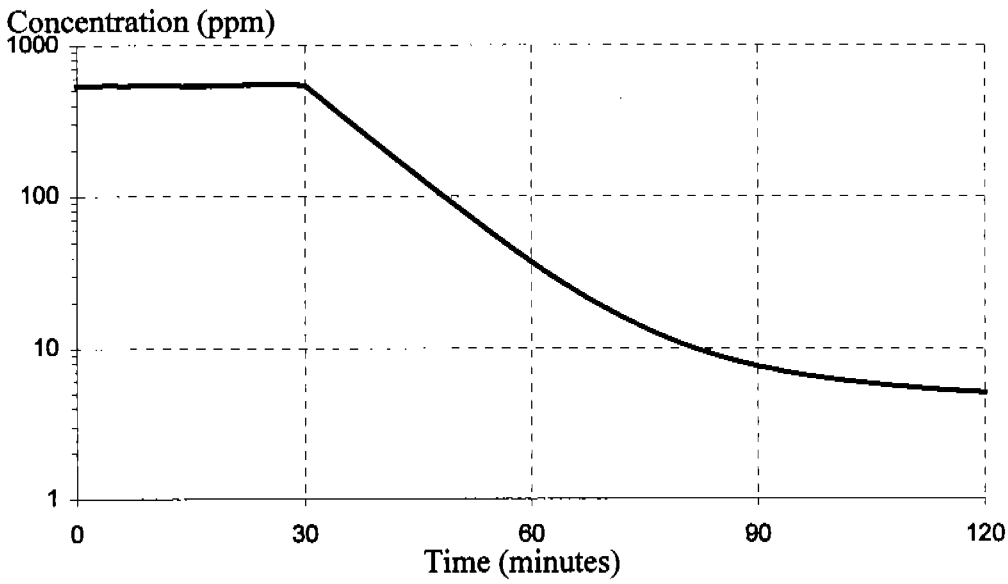

Figure 3 Calculated decay of a tracer gas as a function of time. The decay which starts at 30 minutes is calculated with two zones with different airchange rates. The larger of the two zones has a higher airchange rate. The concentration of the tracer gas decreases fast in the beginning and also for the major part of the decay. The smaller of the two zones has a lower airchange rate and the influence of that zone is dominant just in the end of the decay.

The method described will reveal if there are parts of a volume with different airchange rates. It does, however, not reveal the location of the different zones.

A test chamber with a volume of 19,3 m^3 was used to evaluate this method. The supply and exhaust air devices were located close to the ceiling. The supply air devices in the chamber were deliberately removed to reduce the airmixing function. The supply air was heated to 35 °C, which was 15 °C higher than the surrounding temperature of the test chamber, to divide the volume into zones with different temperatures and as a consequence also different airchange rates. Dinitrogen oxide was supplied at a rate of 1,7 cm^3/s and the airflow rate was 18,2 l/s. The tracer gas was supplied until the concentration in the exhaust air had reached a constant level, about 93 ppm.

3 RESULTS

Figure (4) shows the concentration of tracer gas in the test chamber as a function of time, from equilibrium is reached and two hours of the decay. A two zone model is applied in this case. The characteristics of the decay curve in figure (4) reminds of the curve in figure (2) with a smaller part of the volume with a higher airchange rate. The best result was reached when the calculation was carried out with one zone 15 m^3 large and a time constant of approximately 80 minutes and one zone 4 m^3 large and a time constant of 4 minutes. The agreement is, however, poor in the end of the decay. This suggests that the first assumption with two zones is not correct.

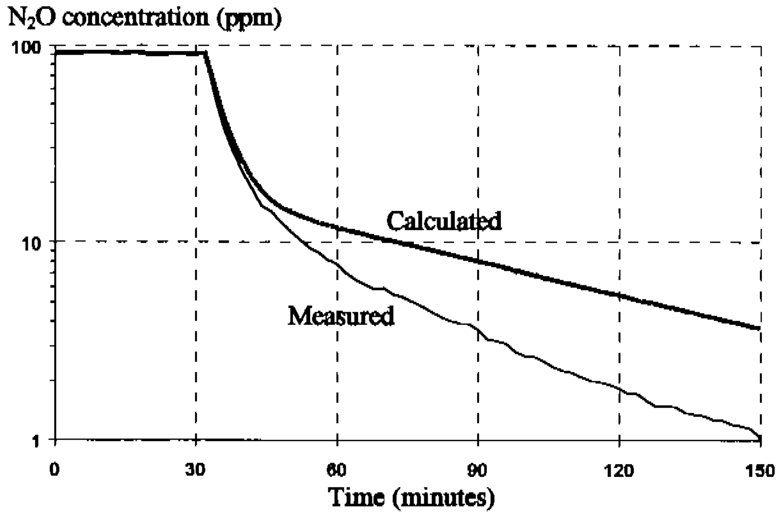

Figure 4 The measured and calculated decay of dinitrogen oxide as a function of time in a test chamber. The ventilated volume is supplied with 35 °C air and the supply air devices are removed. These conditions force the air to divide into a warmer and more highly ventilated zone and another zone with lower airchange rate. The measured concentration curve indicates a smaller zone with high airchange rate and a larger zone with a lower airchange rate. A two zone model is however inadequate to properly simulate the end of the decay where the agreement is poor.

In Figure (5) the larger zone is divided into two almost equally large zones with different time constants. One of the new two zones has a time constant of 35 minutes and the other has a time constant of 25 hours. A time constant of 25 hours is large, but this could be explained by very slow air movements in the volume in combination with an extreme supply air temperature which creates a zone with almost no airflow. The agreement between the calculated data and measured data is now good all through the decay.

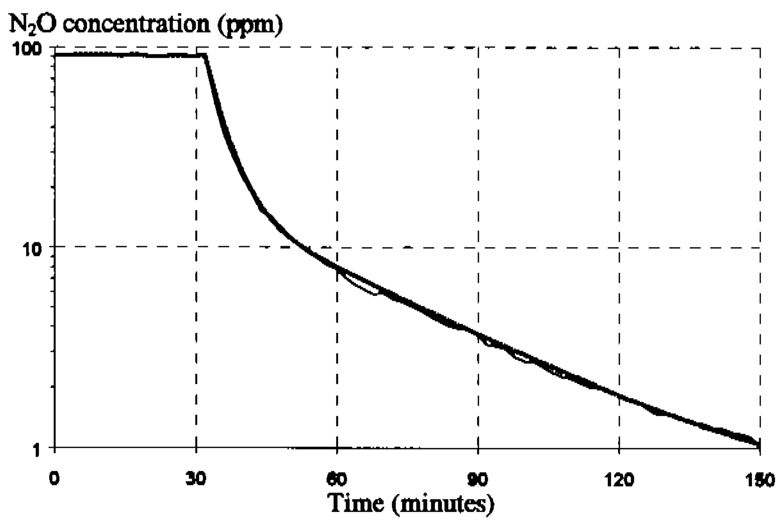

Figure 5 The measured and calculated decay of dinitrogen oxide as a function of time in a test chamber. The measured data are the same as in Figure 4. The model is extended to a three zone model to enable an adequate simulation of the decay.

In order to test the model, tracer gas was pulsed in the supply air. Short pulses of 6 and 4 minutes were used. Figure (6) shows measured and calculated values where tracer gas was pulsed for two 6 minute periods and one 4 minute period, with 6 minutes in between with no supply of tracer gas. The calculation was carried out with same model as that used to calculate the results presented in figure (4).

The agreement between measured and calculated values is good even though the calculation was carried out with a model with two zones. Since the pulses were short the zone with the large time constant could not accumulate enough tracer gas to affect the result. However, if the pulses would have been longer it would have been necessary to use the three zone model used for the calculations presented in figure (5). If the air in the whole volume is considered to be well mixed (one zone), the calculated values differ very much from the measured values which can be seen in figure (7).

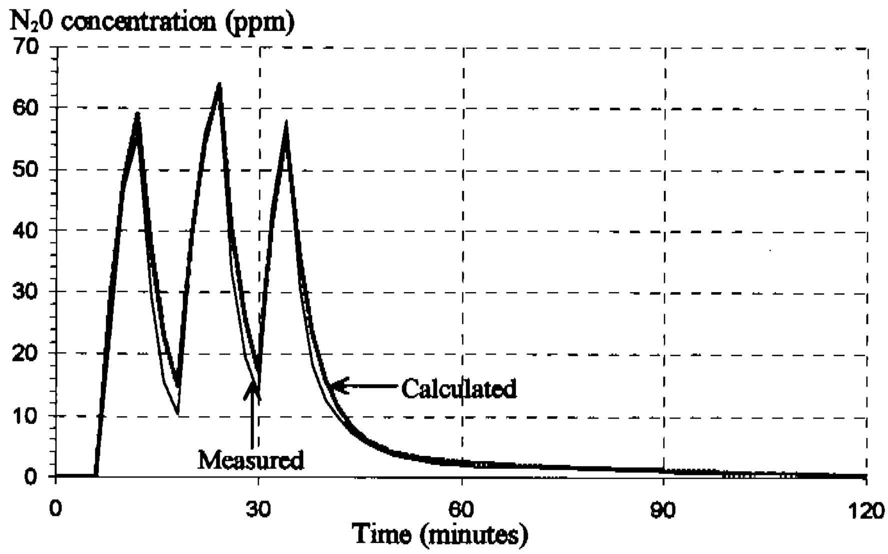

Figure 6 Concentration of dinitrogen oxide as a function of time in a test chamber. The tracer gas was pulsed into the supply air, two 6 minute pulses and one 4 minute pulse with 6 minutes in between, and the concentration was measured in the exhaust air. The calculated values were based on the same two zone model used to calculate the results presented in figure (4).

4 CONCLUSIONS

The method suggested in this paper will give useful information about the ventilation characteristics for a room or a whole building. This method will increase the agreement between the measured and calculated indoor concentration of the pollutant studied if there is a multizone situation. The method does not give any information about the location of the zones but it will reveal if different zones regarding airchange rate are present.

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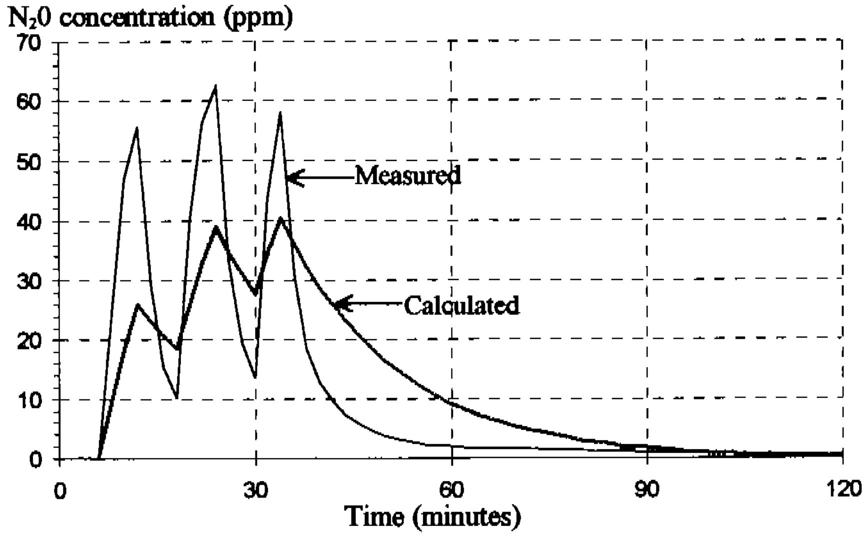

Figure 7 Concentration of dinitrogen oxide as a function of time in a test chamber. The tracer gas was pulsed into the supply air, two 6 minute pulses and one 4 minute pulse with 6 minutes in between, and the concentration was measured in the exhaust air. The calculated data was based on a model with one zone.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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THE PRINCIPLES OF A HOMOGENEOUS TRACER PULSE TECHNIQUE FOR MEASUREMENT OF VENTILATION AND AIR DISTRIBUTION IN BUILDINGS.

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ABSTRACT

The principles of a homogeneous tracer pulse technique for measurement of ventilation and air distribution in buildings.

The principles of a new tracer gas technique are described in the paper. The new technique involves pulse injection of tracer gas and has the same advantages as the previously known homogeneous emission technique. It can, for example, advantageously be used in large buildings and buildings with many rooms and yields information on the distribution of ventilation air within the building.

However, contrary to the homogeneous emission technique, yielding the average ventilation performance during an extended time, the new technique allows measurement during short term periods.

The possibility of performing quick measurement is of value for example when checking the ventilation during office hours, school hours etc. or for routine control of ventilation performance, in which cases it may be allowed to accept a result, which is representative only for the moment of investigation.

The new technique is based on homogeneous pulse injection, which means that tracer gas is injected in each zone in a zone-divided building, with amounts which are proportional to the zone volumes. The technique is therefore tentatively called the "homogeneous pulse technique".

One manner to investigate the ventilation with this new tracer gas technique, can be characterised in the following way:

Using an equipment tracer gas emission, an amount of tracer is injected into each zone in a zone-divided ventilation system, in such a way that the injected amount of tracer is proportional to the volume of the zone. Before then, integrating sampling devices for the tracer gas have been placed at those positions at which it is of interest to know the ventilation performance. The sampling devices are active until essentially all tracer gas has disappeared from the system.

The amount of tracer compound collected with a sampler is proportional to the local mean age of air at the point of sampling.

Theoretical and practical aspects of the technique are described.

Please Note: The full paper will be printed in the Supplement to the Proceedings

VENTILATION TECHNOLOGIES IN URBAN AREAS

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NON ATTENDANCE RATES AMONG CHILDREN IN SWEDISH DAY-CARE CENTRES BEFORE, DURING AND AFTER CLEANING THE INDOOR AIR USING AN ELECTROSTATIC AIR CLEANING TECHNOLOGY – A CONTROLLED TRIAL

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ABSTRACT
To conduct a controlled trial to test the ability of a newly developed electrostatic air cleaning technology (EAC) to improve Indoor Air Quality (IAQ) as defined by levels of air borne particles and to investigate the potential to reduce non-attendance rates due to illness among children in two Swedish day care centres. The EAC technology was shown to significantly reduce the indoor particulate load for very fine particles (>0.3μm) caused by outdoor air pollution by 78% and to reduce the number of particles (>3.0μm) produced indoors by 45%.
Non-attendance was followed for two "treated" centres and two control centres during three years. The EAC technology was in operation during year two. Non-attendance rates among children in the larger day-care centre decreased by 55%, equalling those noted in family based day care.
The EAC technology is cost efficient and might be a way forward to improve IAQ.

INTRODUCTION:
Indoor air quality (IAQ) is a complex function of outdoor air quality, indoor activities past and present, design of ventilation systems, number of air changes per min., building design/size and emissions from the building materials.

Recently, fine particulate matter generated by the combustion process and the diesel engine in particular, has come to the fore as a potential cause of respiratory symptoms among those children and adults suffering from chronic respiratory disorders[1,2] but also as an adjuvant for the development of allergy[3].
In a questionnaire based study covering 39 Swedish schools, Norbäck and Smedje reported on a positive relationship between respirable dust generated indoors and airway infections in adults as well as between viable airborne bacteria and moulds and asthma in children[4].
Upper respiratory tract infections (URTI) are two to three times more common, as is the use of antibiotics, among children attending day-care centres where most of their time is spent indoors[5,6].

In Sweden increased forced air ventilation rates have been tried over many years as a method to improve indoor air quality, in public and private buildings. In 1994, Sweden set a new standard for IAQ, based on a maximum carbon dioxide concentration of 1000 PPM in an attempt to further control the IAQ issue[7]. Surprisingly few data are available to prove how effective the approach has been[4].

Electrostatic mechanisms provide an alternative means to control the movement of fine air borne particles[8]. One way of generating electrostatic fields in a room, is to produce free electrons in the air. Some of these electrons will combine with oxygen and a negatively charged small air ion is produced. There is empirical evidence that such charged air can reduce the growth of micro-organisms[9]. This observation has been further strengthened by the observation that small amounts of hydrogen peroxide are produced with increasing levels of negative air ions[10].
Thus, the delivery of free electrons into the indoor air has the potential to enhance the air quality by reducing the number of airborne particles through electrostatic 'filtering' mechanisms and via the hydrogen peroxide mechanism reduce the growth of micro-organisms[11].

HYPOTHESIS:
Does the production of free electrons into the indoor air have the ability to reduce the number of air borne particles of a defined size in a busy children's day care centre?
Would the potential improvement in IAQ from such a system, reduce the non-attendance rate due to sickness among the children in day care centres?

To evaluate these hypotheses, an electron producing device (Electrostatic Air Cleaning, EAC - system) was constructed and installed in two Swedish day care centres. The non attendance rates among the children were recorded over a three year period. The concentration of fine (> 3μm) and very fine (> 0,3μm) air borne particulate matter was recorded. The number of absent children was compared with day-care centres of similar size and design without the EAC technology.

METHODOLOGY:
Although the EAC system is not regarded as a medical device, it's use in children's day care centres was approved by the Ethics committee, Faculty of Medicine, University of Gothenburg, Sweden. Parents were given written and direct information at meetings.

Two day care centres, A and B, were equipped with EAC-systems. Centre A was built 1975 with a large group of children (63) whereas centre B was located in a modern building, built 1991, with half as many children using the premises on a daily basis. Control centres A_{ref} and B_{ref} were chosen on the basis of size, locality and age. The control centres were both located within less then 1.5 km of the corresponding EAC equipped units and covered the same residential area of the town. All buildings had controlled forced air ventilation that fulfilled the standards required. No other changes were undertaken in the four day-care centres during the 3 year trial.

The local Social Services office register and collate figures for non-attendance among pre-school children indicating reasons for the absence. The non-attendance rates due to illness used for this research were taken from this database.
Comparisons of non-attendance rates were made over a three year period with year two being the year of 'EAC'-treatment in centres A and B. centre B_{ref} did not start to operate until august 1993, therefore the period included in the three year analysis of centres B and B_{ref} has been restricted to 8 months (1 Aug. 1993 - 13 March 1994).

THE EAC-SYSTEM:

The EAC- system delivers a high voltage (7 kV negative polarity), DC current (< 0,5 mA) to a carbon fibre thread (the emittor) positioned close to each ceiling mounted forced air inlet. The number of small air ions produced was regularly measured using an atmospheric ion analyser (Medion type 134A). EAC - systems were only installed in rooms used by the children in the day care centres. Throughout the time of the study negative air ion levels of 20.000-40.000 per cm^3, at a height of 1m above the floor were recorded. A negatively charged electrostatic field of -30 kVm was recorded by a standard DC electrical field recorder (Eltex Q475C), at a distance of 30 cm from the emitters. The field strength one meter from the emittor was -15 kVm which is equivalent to the field strength of a TV set (positive electrostatic field). The walls of the rooms became slightly negatively charged (1.5 - 2.0 kVm) compared with a zero or slightly positive charge in a standard room.

The EAC systems were in operation throughout the second year from the first week in April 1994 to the first week in April 1995. They were then turned off with the equipment left in place throughout the third year.

MEASUREMENTS OF AIR BORNE PARTICLES:

A MET-ONE model 2110 (Met-One, Oregon, USA) laser beam particle counter was used to record the number of particles per litre of ambient air.

The particle counter was set to measure, particles > 0,3 and >3,0 μm in size. Comparisons were made intermittently between indoor and outdoor particle counts. Indoor particles counts were recorded over 24-hour periods. Measurements were taken during a 30 second period every 5 minutes. Particle counts were made in one playroom at a height of 1.2 m and at a distance of three meters away from the forced air inlet.

Figure 1

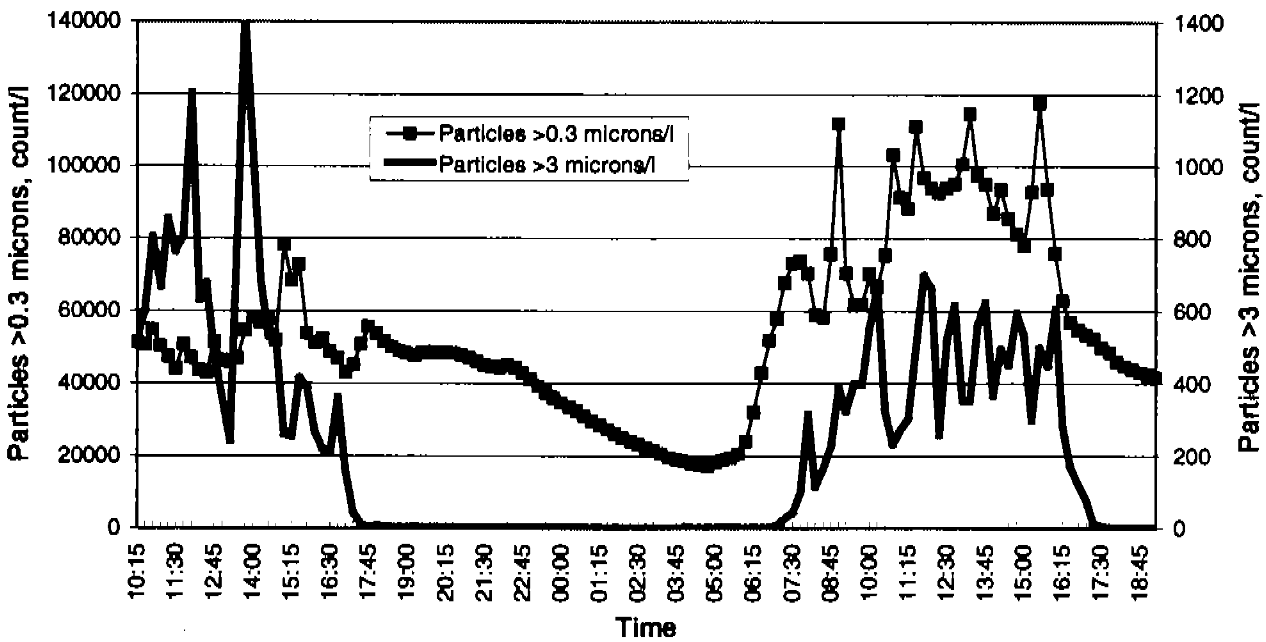


Figure 1 gives an example of a 24-hour recording of how the number of particles varies in a play room, depending on the level of activity in the room. This was most pronounced for particles >3

μm. The number of these particles dropped to zero during the night, increasing again as staff entered the room in the morning. The number of very fine airborne particles also increased in the morning when the ventilation system was switched on, prior to the arrival of the staff.
Thus, the particles measured represented:

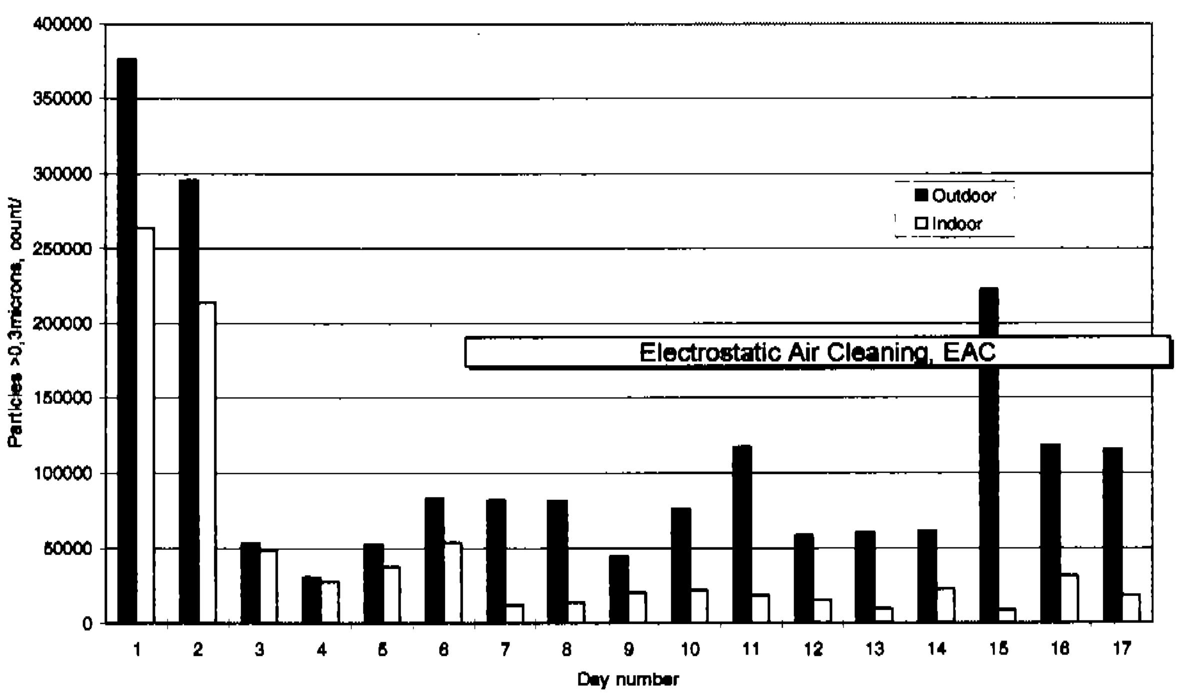
a) Very fine particles, those > 0.3μm, entering the room from the outside air through the ventilation ducts. The relationship between in- and outdoor concentrations was used to quantify IAQ.
b) Particles of a size >3.0μm generated from activities within the room. The average reading recorded during office hours, 08:00 - 15:00 was used to quantify IAQ.

The carbon fibre threads were vacuum-cleaned every third month to ensure their function..
Statistical analysis of the data was performed using two - tailed, paired students T-test.

RESULTS:

The outdoor air was always found to have a higher concentration of particles > 0.3 μm, than the indoor air. This is illustrated in Figure 2 showing parallel in- and outdoor measurements with and without the EAC-system in use. On average, a 25 % reduction of particles > 0.3μm was noted

Figure 2



under normal conditions as the air passed through the existing ventilation system and settled within the room. This difference was markedly enhanced when the EAC-system was in operation showing, on average a 78 % reduction of particles > 0.3 μm ($p < 0.001$).

The average daily count of particles >3.0 μm was recorded on ten occasions, four without and six with EAC. A significant reduction was noted with the EAC-system as the daily averages decreased from 428 (median, range: 340 - 649) particles per litre of air to 232 (range: 166 - 287), p <0.01.

Figure 3

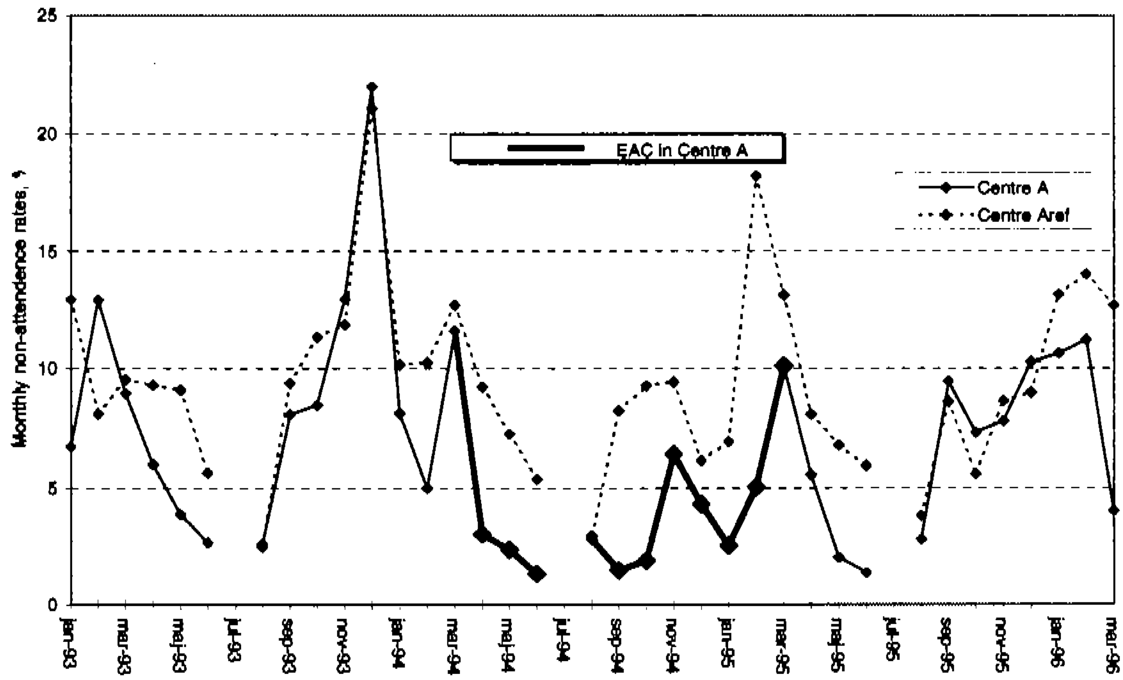


Figure 3 shows the monthly figures for non-attendance rates due to sickness comparing the two larger day care centres A and A_{ref}. The two centres followed a similar pattern during year 1 and 3 whereas during year 2 when the EAC-system was in operation centre A consistently showed lower non attendance figures than centre A_{ref}. (The graphs are disjointed because of summer vacations.)

Table I

Non-attendance due to sickness, annual rates (%)

	Centre A		Centre A_{ref}	Centre B		Centre B_{ref}
1993-94	8.31	**	10.31	9.20	*	5.46
	**					
1994-95, EAC year	**3.75**	***	8.75	**6.09**		6.76
	*					
1995-96	7.94	*	8.76	5.92	**	9.21

* = p<0.05, ** = p<0.01, *** = p<0.001, paired T-test

Table I gives a comparison of absenteeism during the three year period in the four centres. Centre A had a significant reduction in absenteeism from 8.31% to 3.75% returning to 7.94% during the third year. It appeared as if centre A was significantly healthier than centre A_{ref} with 19 and 9% less sick children year 1 and 3, respectively. This difference became highly significant with 57% less sick children in centre A during the EAC year.
Table I also gives the non-attendance rates for the smaller and more modern day-care centres B and B_{ref}. centre B showed a decrease by 33% as compared with an increase in non-attendance by 23% in centre B_{ref} comparing year 1 and 2. These differences did not reach statistical significance. Note the increase in non-attendance in the newly built centre B_{ref} which became significant during the third year ($p<0.05$).

When the EAC system was turned off the staff in centre B complained of the stuffiness of the indoor air and had the ventilation system checked. The system was operating according to specifications. The only side effect noted during the EAC year was an accumulation of dirt around the emitters. This was markedly reduced by placing a metal sheet between the emitting thread and the ceiling. More dirt was noted when cleaning the floors on a daily basis. The parents also noted that the children's socks became more dirty during the EAC year.

DISCUSSION:
The aim of the study was to conduct a controlled trial to test the ability of a newly developed electrostatic air cleaning device to improve IAQ as defined by levels of air borne particles and to investigate the potential to reduce non-attendance rates among children in day care centres. These are known for an almost three-fold increase in non-attendance, primarily due to viral URTI which is related to the number of children[6] and possibly the load of biologically active air borne particles.

In the larger centre repeated measurement were undertaken in order to demonstrate effects on the number of air borne particulate matter. The non attendance rate due to illness was provided from the records on absenteeism kept by the Social Services administration. This independent data collection together with the unlikeness that the children *per se* would alter their behaviour due to some equipment being mounted in the ceiling, should reduce the methodological error. This risk was further reduced by leaving the equipment mounted after it was turned off. Furthermore by including data obtained on a yearly basis short term trends due to seasonal variation in URTI can be excluded.

It was obvious that the EAC - system altered the pattern of dirt deposition with more dirt deposited on the floor and around the emitters. It appears logical to assume that the very fine particles generated outdoors, and reduced by 78%, got trapped as the air entered the room and passed close to the EAC system, the site where the negative electrostatic field was the strongest. The larger size particles generated by the activity within the room became less airborne (45% reduction) either by not leaving their source (humans or horizontal surfaces) so easily due to the alteration of the electrostatic field within the room and/or being captured by the strong electrostatic field operating close to the EAC emitters. It took approximately two weeks for the walls to obtain a slight negative electrostatic charge as compared with the overall positive charge noted initially. Not until this was achieved did the reduction in particles become maximal, indicating that the negative electrostatic field effect is important.

IAQ and its impact on the indoor environment is not only a function of the concentration of air borne particles. Equally relevant is the potential biological activity of these particles[13]. This bioload concept includes fine respirable particles generated by micro-organisms. Our own experimental work on enhanced negative air ionisation has demonstrated the generation of hydrogen peroxide in the range of 0.7 to 1 μM at 20-50 000 negative air ions per ml of air[10]. Hyslop and collaborators recently reported on hydrogen peroxide as a potent antibiotic[11]. They showed a bacteriostatic effect at 25μM without any signs of affecting the growth of human fibroblasts. To what extent a hydrogen peroxide concentration of 1μM operating over time would affect the growth of micro-organisms remains to be tested. However, own observational data has indicated a marked decrease in air borne moulds in rooms after two to five months of EAC treatment.

To our knowledge no previous attempt has been made to study interventional procedures and their capacity to improve indoor quality, relating the effects on the non attendance rate among children. Hawkins, in a previous controlled trial on negative air ionisation showed positive effects on subjective parameters such as headaches etc[12]. Such observational studies need to be substantiated by more detailed research into possible mechanisms. In the current study a substantial reduction of indoor air particles was achieved by altering of the electrostatic fields within the rooms. The impact of this on non-attendance rates among children in the larger day-care centre was most striking with a 55% decrease and non-attendance rates equalling those noted in family based day care[6].

Un-expectantly, centre B_{ref} which was established in a new building in August 1993, showed a significant increase in the non-attendance rate from 5.46 to 9.21% ($p < 0.05$) during the three year period. Perhaps the biological history of a building and its accumulated bioload should also be considered when assessing the state of a building from a health perspective.
Whatever the complexity of factors affecting the indoor environment, it appears as if electrons released into the room thereby generating a weak negative electrostatic field and an increased level of negative air ionisation could significantly enhance IAQ with a potential to reduce URTI among children attending large day care centres.

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The authors are indebted to Erland Holmdal, Bia Håkanson-Michalski, Lena Johansson and the municipality of Uddevalla, Sweden for financial support and help with data collection, to Rickard Bergström, Ph.D. Department of Environmental and Occupational Medicine, University of Göteborg for guidance on IAQ issues and to John Curnow, Department of Biomedical Engineering and Medical Physics for help with the electrical science. Financial support was also provided from a Regional Enterprise Grant no ERG (PL) 151, The Government Office For The South West, UK.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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Title: On the Time-Dependent Efficiency of Building Ventilation on the Indoor Air Quality in a Medium Sized Urban Area in Greece

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ON THE TIME-DEPENDENT EFFICIENCY OF BUILDING VENTILATION ON THE INDOOR AIR QUALITY IN A MEDIUM SIZED URBAN AREA IN GREECE

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Synopsis

From an air pollution study in a medium-sized, seaside town in Central Greece (Volos) it was found that some common air pollutants (CO, NO, NOx, SO_2, O_3), whose emissions are connected to activities and conditions that reveal some characteristics of periodicity on a daily, weekly or yearly basis (e.g.: production activities, meteorological conditions), are monitored in the atmosphere in concentrations that reflect this periodicity. Additionally, characteristics of periodicity can also be found in the concentrations of indoor gas pollutants whose emissions are connected to some inhabitants' activities (e.g.: cooking) or are influenced by the microclimatic conditions in the building interior (e.g.: temperature, humidity). By processing this data, by statistical methods, periods were identified during the day, week or year when it is feasible to predict that indoor or outdoor air is more contaminated and harmful and, accordingly, to appreciate the relative contribution of building ventilation to the formation of indoor air quality. Corresponding findings, especially those related to periodicity on a daily basis, are used for the judgement and corroboration of the measures that are taken for the protection of indoor air quality during episodes of environmental air pollution and for the elaboration of proposals for the design of natural ventilation systems for the buildings in the area.

1. Introduction

Building ventilation makes the indoor air quality unavoidably associated with the environmental air quality. Nowadays, due to extensive and intensive pollution, there are areas on the planet where we encounter considerably increased concentrations of air pollutants. Such areas can be found in metropolitan city centres and in industrial zones. In such areas, ventilation can no longer correspond to its traditional role. In similar conditions, a more sophisticated and complex treatment of the building's air change systems is required, and possibly the implementation of additional measures that will guarantee more comfortable and hygienic conditions for the inhabitants. So, for example, in extreme conditions of environmental pollution, the installation of systems to remove the elements of pollution from the air is required. Such systems, beyond the cost of installation, operation and maintenance, also burden the building's energy balance as they presuppose the existence of mechanical ventilation systems. Instead, in moderate conditions of environmental pollution, it is possible that the natural ventilation systems are able, by themselves, to minimise the problem by the application of appropriate measures and techniques. A precondition for this is that the design of the system be made according to the actual climatic and environmental conditions of the area. Additionally, in order to make the applied measures and techniques more efficient, they should be considered and studied carefully during the design phase of the corresponding systems.

In Greece, there are areas where episodes of serious environmental pollution occur relatively frequently. They are mainly in the centres of large cities and in areas with intense industrial activity. However, a large proportion of the population lives and work in buildings which are located in areas with only moderate environmental problems. They are generally the non-central districts in large cities and medium and small-sized urban areas. A representative example of this category is the town of Volos, a medium-sized urban area, with a population of about 100,000, which extends to the cove of the Pagasitikos Gulf in the Eastern part of Central Greece.

In this paper, the conditions of environmental pollution in the area of Volos are examined, as a factor influencing the indoor air quality in the residential and commercial buildings of the area. The objective of the paper is the judgement and corroboration of the techniques and measures which are traditionally applied, or which could be applied for the ventilation of the buildings in the area in order to limit the adverse effects of environmental pollution in their interior.

2. The climatic and environmental conditions in the area

The climate in the area of Volos, like in the greater part of Greece, is Mediterranean type (Tselepidaki, 1994). More specifically, the annual period can, in general, be divided into the climatically cold and humid season (October - March) on the one hand, and the hot and dry season (April - September) on the other (Figure 1). From September to October there is a significant drop in temperature (~ 5° C) which continues gradually until January, which is the

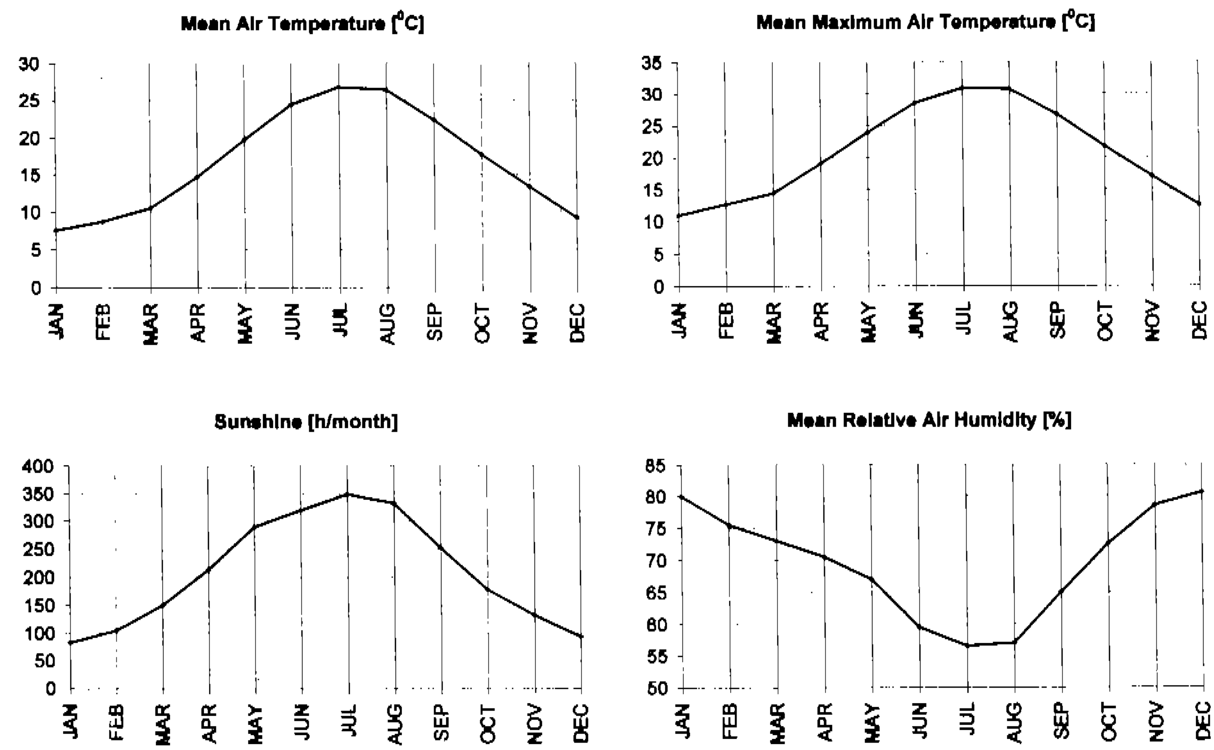

Figure 1: The mean monthly values of some basic climatic parameters in the area of Volos.

coldest month of the year (mean temperature: 7.6 °C). Starting from March, the temperature begins to increase until July and August, which are the hottest months of the year (mean temperatures: 26.8 °C and 26.4 °C, respectively). The intermediate seasons (autumn and spring) are quite clearly delineated and, in particular, the autumn is hotter than the spring by about 2 - 4 °C. Through the area passes the mean annual isotherm of 17.5°C. The mean extreme temperatures are approximately 3°C in winter and 31°C in summer. The mean annual Degree Days in the area is about 1506.

The wind's climatic characteristics, as determined by both the general atmospheric circulation and the prevailing synoptic systems in the wider area, contribute to the predominance of west and north components and moderate speeds (Zabakas, 1981). Nevertheless, in combination with these factors, the landscape, as a factor of canalisation and mid scale thermal circulation, plays a dominant role in the determination of the direction and speed of the prevailing wind in the area (Papamanolis *et al.*, 1996). On a smaller scale, the characteristics of the wind at every point in the city are influenced, in a complex way, by the street layout and the shape and dimensions of the surrounding natural and artificial obstacles (e.g.: trees, houses) (Kobysheva, 1992).

The area, due to the problems encountered in many Greek cities (lack of land planning, unplanned urban development) but also due to the increase in productive activities (industries of all sizes, transport, the port), has displayed problems of atmospheric pollution in recent years. Such problems, while they do not seem particularly serious, must not be overlooked. In a recent study which was based on the recordings from the local air pollution station of the concentration of basic pollutants for the period 1987-1994, their mean annual concentration values were calculated as follows: CO = 1.7 mg/m^3, O_3 = 73.3 ìg/m^3, NO_2 = 58.5 ìg/m^3, NO = 55.8 ìg/m^3, SO_2 = 63.5 ìg/m^3 (Dalezios *et al.*, 1995). From the same study, it becomes apparent that a large proportion of the responsibility for the air pollution in the area can be put down to traffic pollution. Moreover, a significant proportion seems to be the result of burning, both for the heating of buildings during the cold period and for productive activities (factories, industry). An influence on the environmental conditions of the area is exerted by the wind, which, depending on its direction and its speed, contributes either positively or negatively to the build-up of pollution (Papamanolis *et al.*, 1996), as does the sea breeze during the summer period (Papamanolis and Dalezios, 1997).

In some of the above studies, certain characteristics of periodicity in the concentrations of air pollution in the region have been identified. These features have been explained by the periodicity of human activities which are responsible for the emissions, and the meteorological phenomena which affect them. So, on a daily basis, higher levels of the accumulation of pollution can be observed (with the exception of O_3) during the hours where traffic circulation is heaviest due to rush-hour movement. Similarly, on a weekly basis, the accumulation of pollution is higher on the working days of the week and especially on days when shops and businesses are open both in the morning and in the evening. Periodicity was also detected on a yearly basis: the pollutants which are related to burning (CO, NO_X, SO_2) show high concentrations during the winter, when the heating requirements of buildings are at their highest. Ozone, as a product of photochemical reactions, was found to exhibit a special behaviour, with clear indications of periodicity on a daily, weekly and yearly basis, but with extreme values, which are generally in contrast to those of the remaining air pollutants.

These findings concerning the periodicity of mean and extreme values of the concentration of pollutants which are derived using Descriptive Time Series Analysis methods, were verified by the Fourier Transformation processing of the hourly concentration values of the corresponding gas pollutants for the period 1991 - 1994. As a result of this process, the seasonal indices of the relative, to their mean annual, air pollutants concentrations, over the 24-hour period, are shown separately, for the hot and the cold periods of the year, in the diagrams in figure 2.

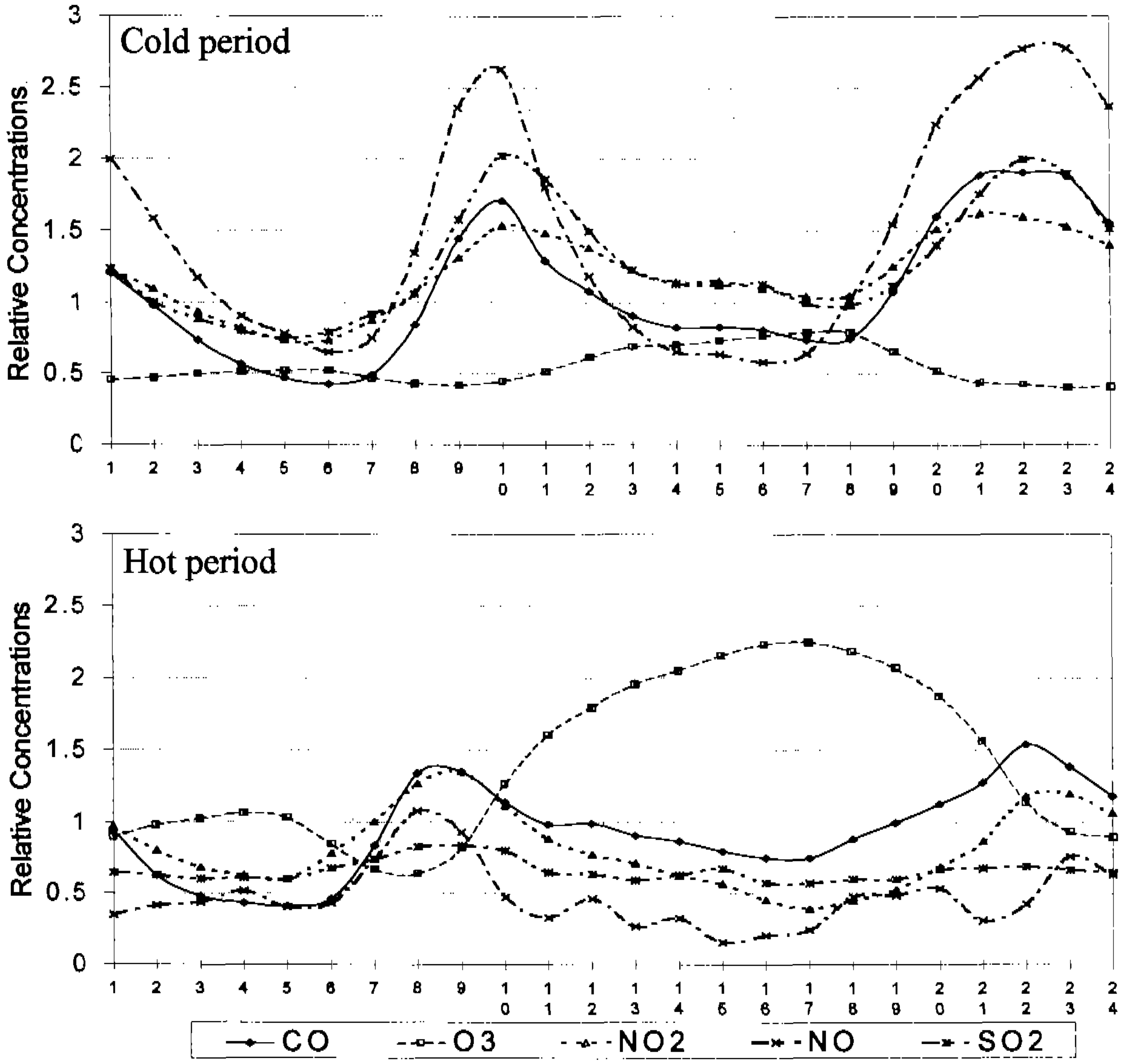

Figure 2: Seasonal indices of the relative, to their mean annual, air pollutants concentrations in the area of Volos, over the 24-hour period (based on hourly concentration values for the period 1991-94)

These diagrams show clearly the intervals during the 24-hour period when, with periodicity, increased concentrations of air pollutants can be observed. These particular pollutants, even when encountered in concentrations below their standard threshold values, have harmful effects on human health (Namiesnik *et al.*, 1992). Moreover, their presence in the atmosphere, and the fact that they were found to be largely due to traffic pollution, is unavoidably connected to the presence of other pollutants - which can not be measured with the equipment available - but also to high noise levels. Consequently, it is crucial over the corresponding periods to protect the internal environment of the buildings in the area from the influence imposed by the external environment primarily through ventilation processes. Buildings close to roads with heavy traffic are obviously more vulnerable. Especially, their lower floors, which are closer to the street level, are exposed to pollutant concentrations much higher than those recorded at the station located on the terrace of a five-store building.

3. The building ventilation conditions and the indoor pollutants in the area

In Greece, as in most countries in mild and moderate climates, the majority of residential and commercial buildings use natural ventilation systems. To a large extent, the air change takes place via infiltration and, when that is not sufficient, the residents open their windows. Furthermore, in certain spaces - the so-called 'wet' rooms (kitchen, bathroom, WC) - an exhaust fan is also often installed which, when required, is operated at the discretion of the residents.

A large number of the buildings in Volos, especially in the centre of the town, were built in the late 1950's within the framework of an extensive programme of rebuilding following a series of catastrophic earthquakes. As a consequence, there is a similarity in the architectural and constructional characteristics of these buildings. These characteristics, which include for example the method of construction, the building materials, the dimensions of the rooms, the dimensions and the type of openings, the roofing, etc. in the degree that are comparable to those prevailing in other urban centres in Greece, lead to similar conditions of natural ventilation in the buildings. From the studies which relate to the ventilation conditions of conventional building constructions in Greece, although not supported by statistically reliable samples of measurements under normal conditions, they can be classified (at least from a quantitative point of view) in the same group as corresponding buildings in a wide range of countries (mainly in Europe and North America) which offer richer bases of similar data (Papamanolis and Koinakis, 1996; Papamanolis, 1998). In other words, for an ordinary building, under meteorological conditions which are normal for its location, the range 0.5 to 1.0 ac/h constitutes a reasonable first approximation for its air change rate. Yet, such air change rates in residential and commercial buildings are frequently connected to indoor air quality problems (CEC, 1992).

The studies which refer to the quality of the internal environment in Greece are limited (e.g.: Lagoudi *et al.*, 1996). For the commercial and residential buildings in the area of Volos, as also in other urban centres in Greece, the fact that gas is not really used and that instead central heating systems and electrical appliances (for heating, cooking, etc.) are favoured, limits the range and the quantity of unavoidable pollutants inside them. Therefore, it is reasonable to assume that the main pollutants are those associated with metabolism (carbon dioxide and odour) as well as moisture (from cooking and washing) and odour (from cooking). Besides these, there are always pollutants present which are emitted from building materials and furniture as well as from smoking, if applicable (Namiesnik *et al.*, 1992). Some of the pollutants which are the result of the activities of the residents can be monitored, showing characteristics of periodicity in the interior of many buildings. The most obvious pollutants in this category are those which are related to the presence of the residents themselves in the buildings and to cooking, which, as is reasonable, takes place prior to eating times. As far as the emissions which are not related directly to the activities of the residents are concerned, it is possible that, to the extent that they are influenced by the micro-climatic conditions inside the building (e.g.: temperature, humidity), they are also subject to periodicity (Namiesnik *et al.*, 1992). However, considering the complexity of the procedure concerned and the lack of relevant data, no clear predictions can be made.

4. The effect of building ventilation on the indoor air quality in the area

Indoor pollutants are derived from both indoor and outdoor sources. Each of these sources tends to impose different demands on the control strategies needed to secure good health and comfortable conditions. As regards the pollutants that are emitted in the interior of the buildings in the area, it seems that, based on the appreciation of their qualitative and quantitative characteristics, they do not cause any particular problem. The episodes of indoor pollution seem to be generally recognisable (part of the pollutants are odours) and easily solvable (by opening windows and with the operation of exhaust fans).

On the contrary, the assumed problems of indoor air quality in the area seem to be more closely related to environmental pollution. More specifically, during periods of increased concentrations of outdoor pollutants, the effectiveness of ventilation, as a process of supplying clean air into the building's interior, is devaluated or even reversed. This particular problem can not be confronted either by filtering the incoming air or by cleaning the indoor air, since natural ventilation systems, which prevail in the buildings in the area, are not compatible with the operation of corresponding systems. Therefore, the only way to confront this problem is by controlling the ventilation openings in the building shell. This involves closing windows (during the hot period) or keeping them closed (during the cold period) when high levels of pollution are recorded in the atmosphere. This measure, however, besides the possible limitations which are involved (e.g.: in ventilation cooling), also varies in effectiveness as the change in the quantity of ventilation provided depends on the intensity of the driving natural forces i.e.: wind and indoor/outdoor temperature difference as well as the dimensions, the type and the distribution of the remaining openings, including the infiltration openings. Moreover, this measure, which by nature is applicable for only a short period of time for inhabited places, is subject to further limitations of duration because of the generally small dimensions of the internal spaces in buildings in Greece, which means that they have only a limited ability to operate as fresh air reservoirs. The measure's applicability and effectiveness is very depended to indoor pollutant emissions (e.g.: cooking, smoking). Therefore, the reduction in the internal pollution for the corresponding period means an extension of the time it is applicable and beneficial.

According to this data, the identification of the main sources of emission of air pollutants and of the characteristics of periodicity in the build-up of air pollution in the area offers limited capability for the application of direct measures for the protection of the quality of the internal

air. This fact, however, to the extent where it is caused by the inherent weaknesses in the natural ventilation systems, is not enough to reverse the clear advantages of natural ventilation systems in favour of mechanical systems, which, under the climatic and environmental conditions of the area, are documented mainly in terms of energy cost (Liddament, 1996). A series of techniques of natural ventilation -if they are planned and implemented in the design of corresponding systems - can contribute through their implementation to the reversal of the disadvantages which were detected and, still further, contribute to the improvement of the ventilation efficiency and the building's energy balance. Similar techniques and corresponding design approaches are described in related technical notes (e.g.: Liddament, 1996). As an example, some of these, which are considered to be compatible with the architectural and constructional characteristics of the buildings in the area, concern:
- The increase in the airtightness of the building shell, so that the intervention of the residents by opening and closing large openings will be more effective.
- The elimination of spaces with high demands on air quality (e.g.: living rooms) from the sides of the building which are nearest to the external sources of pollution (e.g.: roads with heavy traffic)
- The facilitation of interzonal air circulation by locating openings on the sides of the building furthers away from the sources of pollution.
- The use and exploitation of the advantages of louvres and "top hung" windows as well as air vents and "trickle" ventilators. Automatic air inlets, even though those which are sensitive to air quality parameters are costly, can be proved useful in many cases.

5. Conclusions

The relatively high air pollution values which, according to the conclusions of relevant studies, were met in the area of this medium-sized town in central Greece (Volos), affect, through building ventilation, the indoor air quality. This is more perceptible in buildings close to the pollutant emission sources (e.g.: roads with heavy traffic). The resistance to their influence, concerning the natural ventilation systems that prevail in the residential and commercial buildings of the area, is achieved to some degree by the control of the large openings (windows, balcony doors) in their shells. For this measure, in counterbalance to its inherent disadvantages, the periodical characteristics that were discovered for the concentration of pollutants together with the identification of their major emission sources contribute to its better application. Additionally, provided that the climatic and environmental conditions in the area do not justify the use of mechanical ventilation systems, a series of disadvantages in the performance of the natural ventilation systems could be lessened by the application of proper design approaches

This study is an attempt to investigate the issue of influence of the environmental pollution on the indoor air quality as well as possible measures to confront it using building ventilation systems in a particular area, based on the available data for the real conditions that are valid there. The variations in these data (environmental, climatic, ventilation conditions, etc.) from country to country and from region to region obviously change the whole image and impose the need for different approaches in every case for the confrontation of this particular problem.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

ASSESSING NATURAL URBAN VENTILATION THROUGH AN INTEGRATED MODEL

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Assessing Natural Urban Ventilation through an Integrated Model

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SYNOPSIS

The paper presents further then an integrated model the supporting methodology that allows to assess natural urban ventilation conditions both outside and inside constructions.
Though some particular aspects and procedures can be complex and time consuming the general structure is quite simple:
1. to establish wind regimes as a boundary condition - information can come from wind measurements at undisturbed areas like airports;
2. to integrate these regimes within the site - using numerical models to transfer information to the site;
3. to assess local wind velocities and pressures - promoting wind tunnel tests over physical models reproducing at a convenient scale its main characteristics;
4. to estimate ventilation rates - outside, from measurements, and inside, computing internal flow rates as dictated by both external and internal conditions.
Results can go from drawing general patterns of ground level winds, allowing to assess external ventilation and comfort conditions for pedestrians, to a computation of the flows and air proprieties, promoted inside a room taking into account small heat sources and sinks as well as the external conditions imposed by the wind.

LIST OF SYMBOLS

A area	h heat conductance	T temperature
C_{pk} pres coef. out. k opening	H height	U velocity
cp_{ar} specific heat of air	P pressure	ρ density
g gravity acceleration	Q heat power release	ζ head loss coef.

1. INTRODUCTION

In the middle of the 70's LNEC Applied Dynamics Division (NDA) promote its first works on natural ventilation in urban areas when pressure distributions were assessed in a physical model of Caracas Parque Central, Venezuela, and, further from static and dynamic wind loads, locations for natural ventilation intakes and exhaust of kitchens and sanitary rooms were defined taking into account the local wind regimes (*1*). By the end of the 70's and again for Caracas urban planners of Morellos, La Hoyada e Carabobo physical models were installed in the wind tunnel and the work was extended to assess external (ground level) ventilation conditions namely regarding pedestrian comfort (*2*). The beginning of the 80's brought out the development of the first numerical models based on integral equations and using data from wind tunnel tests (*3*) and by mid 80's the first numerical models based in differential equations were developed (*4*).

Along the years windows, doors and facades, static ventilators, ..., together with all types of construction components have been tested both in Laboratory facilities and *in situ* and results collected, treated and included in what is now a large data base at LNEC Behaviour of Construction Components Division (NCCp). Air tightness, for windows (*5*) and facades (*6*) and pressure coefficients for static ventilators (*7*) are included in that data.

In the late 80's fire studies in buildings allows for the construction of a new outdoor test facility simulating at a scale not smaller then 1:1.5 compartments of a house and their communications both internal and to the exterior (*8*) and, at the same time, a new wind tunnel of the Boundary Layer type with a large test chamber was built in order to develop studies where large areas and complex terrain were an important issue (*9*). Furthermore new numerical models were adopted or developed (*10*).

So being we consider that conditions are now established in such a way that an integrated approach to the study of natural ventilation in urban areas is available. The integrated model comprises both physical, at large and small scales, and numerical, integral and differential, models. Furthermore it is being developed both in time and frequency domains.

2. METHODOLOGY AND MODELS

Assess ventilation in urban areas demands the knowledge of the wind, the topology of the surroundings, the geometry of the site, the characteristics of the envelopes and the internal partitions and systems.

Wind information can be transferred through numerical modelling from undisturbed area like nearby airports to the site and used as a boundary condition to test in the Boundary Layer wind tunnel not only the pressure distribution of the specific building but also the general patterns of flow around it.

Window and facade characteristics can be measured at full scale in the test facilities the same applying for specific equipment like static ventilators tested in the wind tunnel.

Information can then be integrated with internal characteristics and internal heat sources, or sinks, so that numerical models both integral and differential can give a general idea of the internal flow patterns, the integral models providing a first approach for a complete description of the internal flow in a room.

Turbulence and its effects on ventilation rates can be analysed both in the time and in the frequency domain. Local values can be measured over the physical models and both time series and its statistical proprieties and power spectral density functions can be estimate.

This information allows to proceed to a time step integration either directly or through the generation of synthetic time series from the "spectra" and so assessing the time variation of the ventilation rates that can then be characterised either in statistical terms or in the frequency domain. This second possibility, now under development, aims to define appropriate transfer functions for the internal flows.

Case studies where one or various of the different experiments, software and procedures referred will be presented to illustrate the present state of the integrated model.

2.1 Wind Regimes

Figure 1 presents the wind rose as measured along a year (hourly mean velocities) for Lisbon airport as well as the result of its transposition for the EXPO'98 area assuming the equivalent roughness of the city as defined from its bouroughs, quarters, type of construction, ... and the orography (figure 2) (*11*). Since the height considered is clearly above the mean height of the buildings the information can be used as a boundary condition for the site (*12*). The software used, WAsP (*13*), is currently adopted by those working in the market of wind energy, though some additional information has to be considered. Normal inputs would be the wind raw data - velocity and direction - and the orography and roughness class of the terrain.

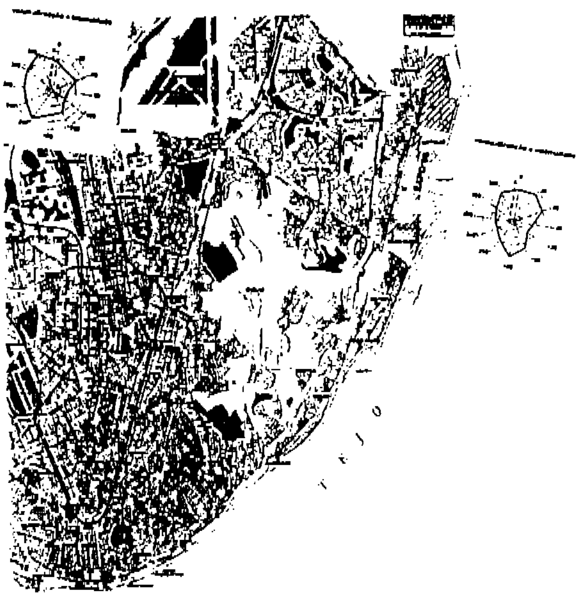

Fig. 1 Wind regimes in Lisbon airport and EXPO'98

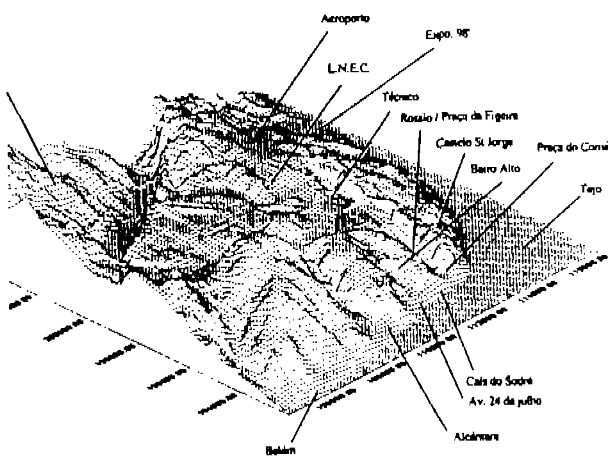

Fig. 2 Lisbon orography

2.2 Local Wind Patterns

Figure 3 presents general lay-out of the EXPO'98 area, in Lisbon, as from a model built at 1:2500 scale and installed in the wind tunnel, the flow field at ground level being measured through the erosion technique (*14*) for different wind incidences (*15*). Values are presented in non-dimensional form as a relation between the local velocity and that that will be observed if no constructions were presented at the site.

It is also possible to use non distorted scaled models of large areas in the BL wind tunnel and reproduce details (typical are 5 m obstacles) allowing for the flow to develop as it runs over the model from a very clear boundary condition upstream (for instance the sea (*16*)).

2.3 Pressure Distributions and Components Characteristics

Pressure distribution can be measured over the external walls of models equipped with pressure taps as is the case of the Multiusos pavilion represented in figure 4 (*17*) the neighbourhood of which was also reproduced in the wind tunnel. Boundary Layers not only

with velocity and turbulence intensity profiles but reproducing length and time scales can be generated either through natural evolution or through the Couningham technique (*18*).
Results are normally expressed in terms of mean pressure coefficients.
In what concerns flow characteristics of components like windows and doors, glass facades, ... and ventilators tests in LNEC facilities allow to assess those (*19*), (*20*). Figure 5 shows a large glass facade tested (*21*).

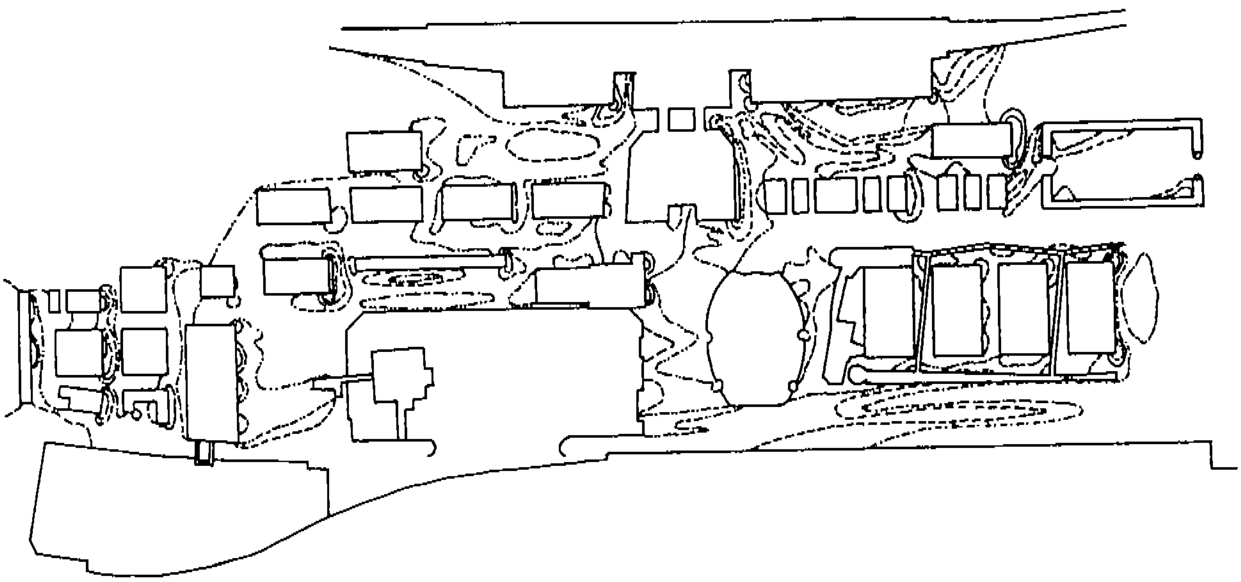

Fig. 3 Ground level winds on EXPO'98 for North incidence (prevailing summer winds)

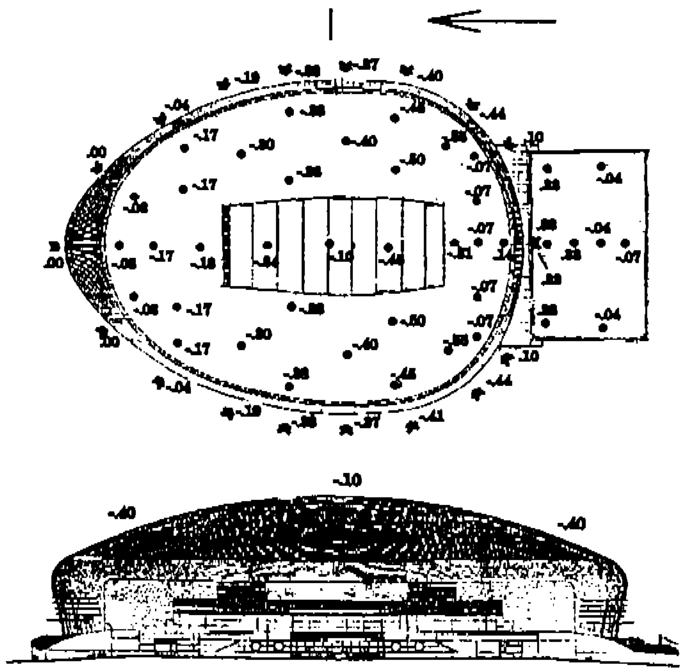

Fig. 4 Pressure over Multiusos pavilion at EXPO'98 for wind blowing from East

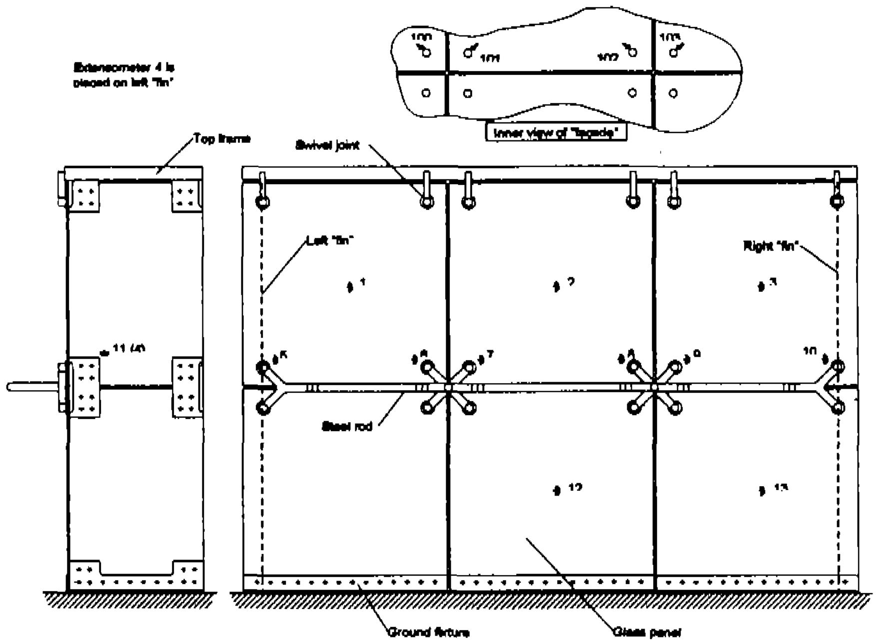

Fig 5 Section of the facade of Lisbon Oceanarium at EXPO'98 tested

2.4 Ventilation Rates

An analytical model for the prediction of ventilation rates, internal pressures and temperatures as influenced by the combined effects of natural wind action and heat generation or removal has been developed (*22*). Model inputs are external pressure coefficients, head loss coefficients of the openings and thermal conductance of walls and roofs, assumed to be known from experimental data.

$$\sum_{kl} U_{kl} A_{kl} = 0$$

$$\sum_{l} \sum_{kl} U_{kl} A_{kl} = 0$$

$$(\Delta\rho_l H_{kl} - \Delta\rho_l' H_{kl}')g + (\Delta p_l - \Delta p_l') - \varsigma_{kl} \tfrac{1}{2} \rho_0 U_{kl} |U_{kl}| = 0$$

$$\Delta\rho_l H_{kl} g + (\tfrac{1}{2} \rho_0 U_0^2 C_{pkl} - \Delta p_l) - \varsigma_{kl} \tfrac{1}{2} \rho_0 U_{kl} |U_{kl}| = 0$$

$$Q_l + \sum_{kl} \rho_0 c p_{ar} U_{kl} A_{kl} \Delta T_l + \sum_{ll} h_{ll} A_{ll} (\Delta T_l - \Delta T_l') = 0$$

$$\sum_{l} Q_l + \sum_{l} \sum_{kl} \rho_0 c p_{ar} U_{kl} A_{kl} \Delta T_l + \sum_{ll} h_{ll} A_{ll} (\Delta T_l - \Delta T_l') = 0$$

$$U_{kl} + U_{kl}' = 0$$

$$\frac{\Delta\rho_l}{\rho_0} + \frac{\Delta T_l}{T_0} = 0$$

Basic equations represent an integral balances of:

mass - for each room and for the whole building;

momentum - for each opening (expressed in terms of the Bernouilli equation);

energy - for each room and again for the whole building.

To these to sets of equations are added:

state equation - relating temperature variations with density variations thus allowing to represent buoyancy as a pressure difference in the momentum equation;
velocity compatibility - expressing that the flow through a communicating opening is the same as seen from both interconnected rooms.

2.5 Internal Flow Patterns

Internal flow patterns can be assessed through a rational approach combining 3D numerical simulation of the thermal and dynamic governing equations by means of a κ–ε two equation turbulence model with experimental data on wind pressure and pressure drop coefficients through the openings.
The equations can be written as follows (*23*)

$$\frac{\partial U_i}{\partial x_i} = 0$$

$$\frac{\partial}{\partial x_i}\rho U_i U_j - \frac{\partial}{\partial x_i}\left[\mu_t\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\right] + \frac{\partial P^*}{\partial x_j} + (\rho - \rho_0)g_j = 0$$

$$\frac{\partial}{\partial x_i}\rho U_i T - \frac{\partial}{\partial x_i}\left[\frac{\mu_t}{\mathrm{Pr}_t}\frac{\partial T}{\partial x_i}\right] - S_T = 0$$

$$\frac{\partial}{\partial x_i}\rho U_i \kappa - \frac{\partial}{\partial x_i}\left[\frac{\mu_t}{\mathrm{Pr}_\kappa}\frac{\partial \kappa}{\partial x_i}\right] - G - B + \rho\varepsilon = 0$$

$$\frac{\partial}{\partial x_i}\rho U_i \varepsilon - \frac{\partial}{\partial x_i}\left[\frac{\mu_t}{\mathrm{Pr}_\varepsilon}\frac{\partial \varepsilon}{\partial x_i}\right] - C_1\frac{\varepsilon}{\kappa}(G + B)(1 + C_3 R_f) - C_2\rho\frac{\varepsilon^2}{\kappa} = 0$$

$$\mu_t = \rho C_\mu \frac{\kappa^2}{\varepsilon}; G = \mu_t \frac{\partial U_i}{\partial x_j}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right); B = \beta g_i \frac{\mu_t}{\mathrm{Pr}_t}\frac{\partial T}{\partial x_i};$$

$$\beta = -\frac{1}{\rho}\frac{\partial \rho}{\partial T}; P^* = P + \frac{2}{3}\kappa$$

with

$C_\mu=0.09$; $C_1=1.44$; $C_2=1.92$; $\mathrm{Pr}_\kappa=1.0$; $\mathrm{Pr}_\varepsilon=1.3$; $\mathrm{Pr}_t=0.7$

In the form presented the momentum equations are subtracted by the static pressure equations, $\partial P_0 / \partial x_{ji} + \rho_0 g_j = 0$ to show the nature of the buoyant term $(\rho - \rho_0)g_j$.

In the transport equation for R_f is a flux Richardson number defined as $R_f = 0.5B_l / (B + G)$, B_l being the buoyancy production of only the lateral energy components, and C_3 is a numerical constant equal to 0.8.

Boundary conditions were established for openings - assuming the outside temperature and pressure as derived from the wind dynamic pressure times the local pressure coefficient and the compatibility equation through the opening is expressed in terms of a head loss coefficient - walls - the generalised log law was adopted and the conduction of heat through the wall assumed as usual - heat sources or sinks - considered in terms of its power as delivered in each cell.

The model initially developed for one room was further developed to allow the simultaneous computation in parallel of several rooms and then building a compatibility condition at each internal connection (*24*). Furthermore this model can now simulate the local combustion and the generation and transport of combustion products, namely soot and comply with its main aim - fire simulation.

2.6 Other Features of the Integral Model

The integral model already presented has been refined along the years and is now able to comply with problems like the transport of diluted substances including moisture (*25*) and its possible phase changes; is able to consider crack and other laminar (or at least no turbulent) flows (*26*) and to estimate the effects of wind turbulence as derived from power spectral density of turbulent velocity fluctuations and longitudinal and transverse correlation of gusts (*27*). Now under development is a model allowing to generate synthetic time series from the "spectra" (*28*) and to estimate through a step by step procedure that includes the ventilation transfer function of rooms time variations of air fluxes as induced in natural ventilation systems.

2.7 Experimental and Numerical Results

Validation of the methodology is now under way as a whole though for most of the procedures fair matches can be find between results arriving from experimental work and numerical modelling (29), (30), (31).

3. CONCLUSIONS

A reliable methodology and a set of procedures have been established in order to assess natural urban ventilation conditions based upon numerical modelling and wind tunnel tests over physical models reproducing at scale the main features of the area provided that appropriate boundary conditions for the flow (incidence, ABL characteristics) have been established. Sound data for component characteristics, based in experimental tests both in Laboratory facilities and on prototypes must be used.

Adoption of new models based on description of the external flow in the frequency domain and on appropriate transfer functions defined for each room looks promising.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

**19TH ANNUAL AIVC CONFERENCE,
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REENTRAINMENT OF POLLUTANTS FROM EXHAUSTED AIR - DISCUSSION OF DIFFERENT TYPES OF REGULATORY REQUIREMENTS

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SYNOPSIS

In many existing ventilation systems unintentional reentrainment of pollutant, due to improper location of exhaust and air intake, decreases quality of indoor environment. Unfortunately, the more precise method of assessment of exhaust plume behaviour, the more difficult potential application in regulatory codes and standards. The aim of the paper is to discuss advantages and disadvantages of different types of the models and their applications in regulatory requirements. Discussion addresses two standards: BSR/ASHRAE Standard 62-1989R Public Review Draft (August 1996) and new Polish building code. The conclusions highlight that at the moment there is no good procedure (simple and precise enough) to be commonly used in standards. The necessity for further research is pointed out.

1. INTRODUCTION

Practice indicates that even in cases when comfort mechanical ventilation systems operates only on fresh air some pollutants might be unintentionally recirculated because of improper location of exhaust and air intake. As pollutants also can be introduced to the buildings with infiltrating air, operable doors and windows that are a part of a natural ventilation system shall be treated as outdoor air intake. The problems mentioned above are very well known to industrial ventilation specialists. Although in cases of their interest exhausted gases are discharged by stacks, usually much higher than ventilation exhausts, but waste gases even after cleaning, are much more polluted than air discharged to the atmosphere from typical comfort ventilation systems. Because of this almost all-existing procedures for possible reentrainment assessment were developed originally for industrial buildings. The basic problem for the specialists is to generalise these models (build on limited experimental data) to the form able to cover all possible types and shapes of buildings and all possible combinations of input parameters (intake and exhaust location, discharge velocity, discharge direction, etc.). Moreover, as building codes and requirements have to be commonly understood the available procedures have to be as simple as possible.

2. LOCATION OF EXHAUSTS AND OUTDOOR AIR INTAKES

In number of buildings it is an architect who solves really difficult problem of best intake location. Unfortunately, he usually takes care of aesthetics of the building envelope but rarely understands aerodynamics and ventilation principles.

From engineering point of view, outdoor air intakes shall be located such that they draw on best possible outdoor air. In ideal situation the intake location should be the result of the wide analysis taking into account the following factors:

- shape of the building,
- pollution sources like (ventilation exhausts, plumbing vents, cooling towers, streets or roads, garage entry or loading areas etc.),
- prevailing winds and air flow patterns around the building and building elements,
- location of HVAC equipment inside the building.

Each case needs individual analysis but some very general hints can be formulated (e.g. in [1]). If possible exhaust air should be discharged vertically with relatively high velocity (value > 1,5 times wind speed is recommended) and above all possible obstacles and circulation zones.

In the case of relatively high buildings the air intakes should be located on the lower one – third of the building, high enough above ground to avoid wind blown dust, vegetation and vehicle exhausts. This not only creates big distance between exhaust and intake but also allows taking advantage of the natural separation of wind flow on the upper and lower half of the building.

In case of low buildings or when protection against wind blown dust and debris is especially important, intakes are located on the roofs. When the building has decentralised air exhaust with number of roof fans and other discharge points, it might be impossible to locate the intake according to the requirements.

3. MODELLING OF EXHAUST PLUME DISPERSION IN BUILDING SURROUNDING

Mathematical modelling of plume dispersion in building surrounding is a pretty complicated problem. Simple analytical models describing turbulent injection of aerosol into another twisted turbulent stream of air, all this close to boundary layer, do not exist. Off course specialists may use CFD method. However, one should remember that preparing input data files for such analysis is usually very time consuming. Moreover, in spite of very rapid development of computers, 3-D simulation software sill can not be installed on commonly used units.

In that conditions full-scale tests at existing objects or physical modelling are the best ways to learn about dispersal of polluted air discharged from "low emitters". Term "low emitter" means that wind flows around building have the predominant influence on pollutant dispersal. Several teams trying to generalise outcomes from their investigations developed empirical models.

In seventies *Nikitin et al.* published in Russia the set equations that allow potential designer to evaluate pollutant concentration at air intake for industrial buildings [4]. The methodology covers both stack and linear emitters (e.g. lines of roof windows). Designer may calculate concentration for points located at roof, back wall or at ground behind the building (local co-ordinates x,y). Main limitations of the method are:

- wind is always perpendicular to longer wall of the building,
- air exhaust discharge velocity is not taken into account,
- gas temperature is not taken into account,
- in all equations concentrations are inversely proportional to wind speed (minimum concentrations are obtained for the lowest limit of wind speed, 1m/s).

In USA the results of works carried out by *Halitsky, Wilson, Chui* and others built the combined ASHRAE procedure. Methodology is looking for minimal dilution that may occur when the wind almost directly ("stretched string" distance) transport pollutants from exhaust to air intake. Method takes into account apparent initial dilution at roof level, dilution due to distance between internal turbulence and additional dilution due to stack exhaust. Main limitations of the method are:

- method do not give the opportunity to calculate dilution for conditions other than critical, other wind direction and wind speed,
- method allow to calculate the shape of recirculation zones only for wind perpendicular to walls.

During works on Public Review Draft of BSR/ASHRAE Standard 62-1989R [2] the simplified version of ASHRAE model has been developed. General principles are common but number of additional assumptions were made (e.g. wind speed equals 2,5 m/s, exhaust is on

the roof level etc.). These assumptions can of course be listed as disadvantages of the method but one great advantage appeared - the simplicity.
Full comparison of the models mentioned above is not easy but even very simple example (fig. 1) can show the differences between results from presented methods. The volume of 500 l/s of polluted gases is emitted from industrial building through the stack with negligible height. Gases are discharged vertically with the velocity of 5 m/s.

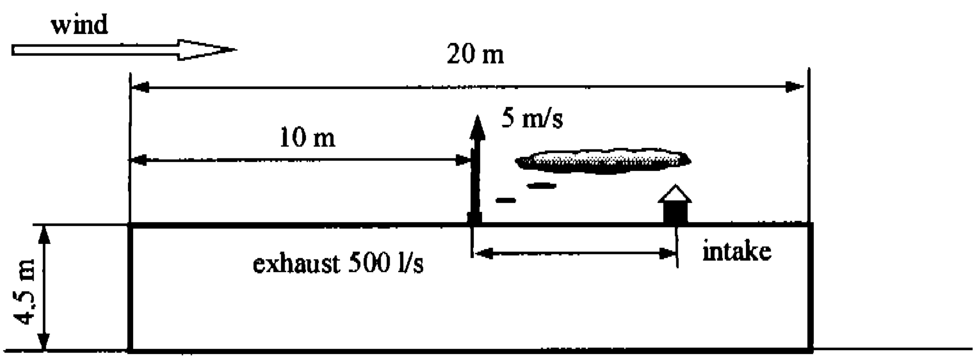

Figure 1. Scheme of analysed industrial building.

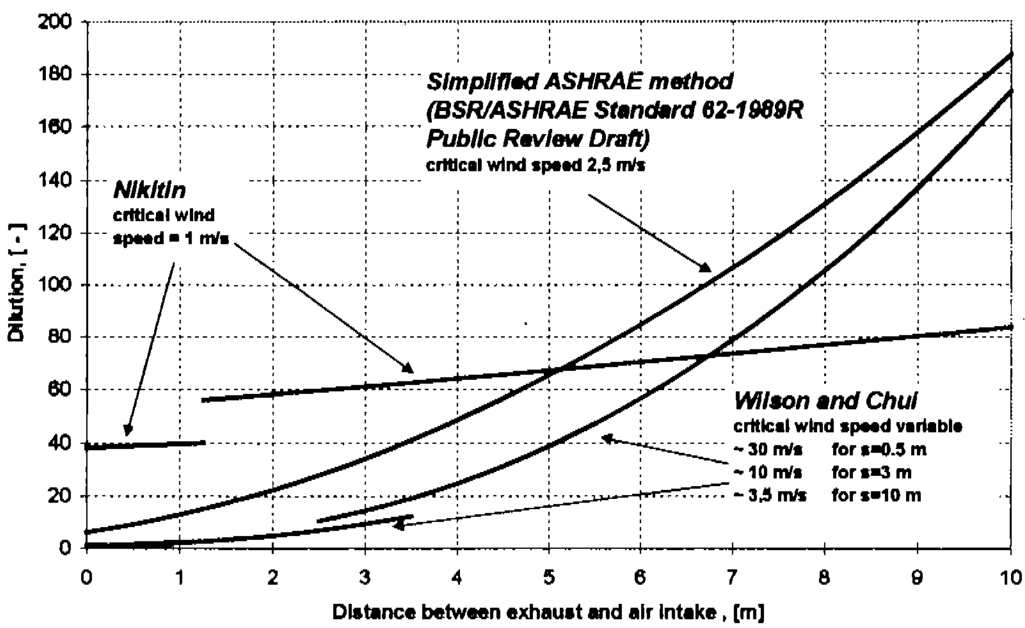

Figure 2. The comparison of dilution of pollutants calculated by different methods (geometry as presented at figure 1).

Analysis of the figure 2 indicates that there are big differences between results obtained using presented methods. The relative errors of the methods (taking the *Wilson* and *Chui* method as the reference one) are:

– for simplified ASHRAE method from ~550% for 0,5 m distance to 7 % for 10 m.
– for *Nikitin* method from ~3700% for 0 m distance to –50 % for 10 m.

Moreover, the differences in assumptions caused that each method estimates critical dilution for other wind speed:

– 1 m/s (constant) for *Nikitin* method,

- 3,5 ÷ 30 m/s (variable) for *Wilson* and *Chui* method.
- 2,5 m/s (constant) for simplified ASHRAE method.

There is no chance to state which method estimates the plume dispersal best in analysed case. One should remember that although presented methods allow designers to perform routine estimation but only full-scale field measurements or physical modelling can provide true answer. This is the suggested procedure for critical applications (e.g. hazardous pollutants).

4. EXAMPLES OF REGULATIONS CONCERNING LOCATION OF EXHAUSTS AND OUTDOOR AIR INTAKES

4.1. BSR/ASHRAE Standard 62-1989R Public Review Draft

Regulations in BSR/ASHRAE Standard 62-1989R Public Review Draft are generally based on ASHRAE model. The standard distinguishes 5 classes of exhausted air stating minimum dilution factor for each of class (table 1).

Table 1. Dilution factor determined as a function of exhaust air class (on a base of [2])

Exhaust Air Class	Dilution factor, D
Class 1: Air drawn from spaces without unusual sources of contaminants	5
Class 2: Air drawn from spaces that may have mild contaminant intensity,	10
Class 3: Air drawn or vented from locations with significant contaminant intensity,	15
Class 4: Air drawn or vented from locations with noxious or toxic fumes or gases,	25
Class 5: Effluent or exhaust air having a high concentration of dangerous particles, bioaerosols, or gases	50

The model uses the idea of the shortest "stretched string" distance measured from the closest point of the outlet opening to the closest point of the outdoor air intake opening, window or door opening, or property line along a trajectory as if a string were stretched between them.

The standard requires that the exhaust air and vent outlets shall be located no closer to property lines, outdoor air intakes, windows, and doors, both those on the subject property and those on adjacent properties, than the minimum separation distance S listed in special table. The distance is either calculated by simple formula taking into account: exhaust air volume, exhaust air discharge velocity and dilution factor (determined as a function of exhaust air class), or stated as the constant value.

Some exceptions are mentioned, regarding e.g. alternative design (approved by the authority having jurisdiction), unitary or factory packaged heating/ventilating unit and systems operating not simultaneously.

4.2. Requirements of the Polish Building Code

General requirement of the Polish Building Code [3] says that air intakes for ventilation and air conditioning systems should be located at places that allow drawing on not polluted air. In case when contamination exceed permissible values (Stated in Polish Environmental Codes) air should be cleaned before supply to rooms. Air intakes and exhausts should be located outside possible explosion zones, and the distance between them should be not less than 10 m.

In case of compact ventilation and air conditioning units that include both air intake and exhaust, the minimum separation can be less than 10 m, but the design should successfully protect air intake from reentrainment of exhaust air. The Code requires also that exhaust points from mechanical ventilation should be located at least 0,4 m above surface at which they are installed and at least 0,3 m above the line connecting the highest points of the nearest obstructions. As obstruction one regards the elements of the building located closer than 10 m from exhaust (in horizontal projection).

Distance between mechanical system exhausts and windows in dwellings and offices should not be less than 3 m in horizontal projection and 1 m in vertical projection. If the exhaust air contains substances harmful for environment and human health, horizontal distance should be increased to at least 6 m.

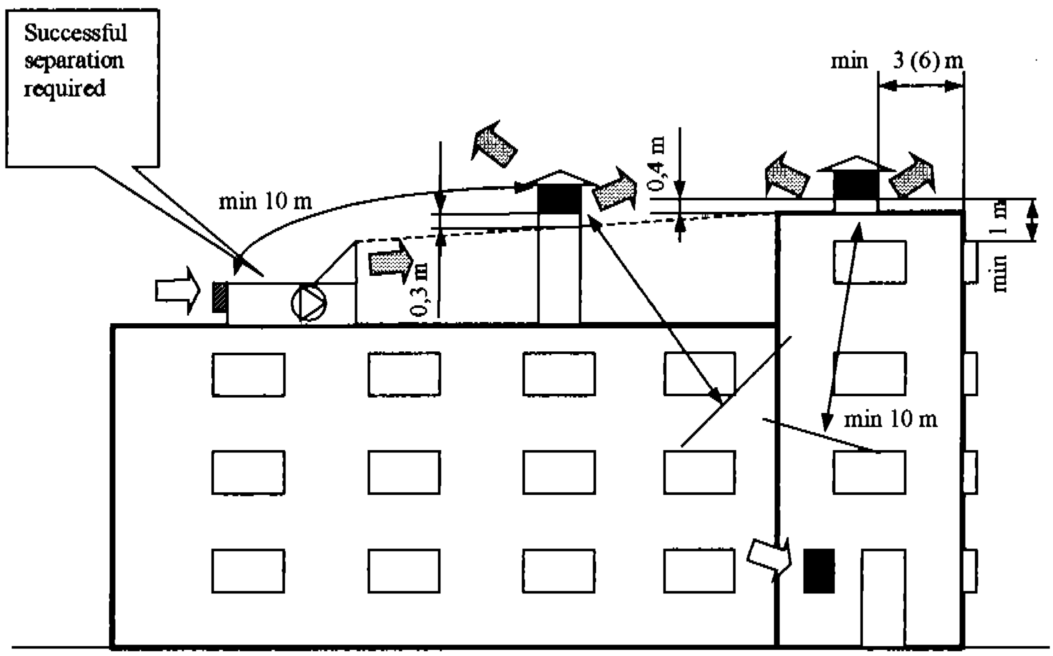

Figure 3. The general ideas of Polish Building Code [3] regarding exhaust and air intake locations.

5. DISCUSSION OF DIFFERENT TYPE OF REGULATORY REQUIREMENTS

The standards described above present two different type of regulation. Public Review Draft of BSR/ASHRAE Standard 62-1989R is pretty thick, with number of appendixes, simple calculation methods and additional notes. It looks like the "first aid kit" for ventilation systems designers. Polish building code looks as it has been developed for verification authorities. There are no explanation notes or calculation methods. The code contains number of simple, easy to check requirements.

Figure 4 presents the comparison of minimum exhaust-intake separation distance required by discussed standards. Negative discharged velocity means that exhaust is directed less than 74° towards the intake. Moreover according to simplified ASHRAE method, discharge velocity shall be set to 0 for vents from gravity (atmospheric) fuel fired appliances, plumbing vents, and other non-powered exhausts, or if the exhaust discharge is covered by a cap or other device that dissipates the exhaust air stream. For hot gas exhausts such as combustion products, an effective additional 2.5 m/s upward velocity shall be added to the actual discharge velocity.

Figure 4 indicates that generally Polish Building Code requires much higher separation distance than proposal of new BSR/ASHRAE Standard 62-1989R. For example presented at figures 2 and 3, depending on the method, dilution factor at 10 m reaches 84 (*Nikitin*), 174 (*Wilson* and *Chui*) method) or 187 (simplified ASHRAE method). All this values are much higher than dilution required by Public Review Draft of BSR/ASHRAE Standard 62-1989R even for effluent or exhaust air having a high concentration of dangerous particles, bioaerosols, or gases. According to simplified ASHRAE method even discharge of polluted gases towards air intake with velocity of 8 m/s allow keeping dilution of 50 at separation distance of 10 m. In this context it is worth to notice that in analysed example, Polish code would not allow to locate air intake on the roof at all.

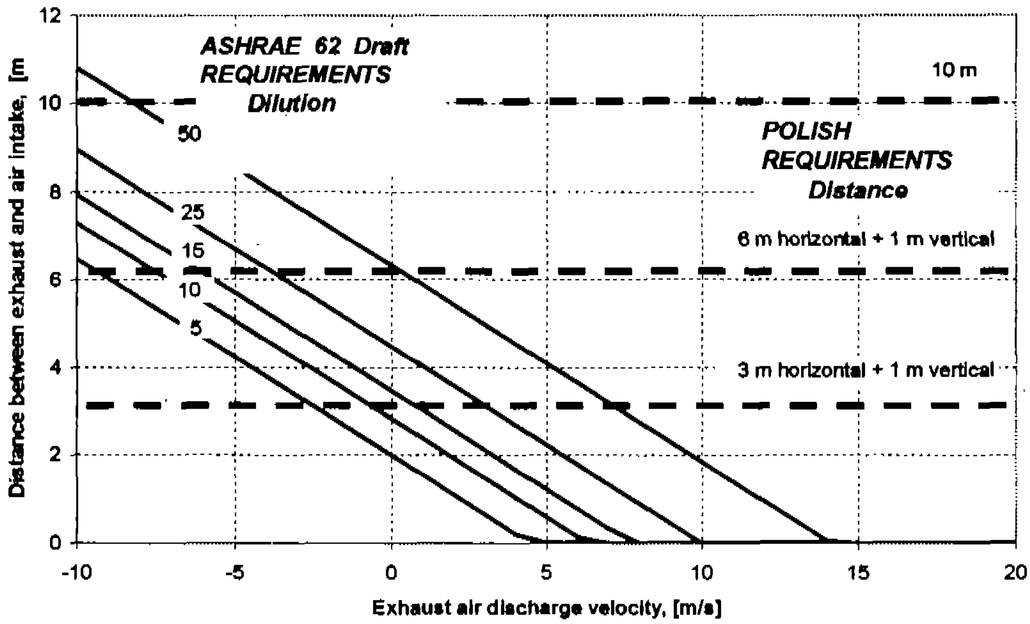

Figure 4. The general ideas of Polish Building code regarding exhaust and air intake locations.

Both BSR/ASHRAE Standard 62-1989R Public Review Draft and Polish Building Code allow drawing some conclusions concerning advantages and disadvantages of different types of requirements protecting ventilation systems from reentrainment of pollutant due to improper location of exhaust and air intake. Brief summary is presented in table 2.

6. CONCLUSIONS

There is very little well-verified and commonly accepted methods for exhaust plume behaviour. Existing models are usually to complex to use them in the regulatory codes.
Authors of ventilation codes willingly use requirements describing location of exhaust and air intake or separation distance. These statements are usually not flexible and probably over estimate potential risk. Simplifications of the models increase uncertainty of estimation and than it is very important to understand that even where the required minimum separation distances are maintained, reentrainment of odours and toxic gases may still occur depending on wind conditions, building geometry, and exhaust design.

It seems that further research should be carried out to prepare more reliable way of incorporation methods of protection against reentrainment of pollutants from exhausted air into the standards and ventilation codes.

Table 2. Advantages and disadvantages of different type of requirements protecting ventilation systems from reentrainment of pollutant, due to improper location of exhaust and air intake.

Type of the requirement		Advantages and disadvantages of the requirement
Concentration of pollutants in air intake or supplied air	Adv:	– addresses the most important factor, – checking is possible.
	Disadv:	– checking is pretty complicated (chemical analysis required), – precise measurements interpretation procedure is necessary, – difficult to apply during designing process, – in case of requirements regarding air intake, at highly polluted areas location of air intake may not be possible.
Minimum dilution factor	Adv:	– addresses very important factor (at not polluted areas possibly critical), – checking is possible (e.g. tracer gases).
	Disadv:	– does not take background of pollutants into account, – precise measurements interpretation procedure is necessary, – without standard method of estimation do not applicable during designing process.
Standard method of estimation	Adv:	– allow wider analysis during designing process, – provides flexibility to the designer or architects.
	Disadv:	– usually do not allow calculation for several emitters, – may create the impression that after analysis reentrainment will not occur in any circumstances.
Location of exhaust air intake or separation distance	Adv:	– allows to formulate very simple and easily understood statement, – checking the compliance with regulation is very easy.
	Disadv:	– difficulties with prediction of all possible situation cause that regulation is usually not flexible. and overestimate potential risk

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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NatVent™: Aims and Vision

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ABSTRACT
This paper gives an overview of the EC NatVent™ project on 'Overcoming Technical Barriers to Low Energy Natural Ventilation in Office Type Buildings in Moderate and Cold Climates' which has been carried out under the European Commission Joule Programme 1994-98. The project was targeted at countries with low winter and moderate summer temperatures where summer overheating from solar and internal gain can be significantly reduced by low-energy design and good natural ventilation. In addition, the project addressed natural ventilation solutions to buildings located in urban areas where external air pollution and noise levels are usually regarded as being high. In this paper, the background to the project together with the objectives have been outlined and a brief account of the various research areas studied, solutions provided and products resulting from each area have been given.

1. BACKGROUND
Concerns over the adverse environmental impact of high-energy usage required for mechanical ventilation and air-conditioning have been growing over recent years. As a result, the design of energy efficient buildings employing natural ventilation strategies has been encouraged.

Natural ventilation is not just about openable windows. It is rather a holistic design concept, which according to Cook and McEvoy (1996) is now being used in the architectural design of large offices and other building types. Design is centred on using passive ventilation, based on the 'stack' (temperature) effect and wind pressure differentials, to supply fresh air to building interiors even when the windows are closed. As part of this process, designs incorporate atria or internal stairwells which, in some instances, use low-energy fans to provide 'assisted natural ventilation' (ie low-energy ventilation).

Buildings incorporating such strategies can provide year-round comfort, with good user control, at minimum capital cost and with negligible maintenance. In addition, naturally ventilated buildings can typically consume less than half the energy consumed in air-conditioned buildings. It is estimated that even a modest 10% take up of these strategies could save about six million tonnes of oil equivalent and 25 million tonnes of avoided CO_2 emissions every year within the EU. In addition, a survey (Ellis, 1994) carried out in the UK a few years ago, concluded that 90% of senior management preferred buildings without air-conditioning. Making the best use of natural ventilation and daylighting were at the top of the occupiers' most important design features in both urban and out-of-town offices.

However, despite this many investors have felt that air-conditioning is still necessary for many buildings, in particular where there is an increase in heat load (for example, from computers and office machinery) and in areas where there is an intrusion of external air and noise pollution. At an Experts' workshop in January 1995, the European Network of Building Research Institutes (ENBRI) working group on environmental issues identified solutions to these problems as one requiring priority action. Arising from this and other decisions, the NatVent™ project on 'Overcoming Technical Barriers to Low Energy Natural Ventilation in Office Type Buildings in Moderate and Cold Climates' was set up within the EC Joule Fourth Framework Programme 'Rational Use of Energy in Buildings'. The main action was to address the above issues, provide natural ventilation solutions and carry out effective dissemination.

2. OBJECTIVES OF NatVent™

The main objective of this pan-European project was to reduce primary energy consumption (and consequently CO_2 emissions) in office type buildings without compromising indoor air quality and comfort. This main objective has been achieved through two specific objectives:

- To identify and overcome technical barriers which restrict the implementation of natural ventilation and low-energy cooling in countries with moderate and cold climates.
- To provide practical solutions and guidance and thus encourage the wider uptake of natural ventilation technologies.

The project targets countries with low winter and moderate summer temperatures (i.e. Central and Northern European countries) where summer overheating from solar and internal gain can be significantly reduced by low-energy design and optimum natural ventilation. An additional priority was to develop natural ventilation solutions to buildings in urban areas where external air pollution and noise levels are regarded as being high. The project was aimed at both new-designs and major refurbishments.

3. The NatVent™ Consortium

The NatVent™ Consortium consists of the following:

- Building Research Establishment (BRE) from the United Kingdom as the overall Co-ordinator of the project.
- Belgian Building Research Institute (BBRI)
- Danish Building Research Institute (SBI)
- Dutch Building Institute (TNO)
- AB Jacobson and Widmark (J&W) from Sweden
- Technical University of Delft (TUD) from The Netherlands
- Willan Building Group (WILLAN) from the UK
- Norwegian Building Research Institute (NBI)
- Sulzer Infra Laboratory (SULZER) from Switzerland

All Consortium members are at the forefront of current ventilation technology and most are involved in national and CEN activities (developing codes and regulations). The vision of the Consortium was for NatVent™ to play a catalytic role in promoting common natural ventilation strategies and technologies within the European Union.

4. KEY TECHNICAL ACTIVITIES

The above objectives have been addressed by carrying out the key technical activities as shown in Figure 1.

Activity 1: Identification of technical barriers to natural ventilation

This activity was led by the Danish Building Research Institute (SBI) with the aim of identifying perceived barriers as seen by building professionals that restrict the implementation of natural ventilation and lead to the decision to install mechanical ventilation systems. Barriers were identified by carrying out in-depth structured interviews based on questionnaires among leading designers, architects, consultants, building owners and developers in each of the participating countries. A European-wide questionnaire was produced with input from all nine Partners within the NatVent™ consortium. In total about 105 interviews in the seven countries were carried out.

The survey identified that, with varying degrees, there is a significant lack of knowledge and experience on specially designed natural ventilation strategy in office buildings compared with that on mechanical ventilation. In addition, there is a lack of source material on natural ventilation knowledge in standards, guidelines and building studies throughout Europe. There was also a universal requirement for new design tools on natural ventilation including calculation rules as well as computer programs, which are numerically advanced but still simple to use by the non-specialist. As a result of this activity, a number of published papers and reports have been produced comparing country specific barriers as well as common barriers amongst the participating countries (Aggerholm, 1998).

Activity 2: Monitoring the performance of buildings.
The Belgian Building Research Institute (BBRI) and Sulzer Infra Laboratory (SULZER) led this activity. Its aim was to establish pragmatic innovative design strategies by monitoring the performance of existing low-energy buildings and providing case studies of current best practice. It is recognised that ultimately it is important to demonstrate the viability of natural ventilation in both performance and competitiveness if it is to succeed in the market place. Therefore, performance criteria and cost are important parameters when choosing between natural ventilation and alternative solutions.

To this end, cost effective and pragmatic measuring procedures and protocols were developed and used to evaluate the performance of existing ad-hoc buildings that were designed and constructed specifically as energy-efficient naturally ventilated buildings. Nineteen buildings within the seven EU countries were monitored in detail during the winter and summer periods (Ducarme, 1996). Figure 2 shows that the buildings cover a wide spectrum of shape and size yet they all use low-energy ventilation technology.

During monitoring, parameters such as room temperatures, humidity, carbon dioxide levels and ventilation rates were measured. In one building that was located in a highly polluted area, levels of pollutants such as carbon monoxide and traffic noise were also monitored.

Possible shortcomings and advantages from the ventilation strategies used during the summer and winter monitoring periods were identified. At its most basic level, for example, some users experienced difficulty in operating or understanding the control strategy. In general, these problems have mostly been such that they can easily be rectified. However, it is important that these experiences are disseminated to designers to prevent future problems in other buildings. Therefore, recommendations for achieving overall successful natural ventilation in buildings have been identified and short summary reports as well as a major detailed report on the findings have been produced. In addition, details of all the monitoring activities together with the results and recommendations are also available on the NatVent™ CD-ROM. Further details of this activity may be found in Demeester (1998)

Activity 3: Providing solutions to technical issues.
This third activity co-ordinated by BRE, was aimed at developing 'smart' naturally ventilated technology systems and component solutions to overcome the technical barriers identified in

Activity 1. This was done through the following five key tasks:

(i) ***Low-energy air supply components for use in buildings in urban locations.***
This task was led by the Willan Building Services. Its aim was to develop components and strategies for natural ventilation in non-domestic buildings located in urban areas with high external pollution and noise loads.

As part of this work, existing systems have been evaluated and current standards, performance and specifications compiled. A checklist has been developed which summarises all the pollution problems attributed to urban areas to provide guidance on the location of inlets and outlets. In addition, a spreadsheet based design tool has been developed that determines adequate air inlet sizes depending upon the ventilation requirements for a building. A low pressure drop inlet which is capable of damping noise levels and filtering particles has also been developed. In addition, a number of publications on this work have produced. Further details of this activity may be found in Ajiboye (1998).

(ii) ***Controlled airflow inlets to account for variability in weather.***
This area has been led by the Dutch Building Research Institute (TNO) with the aim of identifying and specifying conditions under which newly-developed natural ventilation 'smart' controlled air flow inlets can provide acceptable indoor air quality for occupants' health and comfort in offices.

Natural ventilation depends upon external climatic conditions and therefore changes in the wind speed, direction and temperatures affect the amount of fresh air which flows through the openings. An important aspect of a controlled airflow strategy is thus to provide an optimum quantity of fresh air for occupants in a manner that is generally independent of short-term external weather fluctuations.

To ensure a controlled flow rate, especially during the winter, when high driving forces could result in excessive heat loss, usually require active and quite complex control strategies. As an alternative, 'passive' vents have been developed which provide a measure of pressure independent flow without the need for mechanically controlled actuators. The breakthrough has been the development of vents that operate at extremely low driving pressures of a few Pascals (say upto 20 Pa). Such a device has been developed under this activity. In addition, to demonstrate their viability, an interactive user-friendly software program was developed. This gives visual indication of ventilation, air quality and thermal parameters for many ventilation and weather configurations. Further details of this activity may be found in De Gids (1998).

(iii) ***Natural ventilation with heat recovery.***
This task was led by the Norwegian Building Research Institute (NBI). Its aim was to develop natural ventilation systems with heat recovery. In very cold climates, an acceptable assumption is that natural ventilation without heat recovery could result in unacceptably high consumption of energy. Therefore, natural ventilation systems with heat recovery provides an energy efficient system for the colder climates where traditionally mechanical ventilation (with high energy consumption fans) has been

used to overcome the high pressure drops associated with such systems.

The study here has included determining the distribution of available driving pressure at key locations within each participating country. An advanced low-energy fan assisted natural ventilation system with heat recovery has been developed. The fan is extremely energy efficient and consumes approximately 0.25W for each l/s of air flowing through the system. Further details of this activity may be found in Brunsell et al (1998).

(iv) *Low-energy cooling*

The Dutch Technical University of Delft (TUD) led this task to develop low-energy cooling strategies. An important aspect of natural ventilation design is to prevent the need for refrigerative cooling. In much of Northern Europe, excessive external temperature and humidity rarely present a problem. Instead, buildings tend to overheat as a consequence of high internal heat loads and solar gains. Hardware and control algorithms have been developed to minimise these problems.

The control strategy for night cooling has focused on:

- predictive control;
- cooling day control;
- set-point control;
- slab temperature control;
- degree-hours control.

Hardware window prototypes for controlled night cooling have been developed. Associated with this is a graphical tool for designing effective window openings. Further details of this activity, its findings and products may be found in van Paassen and Leim (1998).

(v) *Integration of 'smart' systems for optimum year-round performance.*

This area was led by AB Jacobson and Widmark. A simple but reliable design tool, integrating all the elements of NatVent™ was developed so that an optimum solution for any building could be found. Key features in the design tool included the following:

- driving forces (wind and temperature);
- air flow through components;
- solar radiation; and a
- thermal model.

These components have been incorporated into a visual basic model with a simple user interface. Behind this is an extensive numerical database and pre-selected default data. Output includes air change rate, heat losses and related data. Further details may be found in Kronvall et al (1998).

5. DISSEMINATION ACTIVITIES

Effective and widespread dissemination and communication of the results has been a key activity in the NatVent™ project. Results have been disseminated to a wide spectrum of the construction industry, to building designers, architects, researchers and services engineers through national and international conferences and workshops. Successful workshops in all the participating countries have been held and a network of European architects has been established to provide advice to the Consortium. A conference on NatVent™ (Kukadia, 1998) has also recently been held. Further dissemination activities are being carried out through publication of the NatVent™ Guide, CD-ROM and various design tools.

6. PRODUCTS FROM THE PROJECT

The following is a summary of the products that are available from the project:

- Individual country reports on Technical Barriers
- European report on Technical Barriers
- Building case studies reports
- Design tools
  - Spreadsheet based tool for the determination of inlet size and location
  - An interactive user-friendly algorithm which gives visual indication of ventilation, air quality and thermal parameters
- Components
  - Inlet which is acoustically treated and deals with particle attenuation
  - Controlled air flow inlet to compensate for the variations in external climate
  - Low pressure heat recovery system
  - Night cooling devices and controls
- NatVent™ Guidebook
- CD_ROM: This contains all the details about the project and its participants, full information about and reports on all the technical areas covered, design tools and also information on hardware developed from the project.

7. CONCLUSIONS

The aim of the NatVent™ project was to reduce primary energy consumption by overcoming the technical barriers that prevent the implementation of natural ventilation in countries with moderate and cold climates. This has been achieved by providing solutions and disseminating the results to a wide audience in order to encourage and accelerate the use of low energy ventilation systems as the main design option.

NatVent™ has been successful in identifying the technical barriers that exist amongst building professionals. It has established innovative design strategies by monitoring existing low-energy buildings throughout Europe and thus provided case studies of current practice with recommendations. It has provided the following technological solutions to the issues identified in the project:

- A low pressure drop inlet for attenuating external air and noise pollution in urban areas together with a design tools for the sizing and location of inlets.
- A very low pressure drop vent with controlled incoming air flow to account for variable external weather conditions.

- A low energy fan assisted system to recover heat for use in the colder climates
- Hardware window prototypes and control algorithms for controlled night cooling to minimise summer overheating.
- A simple design tool integrating all the elements of NatVent™ giving an optimum solution.

In particular, NatVent™ has contributed significantly to the dissemination of results and promoting the use of natural ventilation strategies throughout the participating countries. As a result, it has raised the interest in many of the EU countries who are now looking at low energy ventilation technology as a first design option.

Therefore, in conclusion, it is our view that NatVent™ has combined 19th and 20th century strategies with late 20th century technologies to provide low-energy ventilation for the new millennium.

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8. ACKNOWLEDGEMENTS

The NatVent™ project has been funded partly by the European Commission under the Joule Programme 1994-98; and by the appropriate funding organisation within each participating country. Within the UK, we acknowledge with thanks the funding of the Department of the Environment, Transport and the Regions under the Partners in Technology (now Partners in Innovation) Programme. The NatVent™ Consortium would also like to thank Martin Liddament, Head of the IEA AIVC and Expert Adjudicator for the project, for his support and advice throughout the duration of the project.

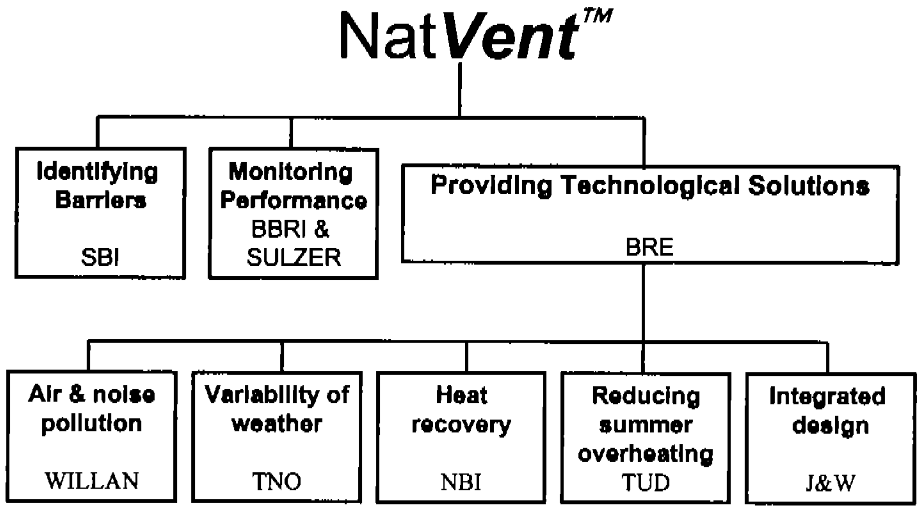

Figure 1: Overview of activities for the NatVent™ project.

Figure 2. NatVent™ Monitored Buildings

VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER, 1998

Perceived Barriers to Natural Ventilation in Offices

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Perceived Barriers to Natural Ventilation in Offices

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Synopsis

The paper describes the results of a Pan-European survey carried out on identifying the barriers that restrict the implementation of natural or simple fan-assisted ventilation systems in the design of new office-type buildings and in the refurbishment of existing such buildings. The survey was part of the ***NatVent***™ project carried out in seven central and northern European countries with moderate and cold climates.

The barriers were identified through an in-depth study with structured interviews based on questionnaires among leading designers and decision makers: architects, consultant engineers, contractors, developers, owners and governmental decision makers responsible for regulations and standards.

On average the interviewees expect an increase in the future use of natural ventilation in office buildings. The survey also identified a lack of knowledge and experience on specially designed natural ventilation. In addition, the results showed that there is currently, a lack of information on natural ventilation in standards and guidelines and also a lack of case studies of the performance of office buildings with natural ventilation. Furthermore, there is a significant requirement for simple tools, which can be used to design for natural ventilation, in particular, calculation rules, and easy-to-use computer programs.

1. Introduction

Mechanical ventilation systems are often installed in office buildings where good natural ventilation would have been sufficient to obtain comfortable indoor climate and good indoor air quality. It is thus important to identify the barriers seen by designers and decision makers which restrict the implementation of natural ventilation systems and lead to the decision to install mechanical ventilation plants in office buildings where it is not strictly necessary. Knowing the barriers is the first step in providing solutions to overcome them.

2. Interviews

A total number of 107 designers and decision makers were interviewed, see table 1. The interviewees were selected with the intention of also identifying the variety in opinions and viewpoints on natural ventilation in office buildings among people with the same profession.

Table 1. Number of interviewees in each country, by profession.

		Architects	Consultant engineers	Contractors	Developers	Owners	Government dec. makers	Total
Belgium	B	7	3	1	1	1	1	14
Denmark	DK	5	3	2	2	2	1	15
Switzerland	CH	5	3	2	2	2	1	15
Norway	N	5	3	2	2	2	1	15
Netherlands	NL	5	2	-	-	2	-	9
Sweden	S	5	3	2	-	2	1	13
Great Britain	UK	10	7	2	2	3	2	26
Total		42	24	11	9	14	7	107

The interviews were based on two questionnaires as follows:

- General view on natural ventilation in office buildings.
  General knowledge, viewpoints, experience and perceived problems with natural ventilation systems in office type buildings.
- Specific building project.
  Decisions actually made during the design or refurbishment of an office type building.

A specific 5 point scale was used where possible. The questionnaires were not too tight and there was ample space for additional comments, remarks and viewpoints not included in the questions. The questionnaires were completed by the interviewee and the interviewer together. In general both parts of the interview were performed with all interviewees. The only exception is the governmental decision makers, where only the general view was relevant.

3 Main results

The main results from the interviews are given in this paper. Details can be found in the references. The results in the figures are given as the average for each group. *All* is the average of all profession groups except the governmental decision makers.

3.1 Knowledge on ventilation

Nearly all the interviewees have less knowledge on special designed natural ventilation compared to their knowledge on mechanical ventilation in offices, see figure 1. Especially in Belgium, Denmark, Switzerland and Norway the knowledge on special designed natural ventilation is very low compared to the knowledge on mechanical ventilation, see figure 2.

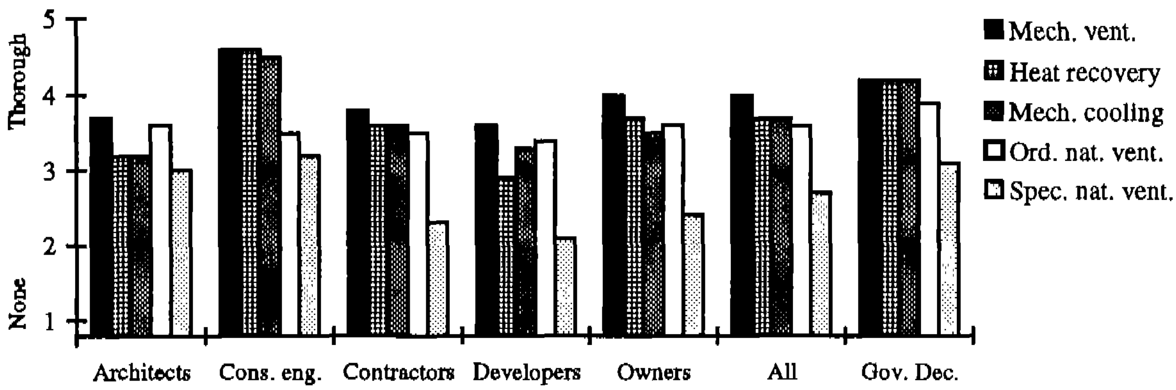

Figure 1. The interviewees' perception of own knowledge, by profession.

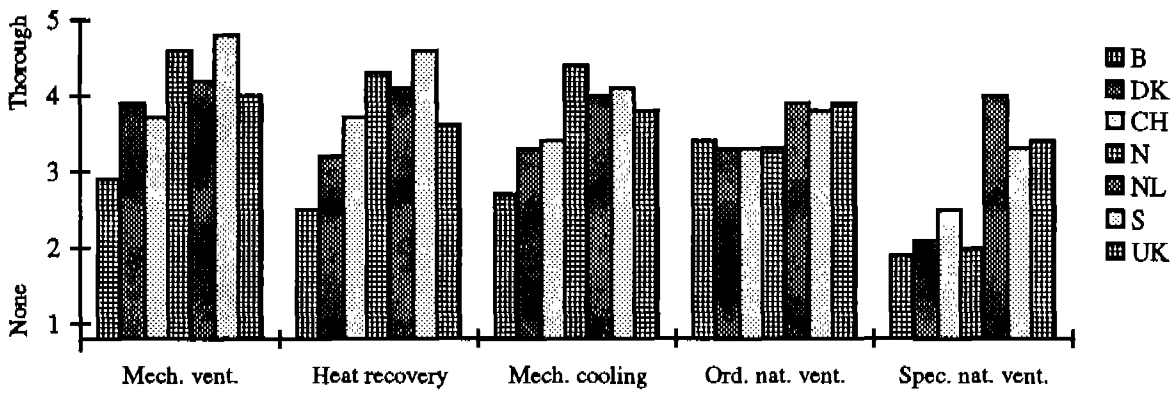

Figure 2. The interviewees' perception of own knowledge, by country

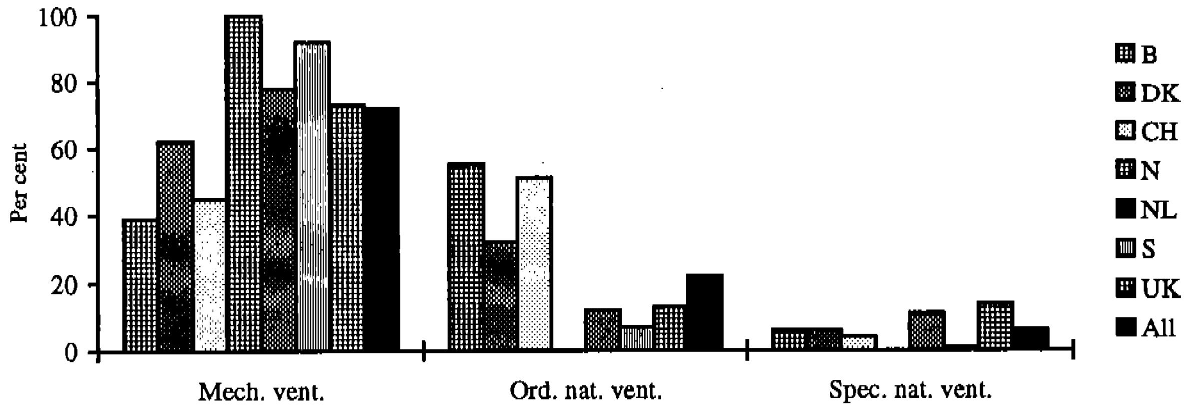

Figure 3. The interviewees' relative experience with mechanical, ordinary and special designed natural ventilation in new offices. The scale is the per cent of offices designed, constructed or owned

On average the interviewees have the same level of knowledge on ordinary natural ventilation as on mechanical ventilation. The exception is the consultant engineers who in general have less knowledge on ordinary natural ventilation compared to their knowledge on mechanical ventilation. The interviewees have indicated their level of knowledge on the five topics based on the knowledge necessary to perform their normal task in the design or decision process. It is therefore not possible to compare the absolute level of knowledge between the professions.

3.2 Experience

Most of the interviewees have much experience on mechanical ventilation in offices, whereas the experience with special designed naturally ventilated offices is very limited, see figure 3. Many of the interviewees have worked with ordinary natural ventilation in office buildings, but the actual number of buildings designed, constructed or owned varies significantly. The exception is Norway where none of the interviewees had designed, constructed or owned an office building with natural ventilation. The experience with ventilation in refurbished offices was about the same as in new offices.

3.3 Source to natural ventilation knowledge

The general opinion among the interviewees is that there is huge lack of good sources to natural ventilation knowledge. The mentioned sources are very sporadic and nearly no specific sources were mentioned by more than one or two of the interviewees.

3.4 Project fee

In many of the countries most architects and consultant engineers are normally paid according to design fee rules of the national Counsel of Practising Architects or Counsel of Consultant Engineers, and with the fee for the detailed design fixed based on a percentage of the estimated construction costs.

3.5 Design

There is no significant difference in the interviewees' perception of the *ease of design* in the four cases: natural ventilation in cellular offices, natural ventilation in open plan offices, mechanical ventilation in cellular offices and mechanical ventilation in open plan offices, see figure 4. Many of the interviewees emphasised that the ease of design also depends on the demands of the indoor climate and on the complexity of the ventilation system.

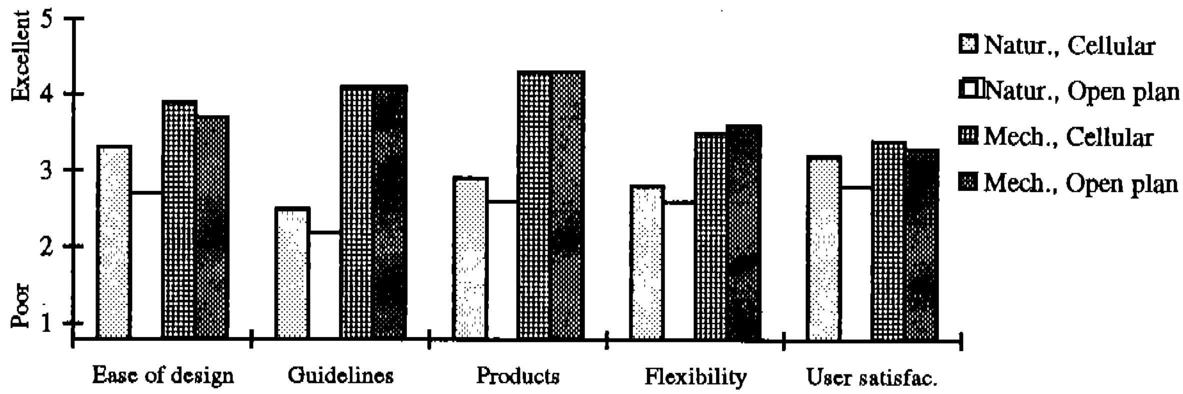

Figure 4. The interviewees' perception of the design of ventilation offices.

Nearly all interviewees found that the availability of design *guidelines* and *products* were better on mechanical ventilation when compared with natural ventilation. The interviewees also expected a higher *flexibility* in mechanical ventilated offices than in natural ventilated offices.

The interviewees expect about the same *user satisfaction* in natural and mechanical ventilated cellular offices. They also expect higher user satisfaction in natural ventilated cellular offices than in natural ventilated open plan offices. If mechanical ventilated the expected user satisfaction is the same in cellular and in open plan offices. It was mentioned that user satisfaction also depends on the expectations, which are normally higher in mechanical ventilated offices.

3.6 Performance in practice

In general the interviewees expect a better performance of mechanical ventilation systems than of natural ventilation systems regarding cooling effectiveness, draught minimisation, ability to remove odours and pollutants, ability to prevent ingress of odours and pollutants and insulation against external noise, see figure 5. Regarding generation or transmission of internal noise the same performance level is expected by natural and mechanical ventilation. Several of the interviewees emphasised that the performance also depends on the design.

3.7 Controllability

In general the interviewees expected a high degree of central controllability of mechanical ventilation systems and a low degree of central controllability of natural ventilation systems especially in cellular offices, see figure 6. The expected degree of local and individual controllability of the ventilation is a little higher in cellular offices than in open plan offices.

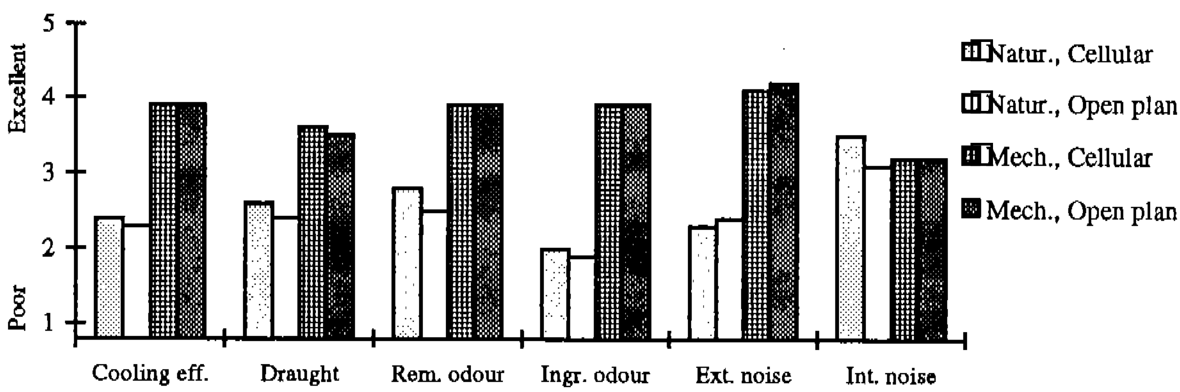

Figure 5. The interviewees' perception of the performance in practice of office ventilation.

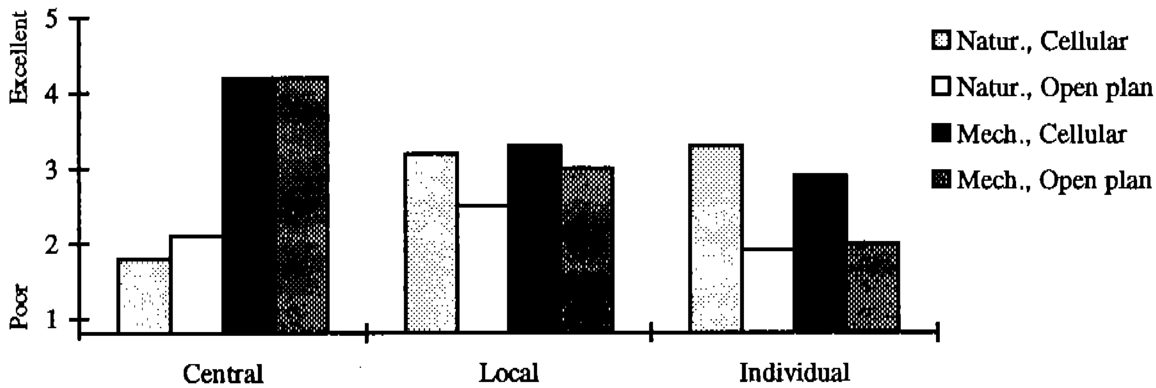

Figure 6. The interviewees' perception of the controllability of office ventilation.

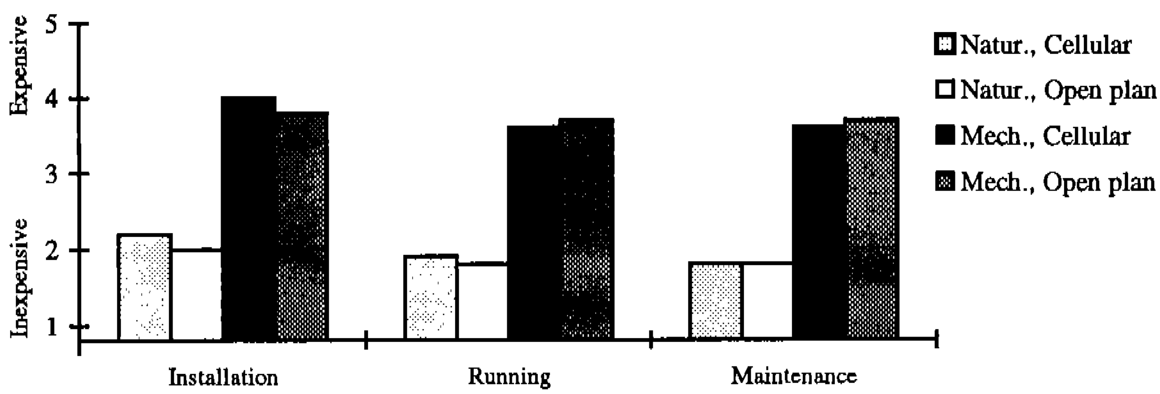

Figure 7. The interviewees' perception of the costs for office ventilation.

3.8 Costs

Most interviewees expect higher installation, higher running and higher maintenance costs for mechanical ventilation systems than for natural ventilation systems, see figure 7. Several of the interviewees emphasised that if mechanical ventilation is installed a perceptible percentage of the total construction costs would be for the mechanical ventilation systems. It was also mentioned that the costs for natural ventilation is high if additional space is required.

3.9 Expected future use of natural ventilation

The architects in general have the highest expectations of an increase in the use of natural ventilation in offices, see figure 8. On average only the governmental decision makers expect a decrease in the use of natural ventilation.

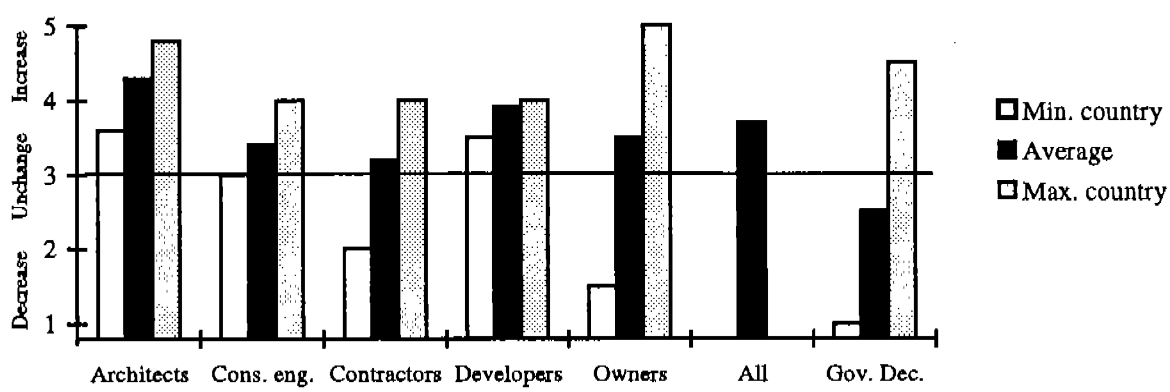

Figure 8. The interviewees' expectations on the future use of natural ventilation in offices.

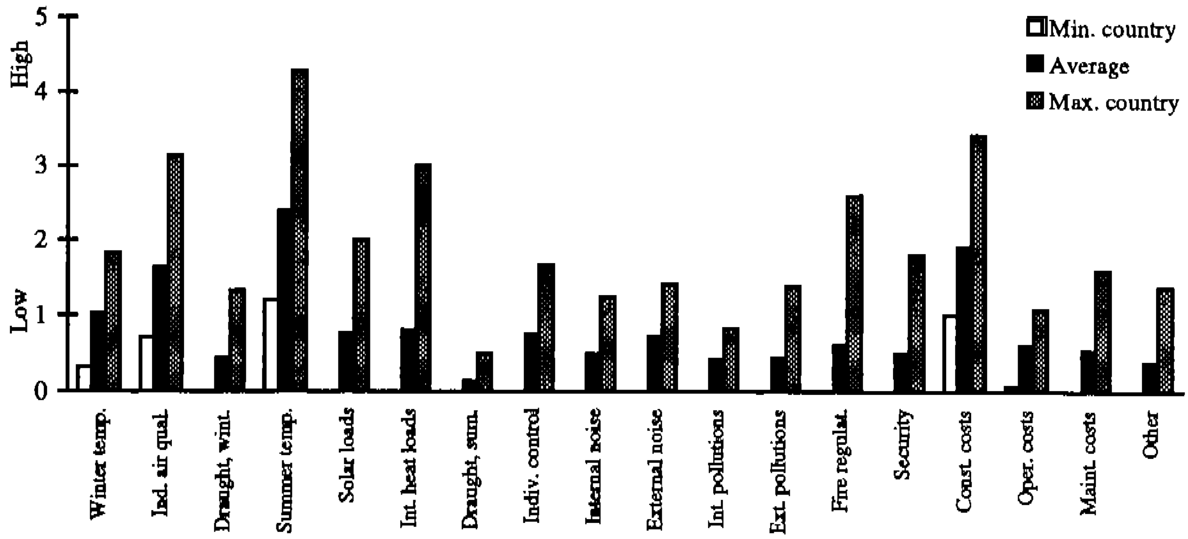

Figure 9. Critical parameters in the design of the buildings.

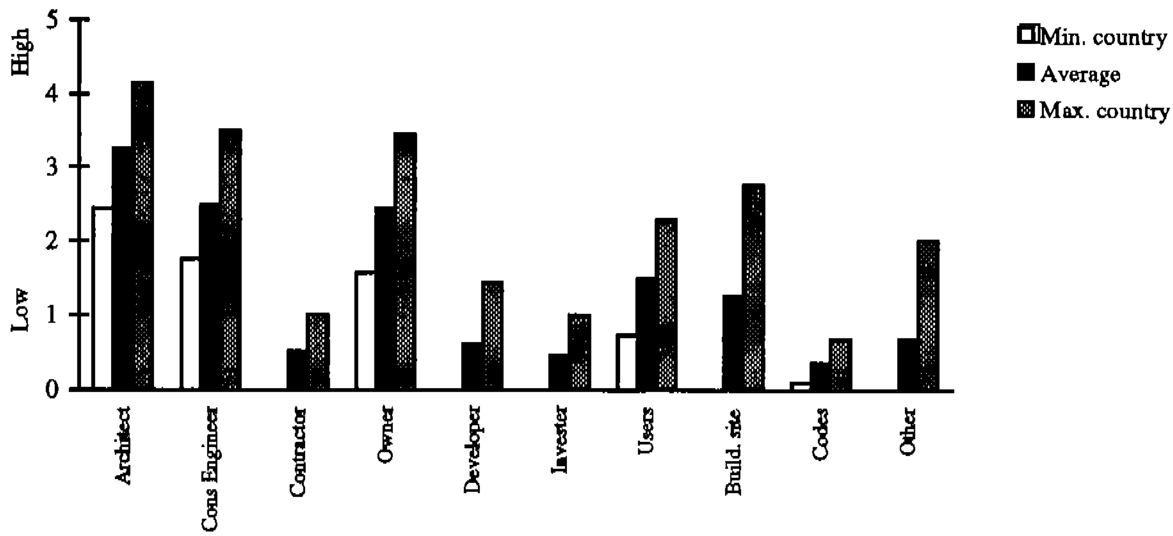

Figure 10. Influence on the design of the buildings.

3.10 Restricting requirements in codes

In Belgium, Norway and Sweden the interviewees perceived significant restrictions in building regulations and standards to the use of natural ventilation. In the other countries restrictions exist but they are perceived to be more limited. The governmental decision makers perceived the restrictions to be much more limited than the rest of the interviewees in a country.

3.11 Critical parameters

The interviewees perceived summer temperature, construction costs and indoor air quality to be the most critical design parameters in specific building projects including buildings with natural ventilation and buildings with mechanical ventilation in the offices, see figure 9.

3.12 Influence

The architects, the consultant engineers and the owners were the ones with the highest influence on the chosen design in the specific building projects, see figure 10.

4. Conclusions

On average the interviewees expect an increase in the future use of natural ventilation in office buildings. In general, the architects have the highest expectation of increasing use of natural ventilation. The interviews also identify significant lack of knowledge and experience of specially designed natural ventilation in office buildings compared to the knowledge and experience of mechanical ventilation. In addition there is a lack of source and information to natural ventilation knowledge in standards, guidelines and building studies. There is also a desire for new design tools on natural ventilation, including also calculation rules and easy to use, simple and advanced computer programmes.

- There is a need for good, standardised and generally acceptable natural ventilation system solutions and for more advanced solutions including heat recovery. In addition, there is a moderate need for new components regarding windows and vents with better air flow and draught performance, better controllability and better design.
- In the interviewees' perception, mechanical ventilation has several advantages compared to natural ventilation with regard to cooling effectiveness, draught minimisation, ability to remove odours and pollutants, ability to prevent ingress of odours and pollutants, insulation against external noise and central controllability, especially if the mechanical ventilation systems are well designed. Nevertheless the interviewees do not expect a higher user satisfaction in mechanical ventilated offices.
- Many interviewees expect higher installation, higher running and higher maintenance costs for mechanical ventilation in offices than for natural ventilation.
- Room temperatures in summer, indoor air quality and construction costs are the most important and critical design parameter. The architects, consultant engineers and owners have the biggest influence on the design of a building.
- Fee structures for design, and the liability of natural ventilation design in relation to lack of calculation rules, standards and guidelines causes problems for the use of natural ventilation in office buildings.
- Restrictions in the use of natural ventilation in office buildings placed by national building regulations and standards are limited, but problems can be caused by fire division requirements, and by guidelines about the need for mechanical ventilation in certain instants.

5. Recommendations

It is possibly, with further and continuing improvement of natural ventilation system concepts, components, controls and design tools, to encourage the wider uptake of natural ventilation in office buildings. This will also accelerate natural ventilation as a main design option in new and refurbished office buildings where good natural ventilation is sufficient to obtain comfortable indoor climate and good air quality with both high user satisfaction and low energy consumption, installation and maintenance costs.

- Simple, energy efficient, low cost natural ventilation system concepts for new and refurbished office buildings have to continue be developed and tested so that the use of natural ventilation in the majority of ordinary office buildings is not a technical and architectural challenge but a simple and well approved design solution.

* Standards and guidelines have to be improved for a better technical and legal background for the design of natural ventilated office buildings. The standards and guidelines should also include generally acceptable, simple and easy to use calculation rules for the design of natural ventilation.
* Simple design tools: diagrams or easy to use computer programmes have to continue to be developed that can be used in the early design process by architects, consultant engineers or design teams to analyse the advantages and disadvantages of different ventilation systems.
* The general knowledge on natural ventilation has to continue to be improved. Among architects, consultant engineers and possibly also contractors, the improved knowledge must come from basic education, post education, source books and building studies. For developers and owners the improved knowledge must be supplied by way of simple, easy to understand descriptions and examples.
* It may also be necessary to adjust the fee structure for the design of office buildings to reward the designers for the energy, indoor climate and total cost advantages of their design solutions and not for the amount of equipment installed in the building.

6. Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19th ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER, 1998

NATURAL VENTILATION IN OFFICE-TYPE BUILIDINGS – RESULTS AND CONCLUSIONS OF MONITORING ACTIVITIES

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SYNOPSIS

Since the beginning of this decade, natural ventilation in office buildings has been receiving specific interest. There are two sorts of application. Natural ventilation can be a strategy for indoor air quality control. It can also be used as night ventilation during warm or hot periods. In this case the objective is to cool down the thermal mass and improve the thermal summer comfort.
The EC JOULE NatVent project wanted to identify the barriers to the application of natural ventilation in office-type buildings in moderate and cold climates and to provide solutions. In the framework of the NatVent project, 19 naturally ventilated buildings across Europe (Belgium, Denmark, Great Britain, the Netherlands, Norway, Sweden and Switzerland) were monitored. This paper briefly presents the monitored buildings and the major findings.

1. THE NATVENT PROJECT

The NatVent project is a seven-nation EC JOULE project. The objective of the project is to overcome the barriers that prevent the application of natural ventilation in office-type buildings. It is intended for countries with low winter and moderate summer temperatures, where summer overheating from solar and internal gains can be significantly reduced by natural ventilation.

2. THE MONITORED BUILDINGS

In the framework of Task 2 of the NatVent project 19 naturally ventilated buildings in seven countries - Belgium, Denmark, Great Britain, the Netherlands, Norway, Sweden and Switzerland - were selected for detailed monitoring. The selected buildings are very diverse. Both existing as well as renovated and new buildings were studied. The objective of the monitoring was to identify the advantages and shortcomings of natural ventilation in ad-hoc buildings. The selected buildings were monitored during one winter and one summer period. Parameters such as temperature, humidity and ventilation rates were measured to identify the efficacy of the ventilation strategies.

[redacted]

Figure 1: List of the 19 monitored buildings

The results of the monitoring campaigns and an interactive presentation of each building, as well as all the other final products of NatVent are collected on the NatVent CD-rom. The major findings of the monitoring campaigns are given below in eight key messages.

3. KEY MESSAGES AND CONCLUSIONS

3.1. The key challenge is to achieve comfortable buildings that in addition are energy efficient

The indoor air quality in most of the 19 buildings is acceptable. The measured CO_2 levels are within the limits, which are generally accepted. However many buildings suffer from serious overheating problems. A high internal temperature is the most common user complaint in the buildings.
It is clear that indoor comfort should be the starting point of each building design. Due to the oil crisis in the seventies and the environmental concern nowadays, top priority is often incorrectly given to energy efficiency. This leads to energy efficient buildings with too high temperatures in summer, acoustical problems, poor indoor air quality, etc.
Priorities have to be reversed. The main objective is a good indoor climate: thermal comfort, indoor air quality, visual comfort and acoustical comfort.
Because of large internal gains thermal summer comfort is for most office-type buildings the main challenge. A comfortable indoor climate must be achieved by good building design and energy efficient installations. Furthermore it is crucial that the users are able to understand the concept of the building and are able to finance the required investments and running costs.

Figure 2: Global context for energy efficient buildings with good indoor climate

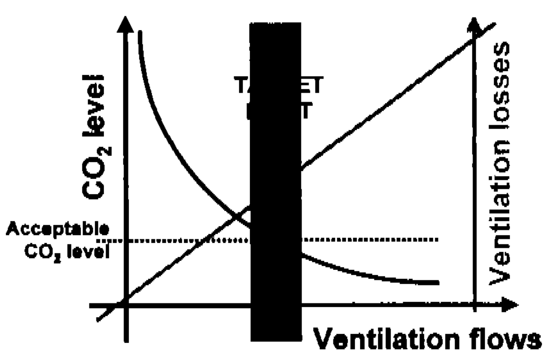
3.2. It is essential to understand the different meanings of natural ventilation

All 19 selected buildings were designated as ‘naturally ventilated’ buildings. However it was not always clear what was meant by ‘naturally ventilated’. This leads to confusion, which is probably not good for the reputation of natural ventilation and which may also indirectly lead to poor design concepts. Therefore, it is essential to clearly identify the meaning of ‘naturally ventilated building’.
Basically, there are three different definitions and concepts of natural ventilation:

- A design in which the air is assumed to enter and leave the building through unintentional openings. In the framework of the NatVent project, this type of air flow is not considered.
- A design in which the air quality is determined by a concept of supply and exhaust openings. Wind pressures and the stack effect are the driving forces for ventilation. This concept is called 'natural ventilation for Indoor Air Quality (IAQ) control'.
- A design in which during the summer a cooling effect is created by intensive ventilation with outside air. This concept is called 'natural ventilation for summer comfort'. Given the low temperatures at night time, intensive night ventilation is useful in moderate and cold climates.

3.3. Ventilation for IAQ is an optimisation of IAQ and energy efficiency

When applying natural ventilation during the heating season to control the IAQ, one has to avoid too large ventilation flows (see figure 3a). Large ventilation flows mean large energy losses. The energy impact can be considerable.
Several of the buildings monitored (e.g. DK2, NO2) where natural ventilation is used for IAQ control, have no advanced control strategy for the air flow rates. Although an acceptable indoor air quality is achieved, air flow rates are often substantially above the required air flow rates leading to unnecessary ventilation losses.
In a number of projects (BE1, BE2, BE3), natural ventilation is only used for summer comfort, while mechanical ventilation for IAQ control. It is evident that at present there is a wider range of products and systems available for mechanical ventilation systems to achieve an energy efficient ventilation: good control of the air flow rates, various systems of demand controlled ventilation, heat recovery, etc.
There is a clear need for new components and concepts to optimise and control the air flows in naturally ventilated buildings. In the framework of the NatVent project work was done on pressure controlled air inlets, heat recovery systems, etc.



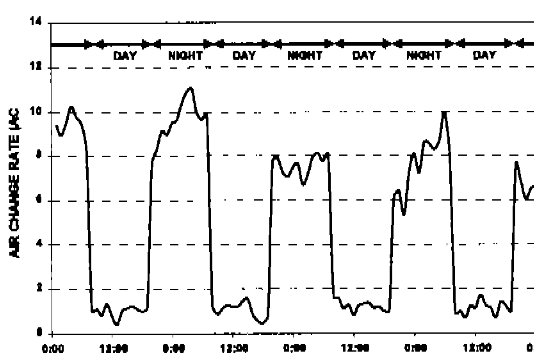


Figure 3a: Optimisation of ventilation flows
Figure 3b: Ventilation flows for summer comfort and IAQ

3.4. The required air flow rates for summer comfort are much higher than for IAQ control and the thermal mass of the building must be accessible

The concept of ventilation for summer comfort is based on two indispensable elements: firstly large air flows of cold external air and secondly accessible thermal mass.

A first target is to achieve the highest possible air flow rate without specific problems. Control of the air flow rates for summer comfort is in most cases not so critical as it is for ventilation for IAQ (see before). A possible problem can be a (slight) subcooling during the early morning hours leading to complaints by some of the building users. Automatic (central) control is in principle not required. An advantage of automatic control is the possibility of optimising the opening and closing of the intensive ventilation provisions (e.g.: only open when outdoor temperature is lower than indoor temperature). In case of automatic control, it is preferable that the users can overrule the automatic control during most part of the time.

As already indicated , most of the monitored case studies have no explicit splitting between devices aimed for IAQ control and those for summer comfort. In some cases, the designer made a clear separation between both strategies (e.g. in GB2). However, the monitoring results indicated that the control strategy was not clear to the users with the result that the required performances were not achieved.

The second target is to bring the cold external air in contact with the thermal mass. The optimal situation is a building with exposed heavy ceiling, floor and walls. In many cases this is not possible for reasons of cabling and flexibility. However less optimal solutions like only an exposed ceiling or an open false ceiling give acceptable results. The experience of several case studies has learnt that the accessibility of the thermal mass has a major influence on the building design in a very early stage. Furthermore it is often a barrier to the application of ventilation for summer comfort in existing buildings.

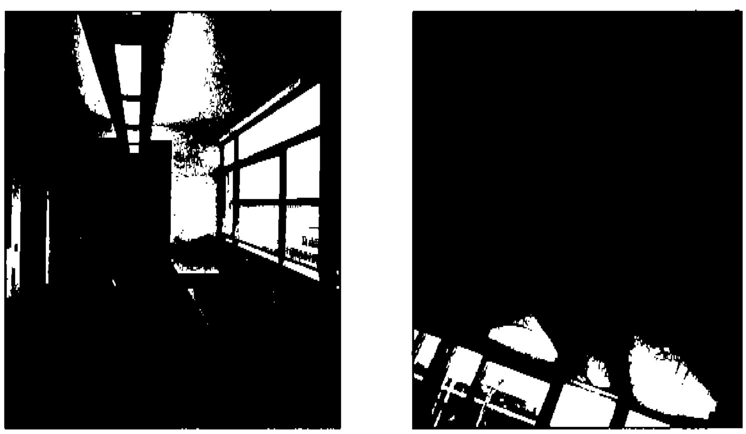


Figure 4: Exposed false ceilings (a. Canning Crescent Centre – b. The Environmental Building)

3.5. Summer comfort requires much more than intensive ventilation

Intensive ventilation for summer comfort can only be a successful strategy if the other elements of the building are also designed for summer comfort. An intelligent choice of glazing type, glazing areas, orientation and shading devices must control the direct solar gains through transparent surfaces. A high insulation level can limit the indirect solar gains through opaque surfaces. Energy efficient and well-controlled lighting systems minimise the internal gains. It is possible that under certain conditions (e.g. heavy internal loads or limited accessibility of thermal mass) the indoor temperatures are still too high. In this case limited active cooling by hygienic ventilation (also called 'top-cooling') can be an effective solution. Thermal simulation is an indispensable tool for the design of buildings with this global strategy. During the pre-design stage a simple, user-friendly tool (e.g. NiteCool, the Natvent program, etc...) gives an estimation of the overall effect of the different measures. At later stage, more elaborated simulations and powerful simulation programs (e.g. ESP-r, Capsol, etc...) may be used.

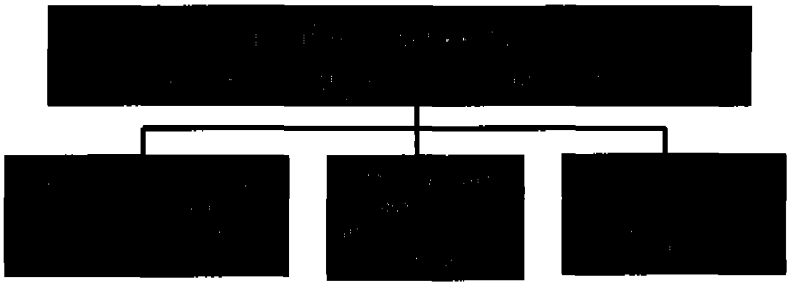


Figure 5: Thermal summer comfort – global strategy

3.6. Estimating the air flow rates is a small part of achieving a successful design

The real target is to realise buildings with a good indoor climate and low energy usage. This comprises the design of the building envelope and the installations. In case of naturally ventilated buildings, it is clear that the design of the natural ventilation is only a small part of the overall design challenge. As far as natural ventilation is concerned, it is evident that one should have a good sizing of inlets and exhausts in order to achieve the required level of ventilation. In addition for a good design it is important that a whole range of other requirements are achieved.

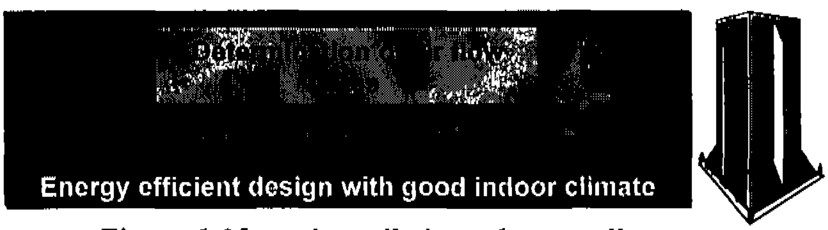


Figure 6: Natural ventilation - the overall context

A successful design can only be achieved if a number of potential barriers are considered and solved. In some of the 19 buildings studied, the natural ventilation operates unsatisfactorily due to incorrect sizing of inlets. However multiple are the design problems due to other reasons. As illustrated in Figure 7, there is a whole range of potential barriers for applying intensive night ventilation. Some of them are of a more technical nature (e.g. the local fire regulations require separation of the various parts of the building, acoustical requirements for buildings used at night time, etc.) others are more linked to the user (e.g. internal doors are closed at night for reasons of privacy, dust entering the offices at night, etc). It is also possible that non-technical reasons are an important barrier for application (e.g. the designer has to take a larger risk, the impact on the architecture is considered unacceptable, the fee structure for the consultants is not stimulating for such kind of studies).

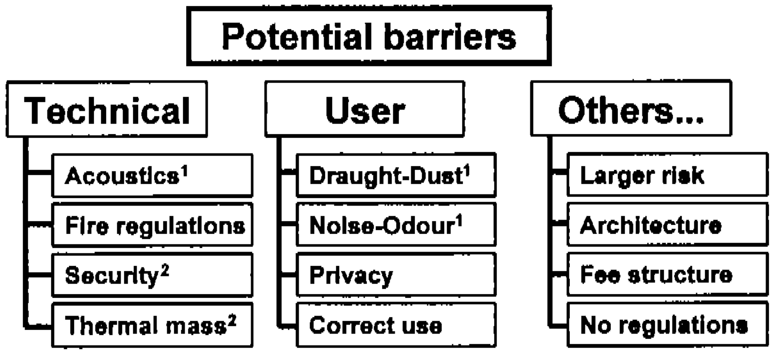


Figure 7: Overview of potential barriers for the application of natural ventilation for indoor air quality[1] and summer comfort[2]

3.7. The creation of an attractive environment for innovation is extremely important

A number of innovative and/or high performance products and systems relating to natural ventilation of office-type buildings are available on the market or are expected to come on the market during the next years. However, experience from other technological areas clearly shows that innovative and interesting technologies are not necessarily adopted by the decision makers (architects, building owners,etc). Lack of awareness is one possible reason, another reason is that the decision maker is not able to correctly interpret the performances of such systems and/or has doubts about the advantages of the technology. Such an evolution can only be avoided by large dissemination of the knowledge on natural ventilation and by performance-based standards for natural ventilation components, respectively.

3.8. Natural ventilation can be an attractive option, NOT the only option

It is clear that many of the monitored buildings would perform better with a well-designed mechanical ventilation system. Heat recovery and demand controlled air flows are some of the extra possibilities of mechanical ventilation.
However, a well-designed natural ventilation system could also lead to good performances. The NatVent project activities in Work Package 'Providing Technological Solutions' aimed primarily at indicating possible ways for optimisation of natural ventilation strategies. Air inlets with high acoustical performances, with constant air flow and with intelligent control as well as hybrid concepts with low-energy fans and heat recovery broaden the possibilities and the application area of natural ventilation.

Natural ventilation is not by definition the best option. Energy efficient designs with good indoor climate is the challenge and one should try to consider the best of all available options.

A Simple Interactive Design Tool for Sizing, Locating and determining Pollution Attenuation features, of Urban Air Inlets suitable for Office Buildings

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SYNOPSIS

This paper identifies successful ways of applying natural ventilation to non domestic buildings located in urban areas. Whilst noise and contaminant pollution sources are a problem methods of avoiding these emissions are discussed. A review of literature has established that pollution problems arise for buildings which are in close proximity to roads, railways, airports and local industries. Location of ventilation air inlets will affect the quality of indoor air, therefore it is essential that they are located in ways that minimise the ingress of external pollutants. Potential pollution avoidance strategies include, locating vents on sheltered facades and positioning central inlets at a sufficient height from emissions. Wind flows patterns around buildings have an important impact on air quality, and a simple model is discussed that determines the decrease in pollutant concentrations between emission sources and air intakes.

Adequate ventilation is required to limit the number of occasions when indoor temperatures are uncomfortable. A series of well established models are presented based on different natural ventilation concepts. These models can be used to size air inlets for any building, to provide specified ventilation rates on any floor. All issues discussed in the paper form part of an interactive design tool that provides best practice guidelines for minimising the impact of urban pollution, selecting suitable air inlets, and sizing them so as to provide adequate ventilation during the summer.

1. INTRODUCTION

The aim of this paper is to breakdown barriers to concepts of natural ventilation. The study is part of a Pan European project titled NatVent, that involves seven countries in the north of Europe. The project leaders are the UK Building Research establishment. Urban pollution is a major barrier to the adoption of natural ventilation, so successful ways of avoiding these problems need to be found.

The traditional approach to ventilating non domestic buildings located in urban areas is to specify mechanical ventilation. This strategy can seal buildings from pollution along facades, and where necessary draw air via cleaning filters to remove contaminants; the pressure drop associated with this process is not a practical option for passive ventilation. The draw back in relying upon air conditioning systems is in the amount of energy required to run them, hence the negative environmental impact. If natural ventilation systems are not adversely affected by external pollution then it offers an ideal alternative.

2. LIST OF SYMBOLS

NO_2	nitrogen dioxide.
NO	nitrogen monoxide.
CO	carbon monoxide.
SO_2	sulphur dioxide.
O_3	ozone.
D_{min}	minimum dilution factor at a fixed distance from an exhaust vent.
U_H	reference wind velocity (ms^{-1}) at height H.
Qe	volume flow rate of exhaust emissions (m^3s^{-1}).
r	distance between exhaust vent and air intake (m).
A	area of inlet (m^2).
Q	air flow rate (m^3s^{-1}).
C_d	discharge coefficient.
ρ_{ins}, ρ_{out}	air density inside and outside a building, respectively (kgm^{-3}).
g	acceleration due to gravity (ms^{-2}).
h, h_{NPL}	height of inlet and height of neutral pressure level, respectively (m).
T_{ins}, T_{out}	temperature inside and outside building, respectively (K).
v^2_{ref}	reference wind velocity (ms^{-1}).
$\triangle C_p$	difference in pressure coefficient between inlet and outlet.

3. SOURCES OF POLLUTION IN URBAN ENVIRONMENTS

Urban pollution arises from a range of sources, all of which should be considered when deciding upon the ventilation strategy for non domestic buildings. Pollution sources include local industries, cooling towers, building exhaust vents and traffic emissions arising from vehicles including aircraft and trains (1). Vehicles pollutants have the largest impact on ambient air quality. Particles (PM_{10}) and noise are the primary pollutants of concern, although other forms of pollution include the gases NO_2, NO, CO SO_2 and O_3. All sources should be identified prior to positioning air intakes on buildings.

Buildings in close proximity to busy roads are exposed to noise and contaminants. A recent investigation revealed that in one of two naturally ventilated buildings had 33% higher concentration of CO; this building was beside a busy road, whereas the 'cleaner' building was 400m away (2). Ambient pollution derived from vehicles emissions reflect traffic intensity and mobility, hence during rush hour periods when vehicles are stationary or congested, air quality will be at it poorest (3).

Aircraft and trains generate noise pollution, for buildings located near airports and railway stations. Emissions from building exhaust vents and industrial stacks may also negatively impact on air quality within buildings. Wind direction and speed are critical factors that will affect air quality at air intakes, (4).

4. MINIMISING THE IMPACT OF URBAN POLLUTION

A number of simple steps can be taken to reduce the impact of external pollution on air quality within non domestic buildings. These involve the intelligent location of air intakes to office blocks. Sheltered facades such as courtyards and enclosures are ideal for locating air inlets, as they are protected from pollutants derived from busy roads. Both contaminant pollutants and noise exposure are significantly reduced by this strategy (5). Buildings with central air inlets at high level are less exposed to pollutants generated at road level, particularly in the case of PM_{10} (6), and also in the case of gaseous pollutants such as CO and the oxides of N (7). Figure 1 is an example of the dilution of pollutants observed along a building facade situated besides a busy London road.

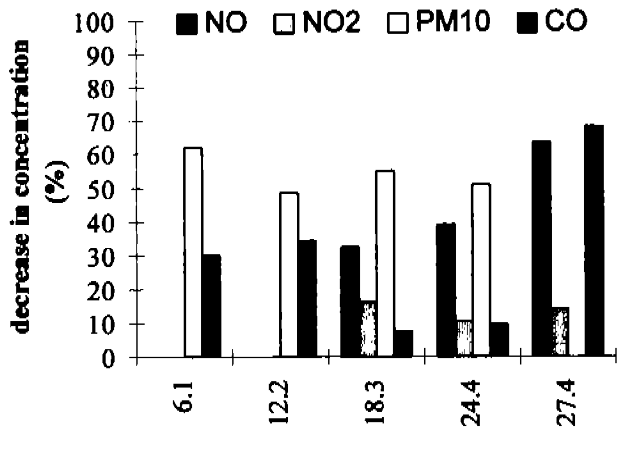

Figure 1. The effect of height on ambient concentrations of pollutants

Roof level installation of central air inlets may have some drawbacks if noise from planes is a local problem. If exhaust vents from host or neighbouring buildings are close to air intakes problems will arise. A simple model has been developed to evaluate the effect of exhaust vent emissions on air quality at air intakes (8). The model is defined by equation 1.

$$D_{\min} = 0.11\left(\frac{U_H r^2}{Q_e}\right) \quad 1$$

Although wind direction is not important in the model wind speed is. The value of U_H should closely reflect typical local conditions. Figure 2 indicated minimum dilution factors that have been calculated for a range of conditions.

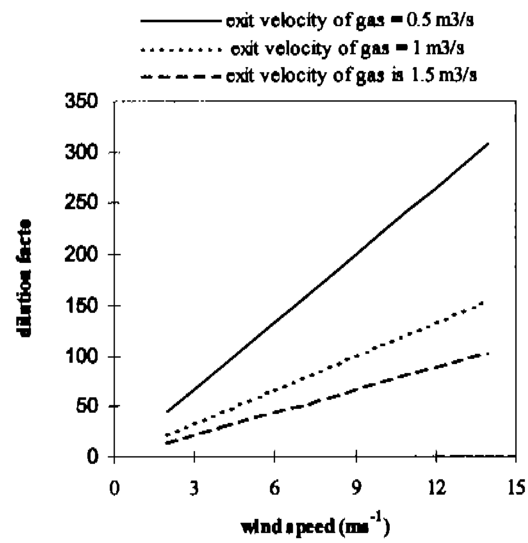

Figure 2. Dilution factors for gas emissions between exhaust vents and air intakes.

5. WIND FLOWS AROUND BUILDINGS

Buildings downwind of pollution sources are more exposed to contaminants than those upwind (4). However the situation is made complex by the way neighbouring buildings also affect air flow patterns. National meteorological wind flows are often not reflected on a local scale (9), so even buildings perceived to be downwind from pollution sources are subjected to re-ingestion of exhaust emissions. In a similar way relying on prevailing winds to avoid and / dilute exhaust emissions ignores the fact that significant sub prevailing winds may be derived from the opposite direction.

Wind forces acting on building generate leeward and windward facades as well as down-wash and up-wash zones (8). Air intakes and exhausts should be positioned on buildings so that are located in different zones. This will minimise the possibility of exhaust fumes re-entering the building. The size of the down-wash and up-wash zones depends on the size and shape of the building. A good design practice is to distance air intakes from exhaust vents by at least 1/3 rd the building height.

6. SUITABLE AIR INLETS FOR URBAN ENVIRONMENTS

It is not always possible to prevent outdoor pollution entering air inlets. To reduce the negative impact on indoor air quality air inlets will need to offer some means of attenuating pollution levels. A range of pollution attenuation strategies for air inlets is provided in Table 1. The aim of the design tool is to suggest suitable air inlets for buildings in relation to their environment. The tool is fully interactive, so allows the user to determine their inlet requirements.

Table 1. Air inlet pollution control strategies, suitable for non domestic buildings

type	pollution control strategy
1	inlets without pollution control features
2	inlets that can be closed during peak traffic periods
3	inlets with noise attenuation features alone
4	inlets with particle attenuation features alone
5	inlets with both noise and particle attenuation features

7. SIZING AIR INLETS

A minimum ventilation rate of 5 air changes per hour should ensure "sensible cooling" for most of the summer, in temperate climates (10). This requirement partly influences the size of air inlets suitable for non domestic buildings, as will the natural ventilation strategy adopted. Three models have been developed based on three approaches to natural ventilation (11). The models are based upon stack, wind and combined stack-wind driven ventilation. Equations 2 and 3 describe the models available in the design tool.

STACK DRIVEN VENTILATION:

$$A = \left(\frac{Q}{C_d} \sqrt{\left(\frac{2}{\rho_{ins}} \right) \rho_{ins}\, g\, (h - h_{NPL}) \left(\frac{T_{ins} - T_{out}}{T_{ins}} \right)} \right) \qquad 2$$

WIND DRIVEN VENTILATION:

$$A = \frac{Q}{Cd \sqrt{\left(\frac{2}{v^2{}_{ref} \Delta Cp} \right)}} \qquad 3$$

where;

$$Cp = \left(Cp\left(0.5\rho_{out}v^2{}_{ref}\right)_{inlet} \right) - \left(Cp\left(0.5\rho_{out}v^2{}_{ref}\right)_{outlet} \right) \qquad 4$$

The combined stack-wind ventilation model draws on both equations 2 and 3. The models can be used for buildings of any number of floors and each can have different ventilation rates if so required. The design temperature can be selected to reflect local meteorological conditions, the same applies to the choice of reference wind speed. Other data inputs relate to the dimensions of a building.

8. CONCLUSIONS

All issues that have been raised in this review are contained within the interactive design tool. Information can be accessed as a simple summary schematic, or if more detail is required, as a series of tables that address the full range of pollution issues associated with urban environments. The goal is to suggest the most suitable type of air inlet given the environment surrounding a building, and then to size them in order to provide adequate ventilation for most of the year. In appendix 1 the start options of the design tool are shown. These amount to the option of entering the full checklist by table format, or referring to the schematics that summarise all noise and contaminant pollution issues. Appendix 2 summarises the air inlet pollution attenuating options that are available, whilst appendix 3 outlines data inputs and outputs relevant to sizing inlets.

9. ACKNOWLEDGEMENTS

This work is part of the Pan-European NatVent™ project co-ordinated by BRE with the participation of Belgian Building Research Institute (Belgium), TNO Building & Construction Research (The Netherlands), Danish Building Research Institute SBI (Denmark), J&W Consulting Engineers AB (Sweden), Willan Building Services (UK), Sulzer Infra Lab (Switzerland), Deflt University of Technology (The Netherlands) and Norwegian Building Research Institute (Norway). The UK participation in the project is funded in part by the European Commission under the JOULE-3 programme and the Department of the Environment under the Partners in Technology (PiT) Programme.

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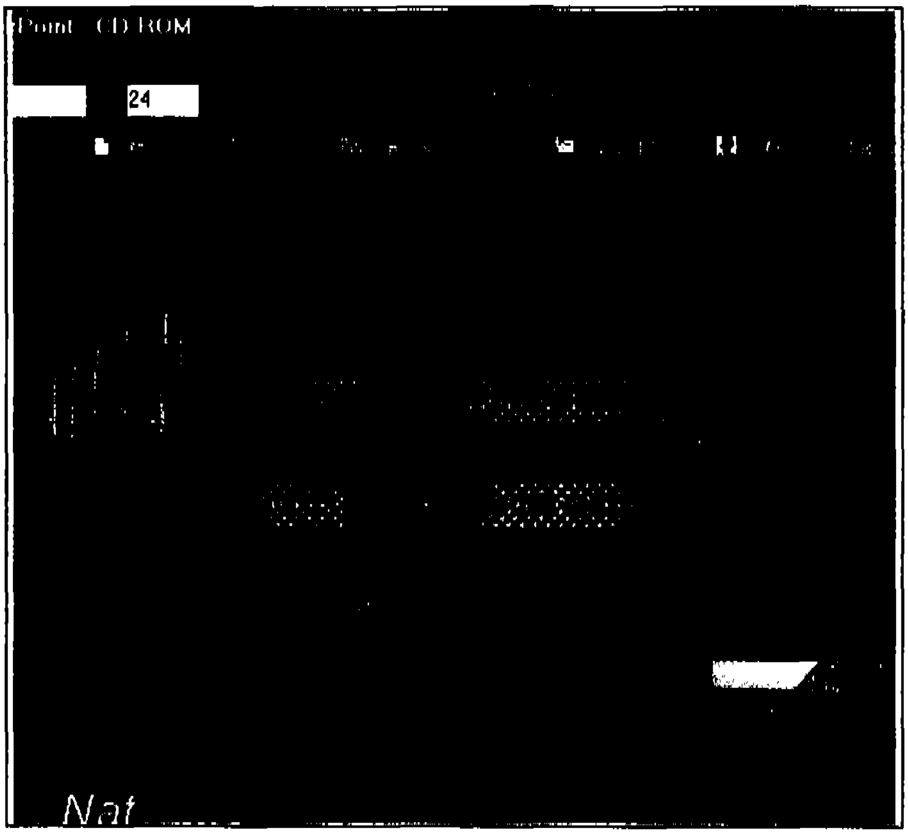
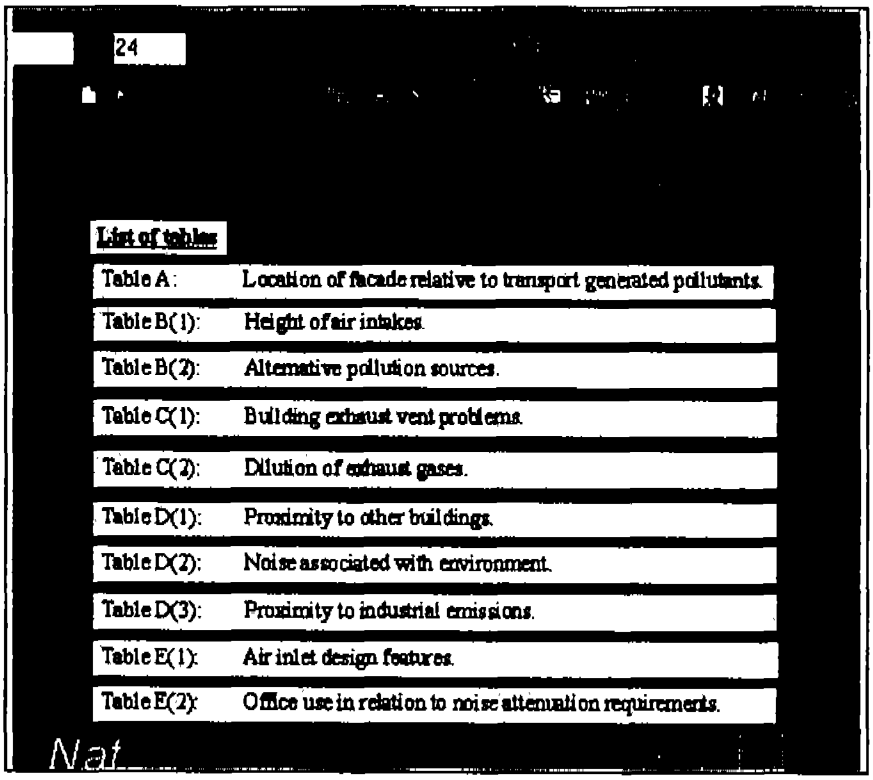
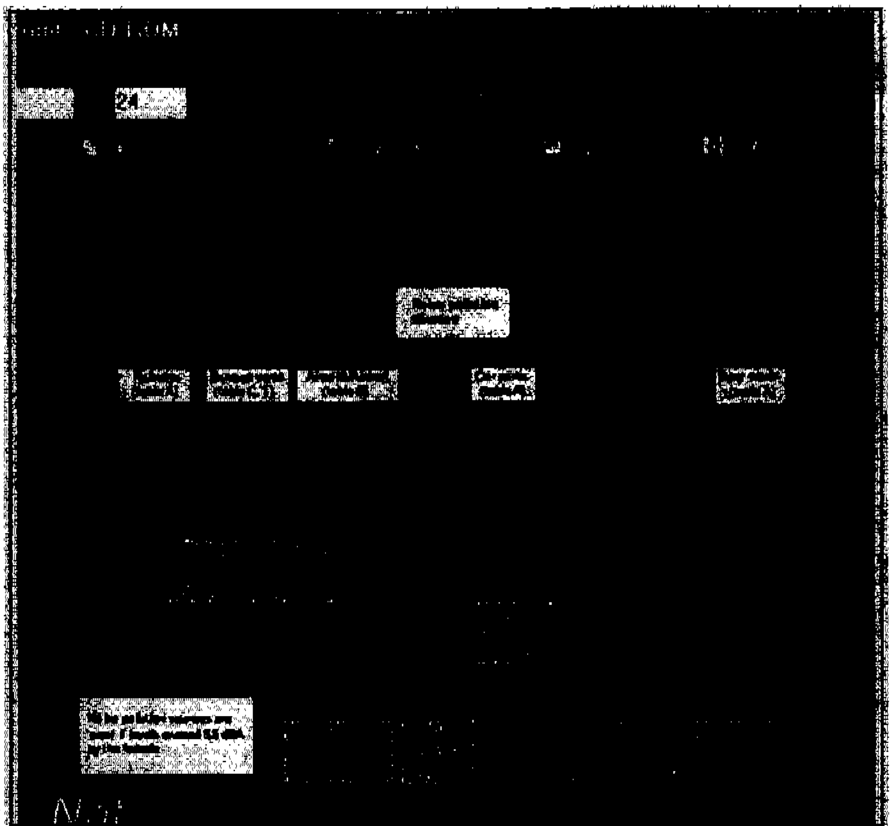
(10) **Irving, S.**, (1996); '*The Role of Ventilation in Cooling Non-Domestic Buildings*', Technical Note 48, AIVC, p.1.

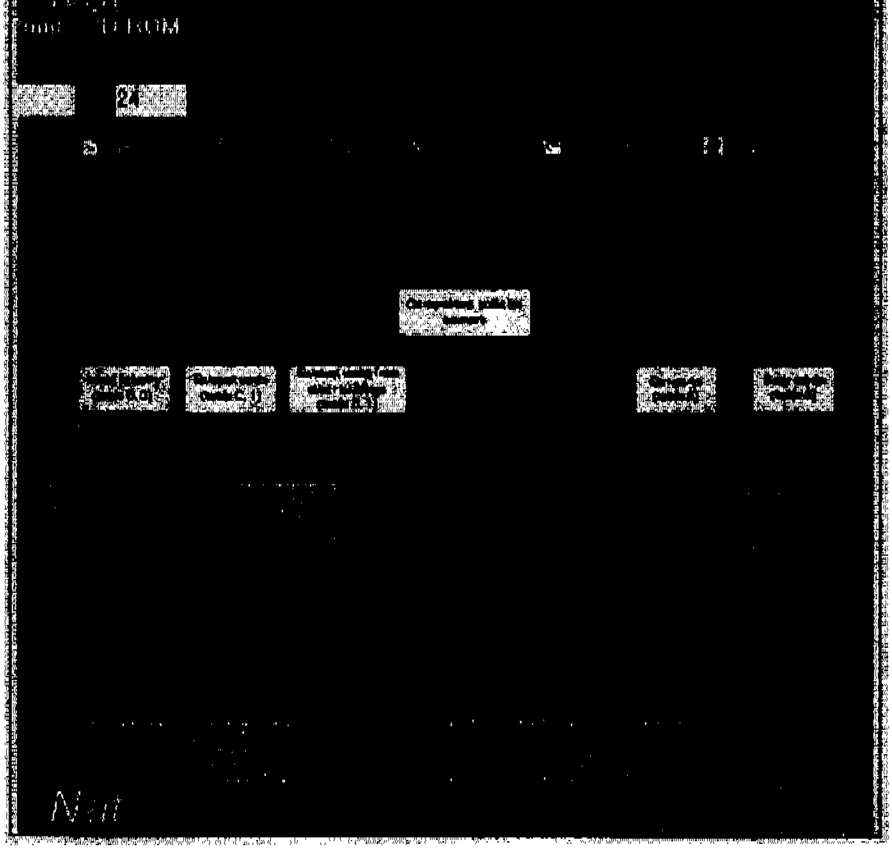
(11) **Irving, S., et al.**, (1997); '*Natural Ventilation in Non-Domestic Buildings*', CIBSE Design and Applications Manual, pp. 55-57.

11. APPENDIX 1: Front end of interactive design tool, for locating and sizing air inlets









APPENDIX 2: Air inlet options suitable for naturally ventilated buildings in urban areas

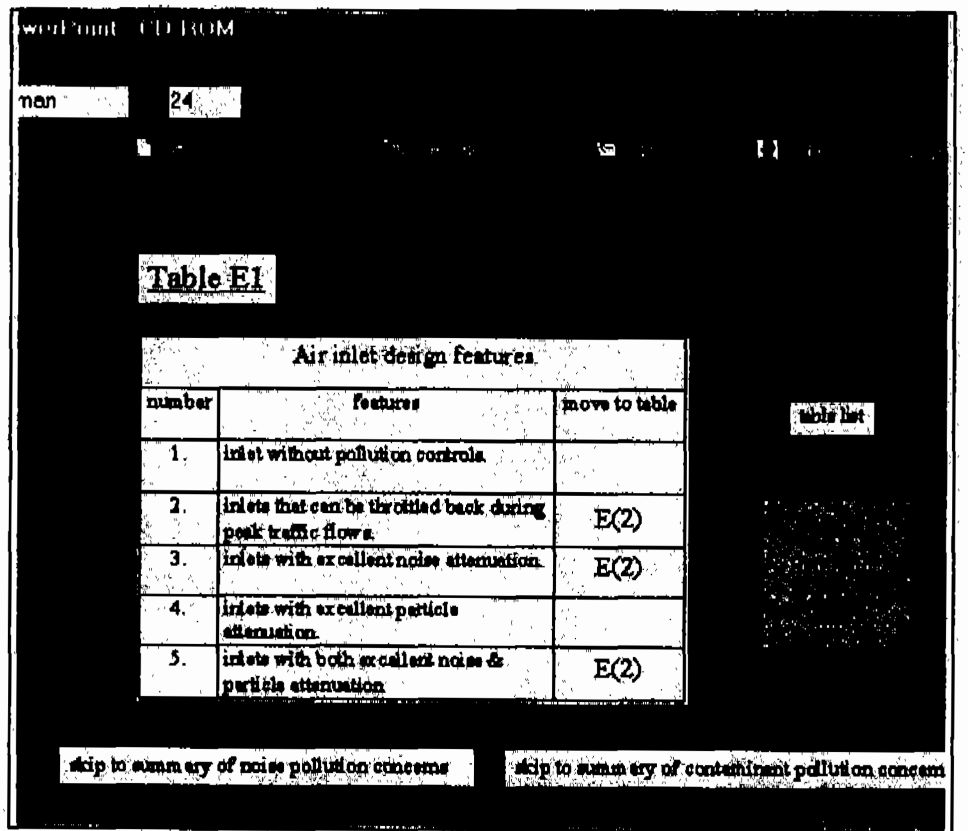


Table E1

Air inlet design features		
number	features	move to table
1.	inlet without pollution controls.	
2.	inlets that can be throttled back during peak traffic flows.	E(2)
3.	inlets with excellent noise attenuation.	E(2)
4.	inlets with excellent particle attenuation.	
5.	inlets with both excellent noise & particle attenuation	E(2)

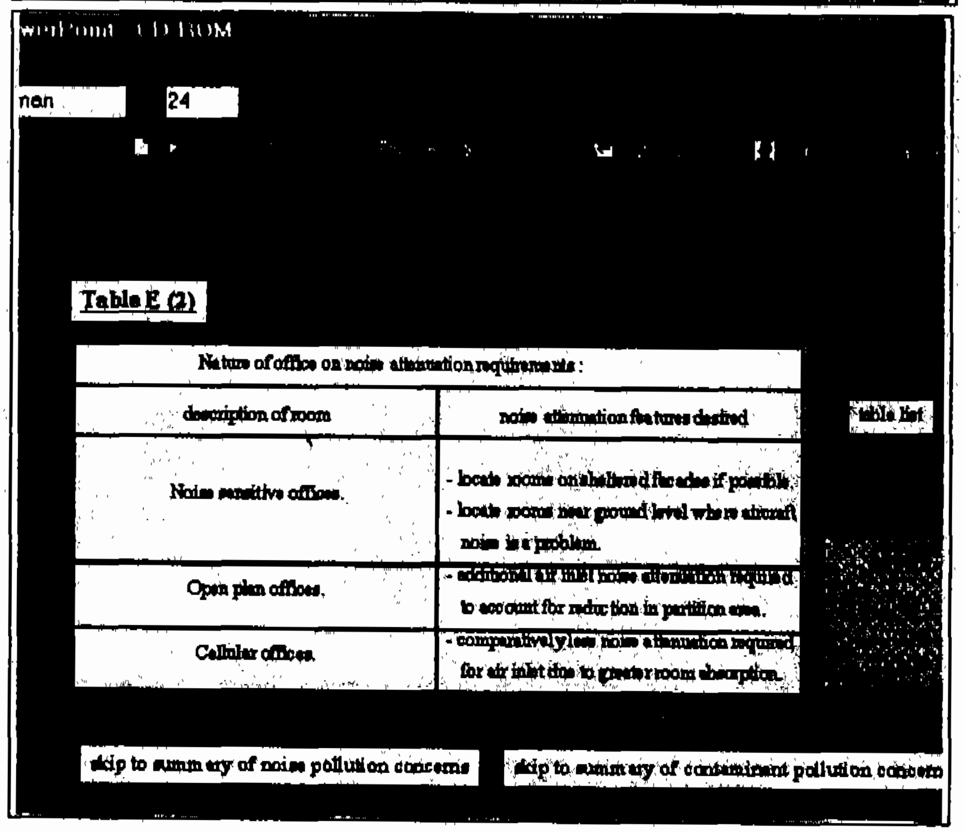
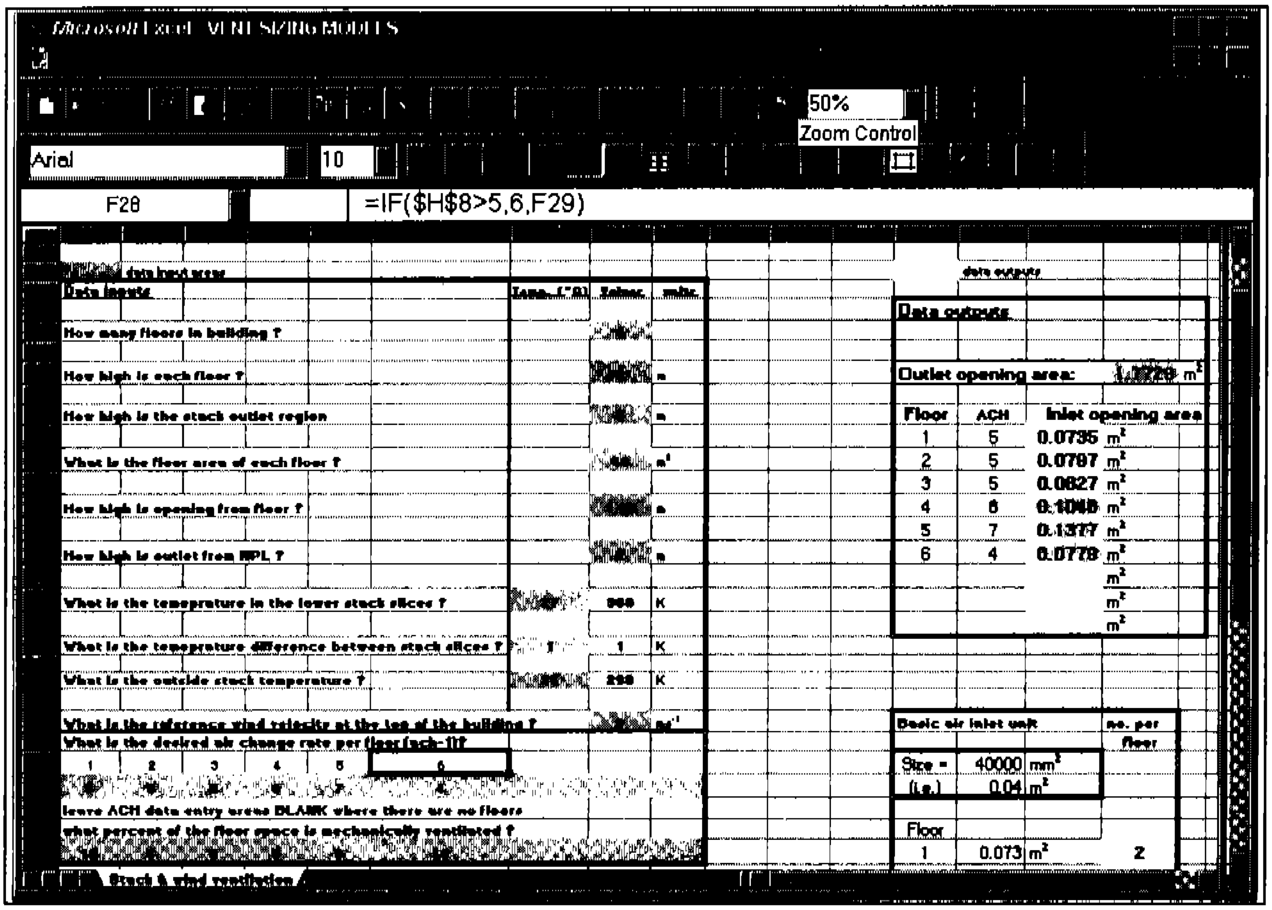


Table E (2)

Nature of office on noise attenuation requirements :	
description of room	noise attenuation features desired
Noise sensitive offices.	- locate rooms on sheltered facades if possible. - locate rooms near ground level where aircraft noise is a problem.
Open plan offices.	- additional air inlet noise attenuation required to account for reduction in partition area.
Cellular offices.	- comparatively less noise attenuation required for air inlet due to greater room absorption.

APPENDIX 3: Data inputs and outputs of the combined wind-stack ventilation model



VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

THE APPLICATION OF CONTROLLED AIR FLOW INLETS

Willem de Gids
TNO Building and Construction Research

ABSTRACT

The application of controlled air flow inlets

In the EU Joule project NatVent one of the work packages was dealing with controlled air flow inlets. During the last conference in Greece and overview was presented on availability, performances and application of controlled air flow inlets. At the presented poster an interactive IAQ computer tool was demonstrated. This tool has been improved and is now available.

Some participating countries in the NatVent project have carried out special tests with the NatVent IAQ tool. The NatVent Participants were asked to design a natural ventilation system according to their national requirements. A second run was asked for a ventilation system which could read the 1000 ppm CO_2 requirement. The tests of the various countries will be shown. The results of this exercise are very interesting. It shows the positive effect of controlled inlets on IAQ in case of natural ventilation in offices.

Please Note: The full paper will be published in the Supplement to the Proceedings.

Heat recovery
In Natural Ventilation Design
of Office Buildings

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Non Nuclear Energy Programme

1. INTRODUCTION

Heat recovery in ventilation systems for office buildings in cold climates is necessary for two reasons:
1. To obtain acceptable indoor thermal comfort by preheating of fresh air,
2. To reduce ventilation energy loss

This paper describes a pilot system built in the laboratory of the Norwegian Building Research Institute, NBI, based upon the concept of an advanced fan assisted natural ventilation system with heat recovery. The concept was developed by NBI. The objective of making the pilot system was to find out how a real system based upon this concept works and to supply the NatVent project with measuring data both from a winter period and a summer period. The concept is new and is not yet tested in a real office type building.

This paper gives a brief description of the components and the system. The paper also summarises the measurements of the resistance in the system, the driving forces, the temperatures and the airflow.

2. DESCRIPTION OF THE PILOT SYSTEM

2.1. General description

The system is designed to make use of natural driving forces (thermal buoyancy and wind). All components in the system are designed to give a low pressure drop. The system is equipped with an energy efficient assisting fan, electrostatic air filters and a "run-around" heat exchanger.

The build up of the system is shown in fig.2.1 (drawing and photo). The pilot system represents a full scale part of a ventilation system, for instance for one wing of an office building with 3 - 4 storeys, with a ventilation capacity of 400 l/s, i.e. for about 40 persons.

The system consists of three major parts:
1. Roof unit with heat exchanger, coarse filter, assisting fan, a wind-boosted air exhaust unit and a wind-boosted air intake unit. See fig.2.2.
2. Floor unit with electrostatic filter, heat exchanger, space for optional fan (no.2). See fig.2.3.
3. Duct system. Exhaust air ducting with exhaust terminals in three levels. Supply air ducting at three levels with supply terminals (simulated with dampers). As a part of the supply system an insulated vertical main duct connects the air intake of the roof unit with the floor unit.

The system design is described in detail in /1/ and /2/.

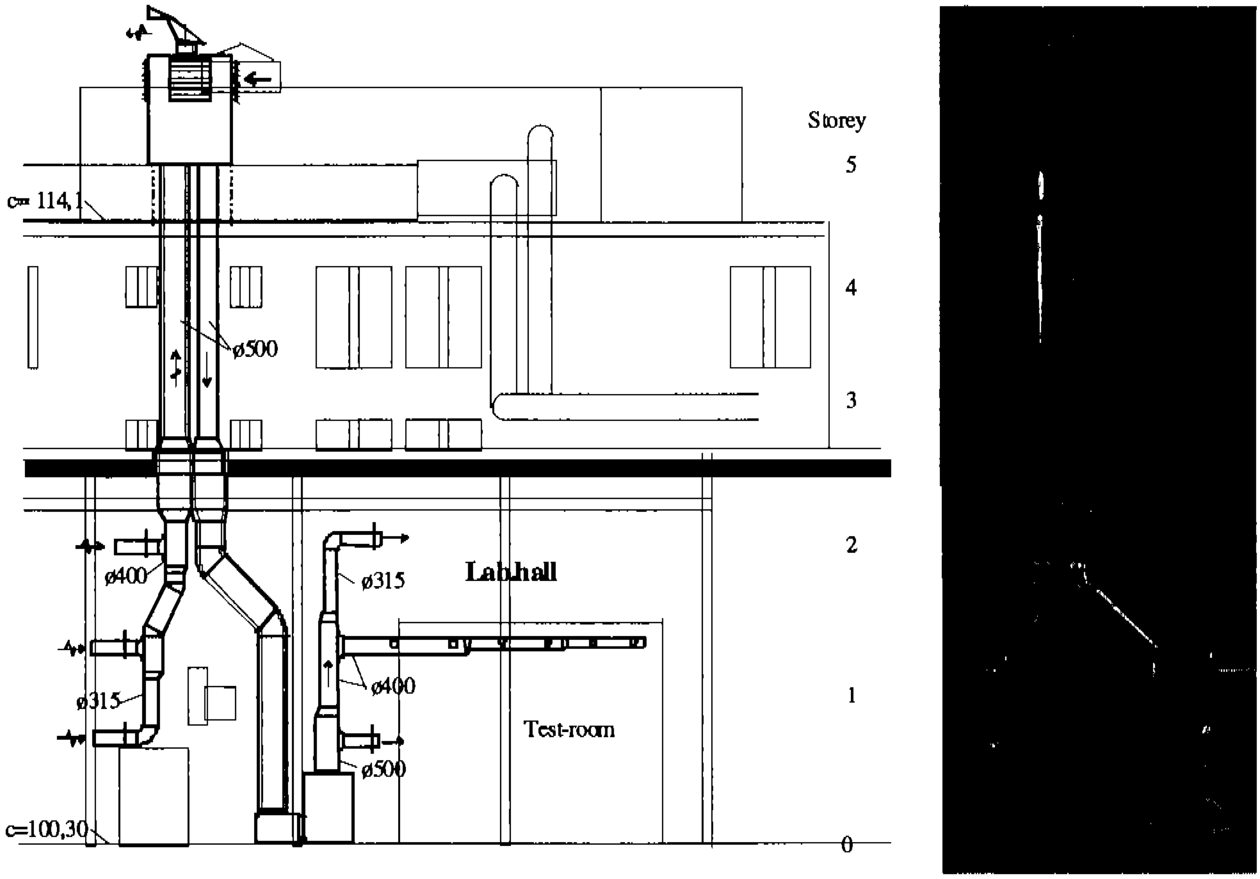

Fig. 2.1. Lab test set up overview

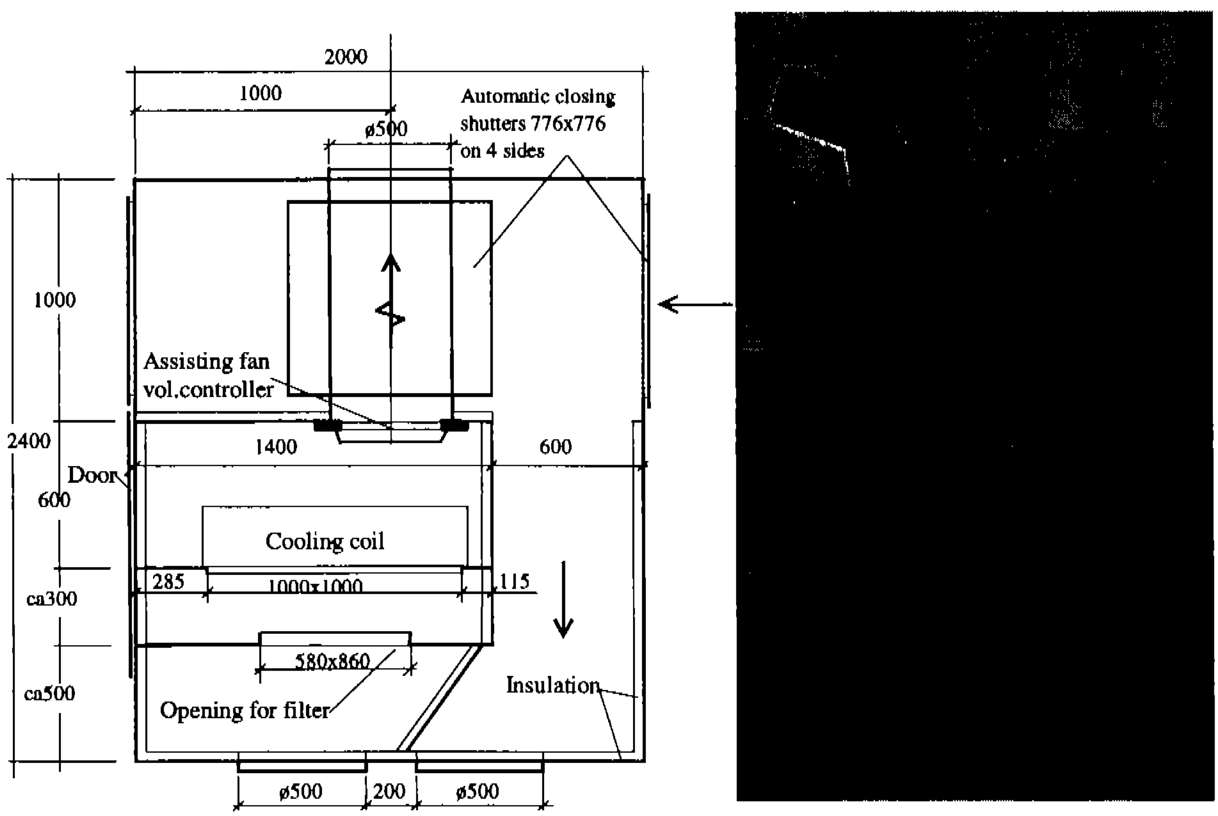

Fig. 2.2. Roof unit

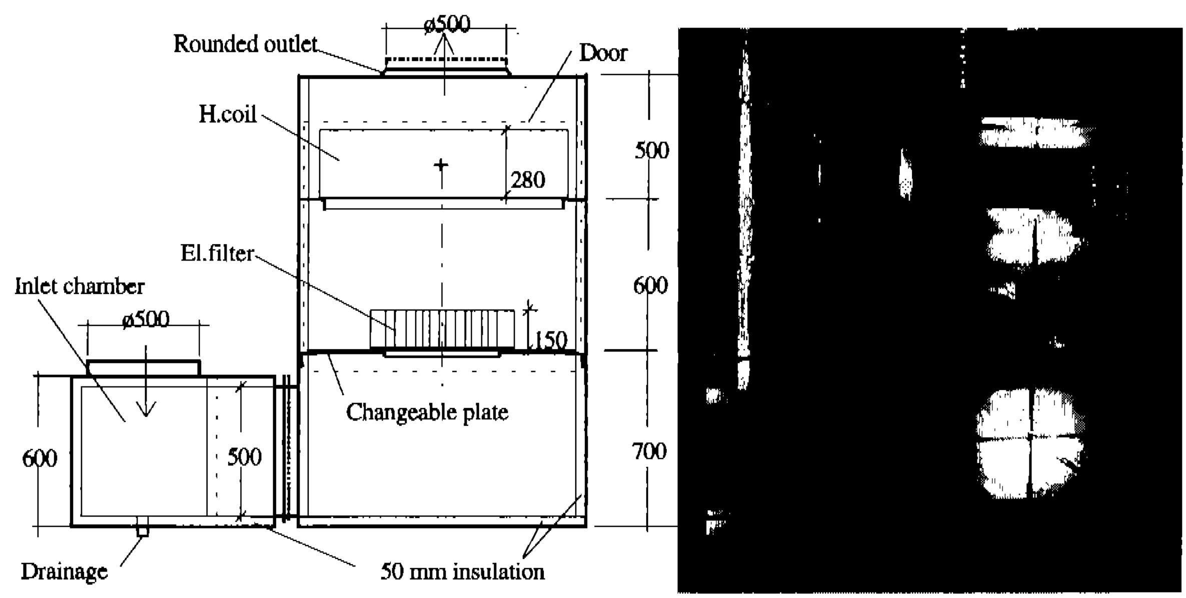

Fig. 2.3. Floor unit (Indoor)

2.2. Pressure drops in the system

Dimensions and flow cross sections in a natural ventilation system must be chosen in order to keep the pressure losses as low as possible. Air velocities must be chosen to comply with the pressure loss requirements. The magnitude of the dynamic pressure ahead of components producing high pressure losses should be between 0,4 and 4 Pa (0,8 - 2,6 m/s air velocity), with the highest velocities in the supply terminals and exhaust openings. The lowest velocities should be applied where high resistance coefficients are expected such as sudden area enlargements, heat exchangers, filters, bends etc. The pressure drop in straight ducts can have an order of magnitude value of 0,15 Pa/m duct length, resulting in an air velocity of 1 m/s in Ø125 mm ducts increasing to 2 m/s in Ø400 mm ducts and even 4 m/s in Ø1000 mm ducts. Aerodynamically good shapes should be applied to avoid sudden area changes and sharp bends.

The pilot system is designed with approximately 2 m/s in the main, vertical ducts and 1 m/s in the smaller, horizontal ducts. This gives approximately designed pressure drops as listed in table 2.1 at nominal flow, 400 l/s:

Table 2.1. Design pressure drop for pilot system

Component	Supply system Δp [Pa]	Exhaust system Δp [Pa]
Air intake grille / exhaust wind vane	4	5
Heat exchangers	6	6
Filters	2	2
Air terminals	6	6
Ducts, bends, take-offs, etc	10	6
Sum	28	25
Total for supply and extract	53	

In addition to the pressure drops in table 2.1 the assisting fan gives about 4 Pa when it is turned off.

The pressure drops in the pilot system were also measured. Corrected pressure drop recalculated to be valid for a flow rate of 400 l/s are shown in table 2.2, supply and table 2.3, exhaust.

Table 2.2. Corrected pressure drop (loss). Supply

Component	Corrected for stack effect, Pa	Corrected for flow rate, Pa
Intake and vertical duct	10,5	11,3
Filter	1	1
Heat exchanger	6	6,5
Supply ducting and terminals	18	19,4
Total sum	35,5	38,2

Table 2.3. Corrected pressure drop (loss). Exhaust

Component	Corrected for stack effect, Pa	Corrected for flow rate, Pa
Outlet to atmosphere	2	1,2
Heat exchanger	5	3,1
Exhaust ducting and terminals	11,7	8,4
Total sum	20,7	12,7

Total system pressure drop, supply and exhaust: =50,9 Pa

The total pressure drop is quite near the design pressure drop, but the distribution between supply and exhaust is different.
The pressure drop in the supply system is higher than necessary due to a not optimised design and adaptation to the existing building.

2.3. Costs

The installation cost for the pilot system and anticipated cost for a similar future system with the volume flow of 0,4 m^3/s are calculated. The yearly running cost for such a ventilation system is predicted taken the results from the measurements of the energy consumption into account. The same procedure is used for a traditional ventilation system by obtaining the installation cost from a ventilation entrepreneur company. The result from this comparison was that the installation cost for the pilot system was slightly higher than a traditional system but the running cost was lower in spite of the low heat recovery efficiency in the pilot system. A comparison of the "present value" based upon 20 years of operation showed approximately the same value for the two systems.
There are many uncertainties associated with these calculations e.g. cost of a comparable traditional system, future electricity price, present contra future price of the pilot system, cost of the "lost floor area" etc. The performed calculation shows however that a pilot system with heat recovery as presented here is competitive.

3. MEASUREMENTS

3.1 Temperatures, wind and air flow in the system

The winter in Oslo was extremely mild in February-98, but seasonally cold in March and April. After an initial measuring period in February, continuous logging has occurred since the last week of February. In the beginning different system modes were tested during some days: Natural ventilation without fan, natural ventilation with extract fan mounted but turned off and at last natural ventilation with assisting extract fan and regulation of fan speed.

Fig. 3.1 shows some selected data series from the first period, before the fan was installed.

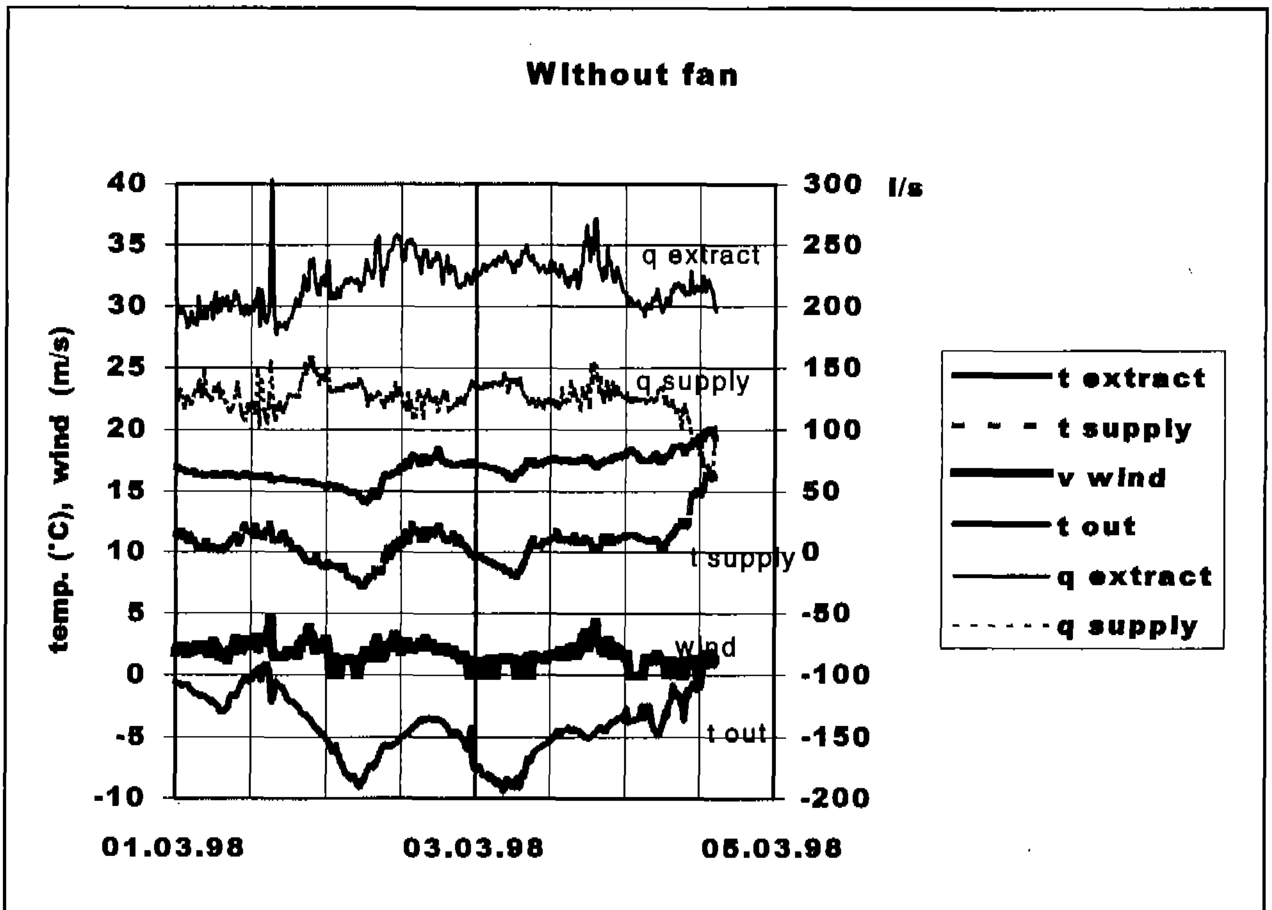

Fig. 3.1. Measurement results without assisting fan installed

The outdoor temperature in this period varied from -10°C to 0°C, and the wind speed from 0 to 5 m/s, (wind speed lower than 1 m/s can not be measured and is registered as 0). The extract air flow was around 60% of design value (~240 l/s) and the supply air flow around 33% of design value (~133 l/s). The heat recovery system increased the supply temperature about 15°.

Fig. 3.2 shows the same data series from a period with the extract fan mounted, but without electric connection, i.e. the fan is rotating because off the natural air flow.

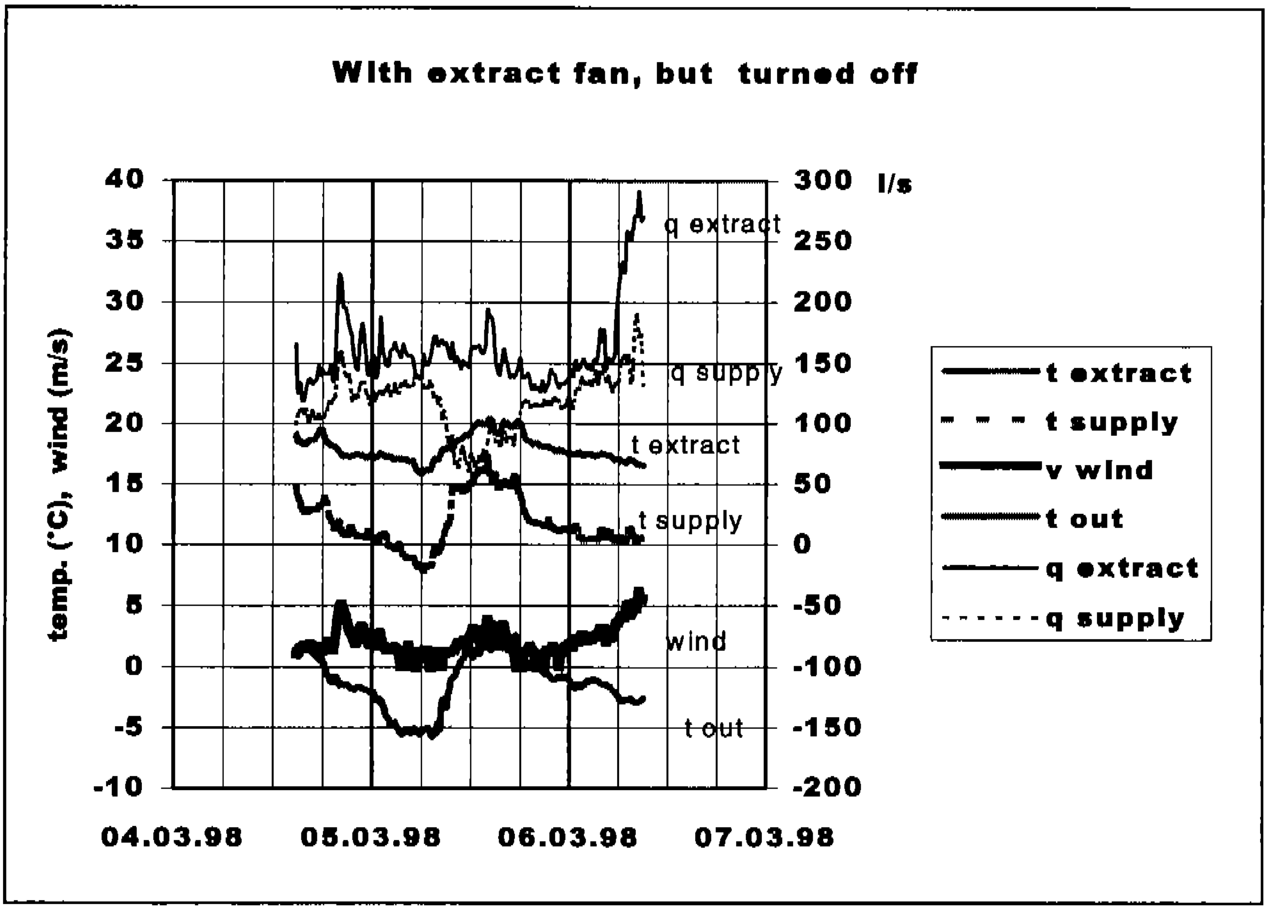

Fig. 3.2. Measurement results with installed assisting fan, but without electric connection

The outdoor temperature and wind in this period were about the same as in the period without fan. The extract air flow is now reduced to about 43% of design value (~170 l/s) and the supply air flow is about the same as before. There is a clear connection between the air flows and outdoor temperature and wind. The air flows increase considerably when the wind speed exceeds 5 m/s.

Fig. 3.3 shows the result of connecting the fan and fan control system. The extract air flow is now quite stable at design value (400 l/s), regardless of variations in wind and outdoor temperature. The supply air flow still varies with the natural driving force, but has increased to about 38% of design value (~150 l/s), due to increased extraction flow.

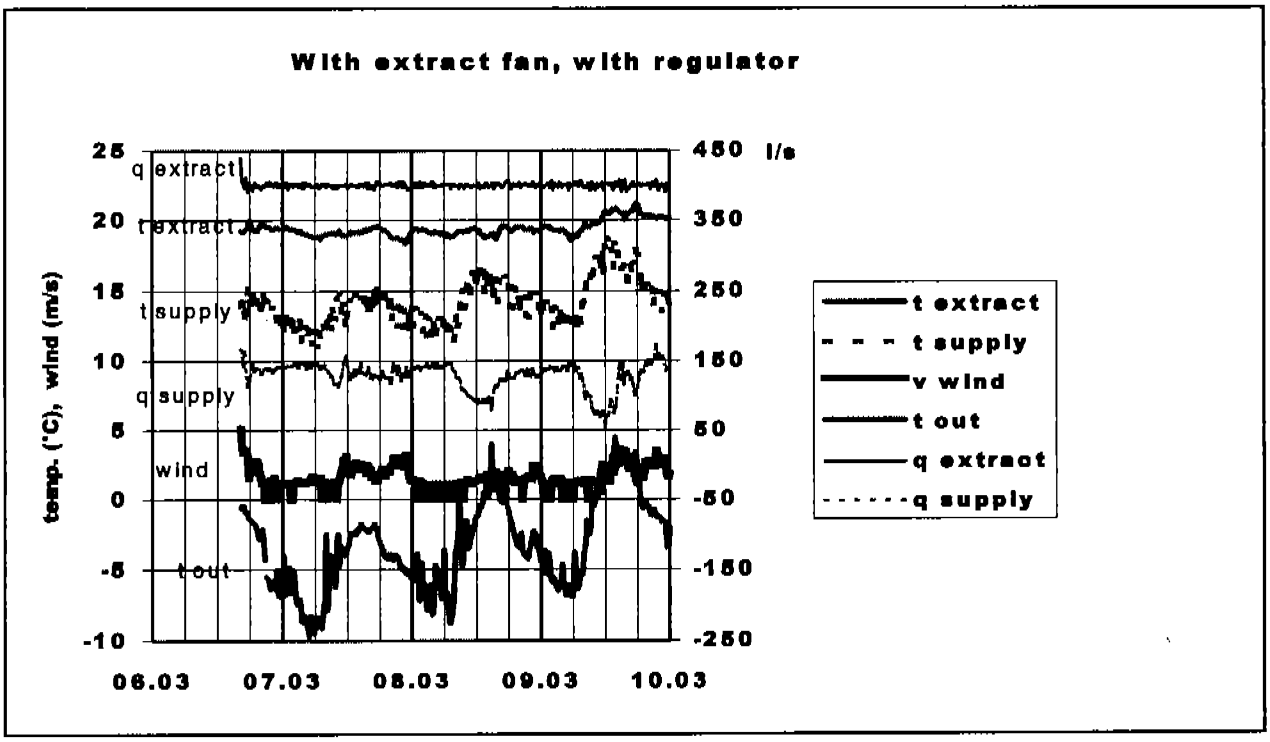

Fig. 3.3. Measurement results with assisting fan and with speed regulator

3.2. Heat exchanger efficiency

The heat exchanger efficiency is calculated from measurements in a period with an auxiliary supply fan installed, fig. 3.4. In the days from April 3. to April 6. the supply flow was kept reasonably constant at design flow (400 l/s). (The auxiliary fan had no automatic flow controller).

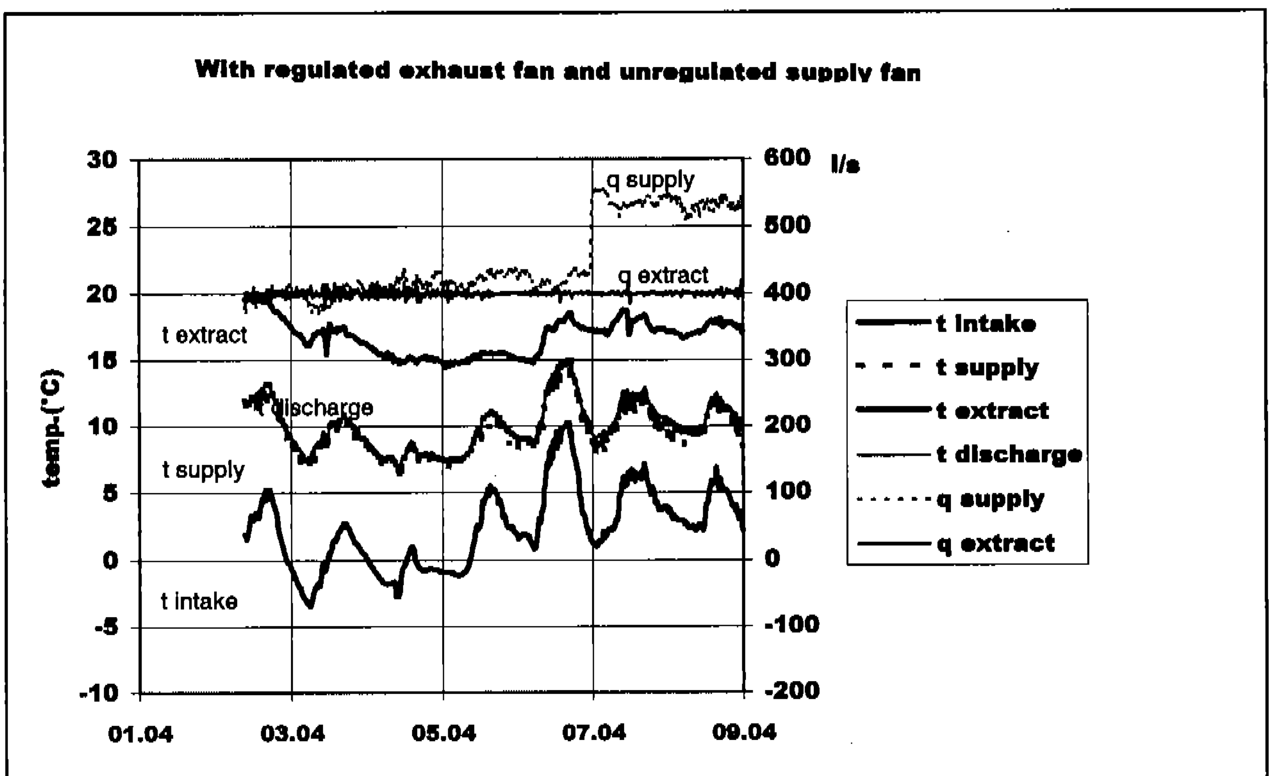

Fig. 3.4. Measuring results for calculation of heat exchanger efficiency

The heat exchanger efficiency is calculated with data from 04.04.98 at 06.59:
Suppl.=7,63°C, intake=-1,71°C, extr.=16,83°C *), disch.=7,73°C, q_{supl}=400 l/s, q_{extr}=405 l/s.

$\eta_{suppl} = (q_{suppl}/q_{extr}) \cdot ((t_{suppl}-t_{int})/(t_{extr}-t_{int}) = (400/405) \cdot ((7{,}63- -1{,}71)/(16{,}83- -1{,}71) = 0{,}50$
$\eta_{extr} = (q_{extr}/q_{suppl}) \cdot ((t_{extr}-t_{disch})/(t_{extr}-t_{int}) = (405/400) \cdot ((16{,}83- 7{,}73)/(16{,}83- -1{,}71) = 0{,}50$

The measured heat exchanger efficiency is lower than the producer's design value, =0,58. The reason may be uneven flow over the coils, specially on the extract side, because of too little filter opening in front of the coil. This will be changed if the measurements continue in a new project.

3.3 Recording of fan power

The electric energy consumed by the assisting fan is measured with a kWh-meter and the data collected in a data logger. Because of the relatively high measuring resolution of the kWh-meter, the average values over periods as short as 15 min are not well suited for calculation of actual fan power. Instead we have calculated the fan power from the 15-min average of fan voltage, which is proportional to the fan speed. The connection between fan power and fan voltage at 400 l/s is determined from laboratory tests, see appendix 1:
$P\,(W) = 0{,}375 \cdot U\,(V)^3 + 9$ (including the regulator)

Fig. 3.5 shows how the fan power consumption varies with the natural driving force in the exhaust system. The driving height is here set to16 m and the wind coefficient equal to 0,6. The figure contains measured 15 min data from the period March 6. to April 23.

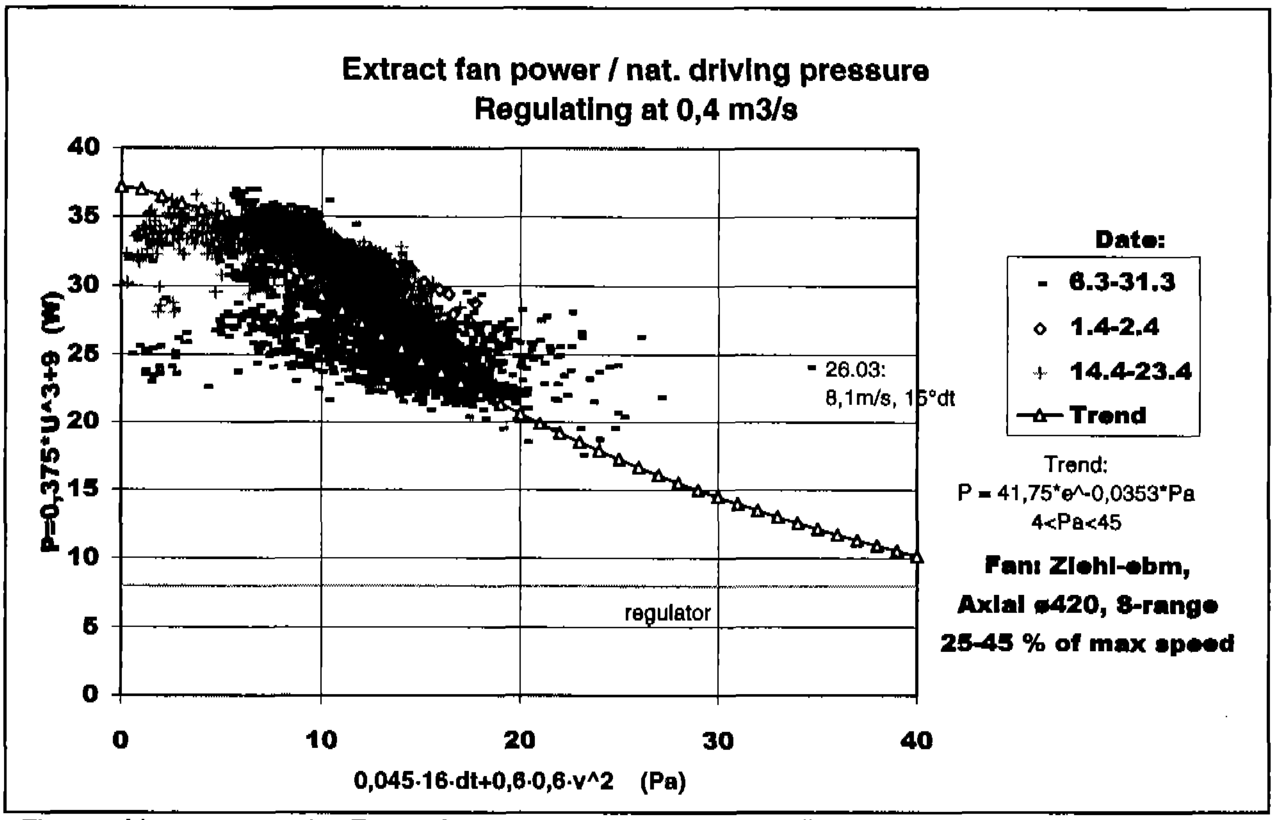

Fig. 3.5. Measuring results: Extract fan power consumption depending on natural driving pressure

The measured values in fig. 3.5 are quite spread out, mainly because of the 15 min averaging. There is no momentary connection between air flow, fan voltage, temperature difference and wind.

However, the trend is quite clear: The power consumption of the extract fan varied between 37 W and 18 W, depending on the natural driving pressure which varied from 0 to 23 Pa for this period.

4. DISCUSSION

In cold climates heat recovery in ventilation systems is necessary both for achieving thermal comfort and for saving energy. This is also the case with natural ventilation. Because of the low natural driving forces it is necessary to use assisting fan to get sufficient flow through the heat recovery system. A test system based on this principle was built in the laboratory at NBI. The objective was to get experience with such a system and to supply the NatVent project with measuring data.

The wind boosted air intake shutters and exhaust opening with wind vane improved the natural driving forces and thus reduced the fan energy consumption. This is demonstrated by the continuous measurement of fan power consumption, which varied with wind and temperature difference. In the recorded spring period the average fan power was about 75 % of the power needed when the driving forces are zero (no wind, no temperature difference).

The calculated driving forces acting on the system are:
Temperature driving pressure = $0{,}045 \cdot H \cdot dt = 0{,}72 \cdot dt$ (Pa). (H=16m).
Wind driving pressure, exhaust = $c \cdot p_{dyn} = 0{,}6 \cdot 0{,}6 \cdot v^2 = 0{,}36 \cdot v^2$ (Pa). (v= wind speed, m/s)
Wind driving pressure, supply = $c \cdot p_{dyn} = 0{,}6 \cdot 0{,}6 \cdot v^2 = 0{,}36 \cdot v^2$ (Pa). ($c_{suppl} \approx c_{exh}$)
In Oslo the wind speed seldom exceeds 5 m/s, therefore the temperature force is normally more important in the heating season.

Another important reason to have wind boosted intake and exhaust devices is to ensure that the flows goes in the correct direction through the system. Never the less there was one day in which there was reverse flow in the supply duct, due to an open door, and the reverse flow continued many hours after the door had been closed. The wind speed was however less than 2,5 m/s in this period.

In climate zones like in Oslo, with low wind speed, the benefit of the wind boosted intake and exhaust openings can be questioned. The cost for these devices can be higher than the energy they save during expected life time. Without these devices the system in principle is more like a traditional balanced system, but designed for very low pressure drop and arranged to use the natural temperature driving force.

It was anticipated that the assisting fans could be used as flow controllers. The tests confirmed that this function was very efficient.

There is no audible noise from the system. Because of the low pressure drop and the fans running on less than 50% of maximum speed, the fan noise is very low.

The installation cost for a future system similar to the tested pilot system has been calculated, with the assumption that the future system can be produced in series and installed in many wings of a new building. But even then the new system costs more than a traditional, balanced system, because of bigger duct- and component- dimensions per m^3/s air flow, and more lost floor area. This will also depend on how well the system is integrated in the building construction, i.e. the co-operation between the architect and the rest of the design team is important.

But the running cost is lower for the new system, because of low electric energy consumption by the fan. At the moment the lower running cost can not compensate for the higher installation cost, calculated with actual Norwegian energy prices. The Norwegian price of electricity is however anticipated to increase more than other prices, and therefore the total cost of the new system may be the lowest in the future.

5. CONCLUSIONS

Practical concepts for natural ventilation with heat recovery have been developed. A test system designed for an air flow rate of 400 l/s was evaluated in the laboratory at NBI. The system consists of:

- Roof unit with heat exchanger, coarse filter, assisting fan, a wind-boosted air exhaust unit and a wind-boosted air intake unit.
- Floor unit with electrostatic filter, heat exchanger, and an assisting fan
- Duct system. Exhaust air ducting with exhaust terminals in three levels. Supply air ducting in three levels with supply terminals (simulated with dampers). As a part of the supply system an insulated vertical main duct connects the air intake of the roof unit with the floor unit. The supply air may also be taken from ground level outside the building through an underground culvert (channel).
- A flow controller controlling the speed of the fans.

The total pressure drop for the system (supply and exhaust) is approx. 50 Pa. There is room for a reduction in pressure drop between 10 and 20 Pa by optimisation of the duct design.

Maximum power consumption for the extract fan is about 37 W, but has been measured as low as 18 W, when the natural driving force is higher.

The tests show that the laboratory building is quite leaky and that reverse flow can occur when the door is left open for a while. Therefore it was decided to install an assisting fan also in the supply system. In a very tight building it should be sufficient with one fan (in the exhaust).

Average power requirements for two assisting fans is about 2 x 28 W = 56 W. This gives a Specific Fan Power (SFP) = 0,14 kW/m^3/s, which is about 5% of a typical system today.

The system may run with the assisting fans turned off, but generally with reduced air flow rate. The fan speed control system controls the air flow rate very efficiently.

The temperature efficiency for the heat recovery is measured to 0,50. With an optimal design of the installation the efficiency will increase to about 0,60.

The installation cost of the system is higher and the running cost is lower than for a traditional balanced ventilation system. The total cost over the lifetime depends on the future price of electricity and how well the system is integrated into the building.

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7. CONTRIBUTORS

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Ziehl-ebm as, P.O.Box 173 Holmlia,N-1203 Oslo: Assisting fans with regulators, intake louvers.
Oddbjørn Hagen AS, P.O.Box 263 N-2381 Brumunddal,: Mounting of ducts, equipm., insulation.
Nor-Vent AS, Prof. Birkelandsv. 35, N-1081 Oslo: Discount on ducts and fittings.
NVP, Industriveien, 2072 Dal: Discount on chambers.
ABB Klimaprodukter, Odalsg. 25, N-2001 Lillestrøm: Discount on heat exchanger coils.
MiljøSystemer A/S, P.O.Box 341 Økern, N-0513 Oslo: Discount on electrostatic filters.
Proff Utleie A/S, Strømsv.226, N-0668 Oslo: Discount on scaffolding.
Ing. Wilberg a.s, P.O.Box 6424 Etterstad, N-0605 Oslo: Discount on velocity sensors.
Norsk Ventilasjon og Energiteknisk Forening, P.O.Box 7174 Majorstua, N-0307 Oslo:
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VENTILATION TECHNOLOGIES IN URBAN AREAS

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OSLO, NORWAY, 28-30 SEPTEMBER 1998

CONTROL OF NIGHT COOLING WITH NATURAL VENTILATION

Sensitivity Analysis of Control Strategies and Vent Openings

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The inverse of C, 1/C, denotes the resistance as shown in figure 3. The driving pressure sources 1 to 9 are calculated with a Cp-generator [3] according $P_i = \frac{C_p \cdot \rho \cdot v^2}{2}$.

The thermal model calculates the air and mean wall temperatures in the rooms and the air temperature in the corridor or hall, see figure 4. The inputs are the outdoor weather data, the ventilation flows from the ventilation model and the window and louvre angles from the controller. During office hours there is an internal heat source in the rooms but not in the corridor. Moreover there are the ventilation heat flows caused by the airflow's calculated by the ventilation model in all spaces. The window and sun-shading device have no heat storage. The solar gain that enters the room is splitted into a convective and radiant part and is determined by respectively:

$$Q_{s\_conv} = Q_s \cdot ZTA \cdot CF \quad \text{and} \quad Q_{s\_rad} = Q_s \cdot ZTA \cdot (1 - CF)$$

With:

CF = convection factor
Q_s = solar radiation on the façade [W/m^2]
Q_{s_conv} = convective heat flow from the window to the room air [W/m^2]
Q_{s_rad} = radiant heat flow falling through the window [W/m^2]
ZTA: = solar atmittance factor

The radiant part is assumed to be equally divided across the entire inner area of the room. In figure 4 the broken arrows represent these solar gains. For a double pane window with the outside sun shades down, ZTA=0.13 and CF=0.07; with the sunshades up, ZTA=0.7 and CF=0.03. The sun shades goes down when $Q_s > 250$ W/m^2. In [1] the ventilation and thermal models are described in more detail.

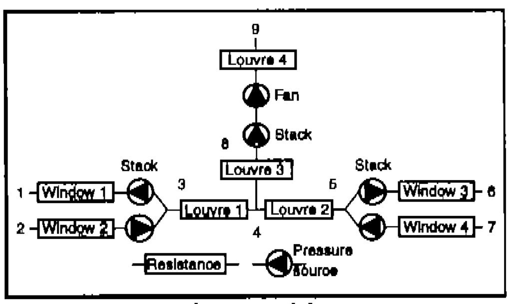


Figure 3. The ventilation model

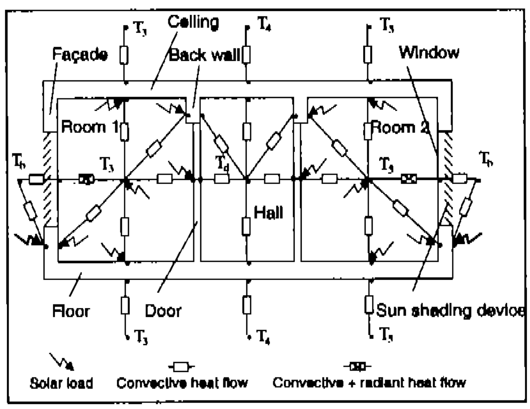


Figure 4. The thermal model

3. THE CONTROL STRATEGIES

In [1,2,4,5,6] various predictive control strategies for cooling are described. Five are compared:

- 'Cooling day' control strategy [1,2]
- 'Degree hour' control strategy [4]
- 'Setpoint' control strategy [4]
- 'Slab temp' control strategy [4]
- 'Manual control'

All control systems use the same control strategy during office hours (08:00-18:00) (PI-feedback algorithm that tries to keep the resultant temperature at the cooling setpoint (22 ^{0}C);

when $T_0>T_{in}$, or when $T_0<12\ ^0C$, the vent openings and louvres are closed), but in off hours (18:00-08:00 and weekends), each system uses its own control strategy to precool the room.

The 'Cooling Day' control strategy uses the same PI-algorithm as during office time, only the precooling setpoint is variable. It is set according a rule based prediction between 18-22 ^0C: after a day with demand for cooling, the setpoint is decreased with 2K until 18 ^0C. No cooling day set the opposite in operation. So it anticipates on a coming warm period [2].
Precooling mode with 'Degree Hour' control strategy. At the end of the office time the degree hours are determined (number of hours $T_{in}>21\ ^0C$). A positive number means heat has been absorbed in the slab. Night cooling is only permitted, when the air temperature during that day exceeds 24 °C for more than 1 hour.
Precooling mode with 'Set Point' control strategy. Night cooling is enabled if the average outside air temperature between 12:00 and 17:00 hours exceeds 18 °C.
Precooling mode with 'Slab Temperature' control strategy. The 'Slab Temperature' control strategy is the same as the 'Cooling Day' control strategy, only now the controlled variable is the floor slab temperature in the room at a depth of about 4 [cm].
Precooling mode with Manual Control. The vent openings or louvres are set manually based on the indoor temperature at the end of the office time and the maximum outside temperature as forecast by the weather station for the next day. For the weekend this means that no corrections will be made at changing weather.

4. THE INPUT VARIABLES

Weather data: The Bilt, May, 1ste - September, 30th, 1964:

- Outdoor temperature [°C]
- Wind velocity [m/s]
- Wind direction [degrees]
- Global radiation [W/m^2]

  The total solar radiation on the 4 vertical orientations (North, South, West and East) are calculated values based on the splitting procedure of Orgill and Hollands [7]. Linear interpolation is used to determine the values per time step.

Building orientation

- NS-orientation (room 1 north, room 2 south)
- WE-orientation (room 1 west, room 2 east).

Building inertia

Simulations were made with 3 different building (room) inertia, M:= 55 (low inertia), 75 (medium inertia) and 100 (high inertia) [kg/m^2]. The inertia, M [kg/m^2] of a room is defined as half of the mass of the four walls, the ceiling and the floor divided by the enclosing area.

Internal heat gain

The internal heat gain represents the body heat of people, the heat from printers, computers, lights and other electrical devices in the room during office hours. The unit is Watts per square meter of net floor area. It is varied from 20 to 40, with steps of 5 [W/m^2].

Comfort criterion

The resultant temperature is allowed to exceed 25.5 ^0C for a maximum of 100 hours; of this only 15 hours may exceed 28 ^0C in office hours in one year.

Windows and effective vent openings

The total transparent window area is 40 [%] of the façade area. Four types of vent openings or louvres are used. The specifications are given in table 1.

If vertical or fan assisted ventilation is used, the fan is switched on if one of the windows are full open, the wind velocity is less than 2 m/s and T_0>20 °C. The fan is dimensioned for 4ach.

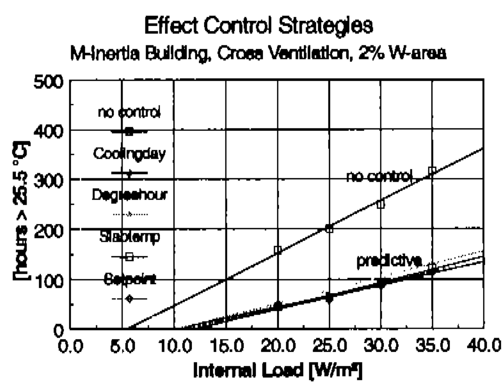
Table 1 Specifications of $A_{eff.max}$

Type	Dimension	Maximum effective vent opening area $A_{eff,max}$	
	w*h [m]	[m^2]	% of net floor area
1	3.2*0.091	0.18	1
2	3.2*0.20	0.41	2
3	3.2*0.31	0.64	3
3/2	2*3.2*0.155	0.64	3

The dimensions of type 1 [6,8] and 3 [7] are chosen based on directives found in literature. Type 3/2 is a split window, one half above and one half below the fixed transparent window (more stack). The type 2 window is a compromise between type 1 and 3. The louvres above the doors, in the back walls of both rooms, have the same dimensions as the windows. The louvres in the air duct have twice the dimensions of the louvres above the doors.

5. SIMULATION RESULTS AND DISCUSSIONS

Here only the results of the control strategies are discussed. The simulation results of a South oriented, cross ventilated, medium inertia office building room, with effective vent opening type 2, using the different control strategies mentioned above, is shown in figure 5.



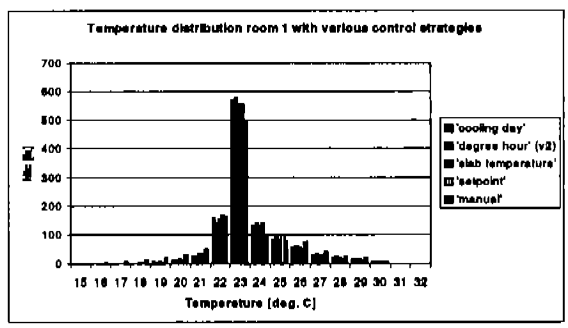


Figure 6. Temperature distribution in a South room with cross ventilation with various control strategies.
← Figure 5. Effect of the control strategies

As reference situation is taken a natural ventilated building with the windows closed after office hours and in the weekend. This is shown in the upper curve (no control) in the plot in figure 5. It clearly demonstrates the merits of night cooling.

Surprisingly, only by using night cooling, whether it be manually (not shown in the graph), or automatically controlled, the overheating hours drop dramatically. Through night cooling with manual control there are hours with too low as well as too high temperatures, see figure 6. Moreover, the overheating hours (Ti > 25 ^{0}C) are also higher than with the other control strategies. It seems, that the differences between the considered control strategies are very small, what really matters, is to cool the building at night and in particular in the weekends. Also the setting of the control parameters shows to be more important than the strategy itself.

Allowable internal heat gain with the three ventilation types

As has been shown in the previous paragraphs, the various control strategies produce nearly the same results (see figures 5 and 6). So only one control strategy, the 'Cooling Day' control strategy, will be used to investigate the influence of the three ventilation types, single sided, cross and fan assisted ventilation (designed for 4 ach) in more detail. The following parameters has been used:

Building inertia: low (55), medium (75) and high inertia (100 kg/m^2)

Building orientation: North-South (NS) and West-East (WE)

Window type: Type 1, 2, 3 and 3/2 (Table 2)

The results are shown in Table 2.

Single sided ventilation depends very much on the wind direction. The north and west rooms (first values) allows higher internal loads than the south and east rooms. A split window with the same effective vent opening gives better results than an undivided window. This is shown with the results of window types 3 and 3/2. This is caused by the extra stack effect that occurs by two vent openings. Light buildings are unsuitable. Medium inertia building can be used, but only with 3% vent openings. High inertia buildings perform best with window type 2 and up. Window type 1 is too small.
With cross ventilation light buildings can only be used with 3% windows; window type 2 is sufficient for medium inertia buildings. Window type 1 can only be used with high inertia buildings.

Table 2 Simulation results

Building inertia [kg/m^2]	Orien tation	Window type	Allowable internal heat gain [W/m^2] for		
			Single sided vent.	Cross vent.	Fan assisted vent. (2 ach)
Low [55]	NS	1	<20/<20	<20	<20
Low [55]	NS	2	<20/<20	<20	20
Low [55]	NS	3	<20/<20	23	22
Low [55]	NS	3/2	21/<20	26	23
Low [55]	WE	1	<20/<20	<20	<20
Low [55]	WE	2	20/<20	<20	<20
Low [55]	WE	3	<20/<20	<20	21
Low [55]	WE	3/2	<20/<20	23	22
Medium [75]	NS	1	<20/<20	<20	24
Medium [75]	NS	2	<20/<20	26	28
Medium [75]	NS	3	23/22	32	32
Medium [75]	NS	3/2	30/27	33	33
Medium [75]	WE	1	<20/<20	<20	22
Medium [75]	WE	2	<20/<20	22	28
Medium [75]	WE	3	22/<20	26	31
Medium [75]	WE	3/2	29/26	28	32
High [100]	NS	1	<20/<20	24	27
High [100]	NS	2	26/22	33	36
High [100]	NS	3	29/27	42	47
High [100]	NS	3/2	38/36	43	40
High [100]	WE	1	<20/<20	21	28
High [100]	WE	2	24/20	27	34
High [100]	WE	3	29/24	32	44
High [100]	WE	3/2	37/32	36	40

Vertical fan assisted ventilation is only slightly better than cross ventilation.
In practice requirements about flexibility do not allow interior sidewalls to be build of stone. Therefore medium inertia buildings are more often built then high inertia buildings.
Although vent windows type 3 is better than type 2, preference should be given to type 2, for safety reasons (burglary). All this taken into consideration leads to the following conclusions:

Cross ventilation with 2% effective vent opening can be applied in medium and high inertia buildings. Smaller effective vent openings should not be applied. Light buildings are not suitable for natural ventilation. The allowable internal heat gain for a medium inertia building with orientation north-south is 26 [W/m^2] and east-west 22 [W/m^2]. For a heavy inertia building these are respectively 33 [W/m^2] and 27 [W/m^2].

6. CALCULATION EFFECTIVE VENT OPENING AREA A_{eff}

A great number of simulation runs have been performed. For each simulation run, only one parameter is varied. By analyzing the outputs of all these simulation runs, it is possible to identify two design methods to determine the effective vent opening area:

a. Method based on a selection chart
b. Method based on simplified equations

Method based on a selection chart

Based on the same simulation outputs a graph is designed in which the effective ventilation opening can be found without any knowledge about building physics. First the internal heat gain should be fixed. Then the type of window with its solar shading system, the accumulation factor M and the control strategy lead to the ventilation systems and effective openings that can be applied. The procedure is shown in figure 7

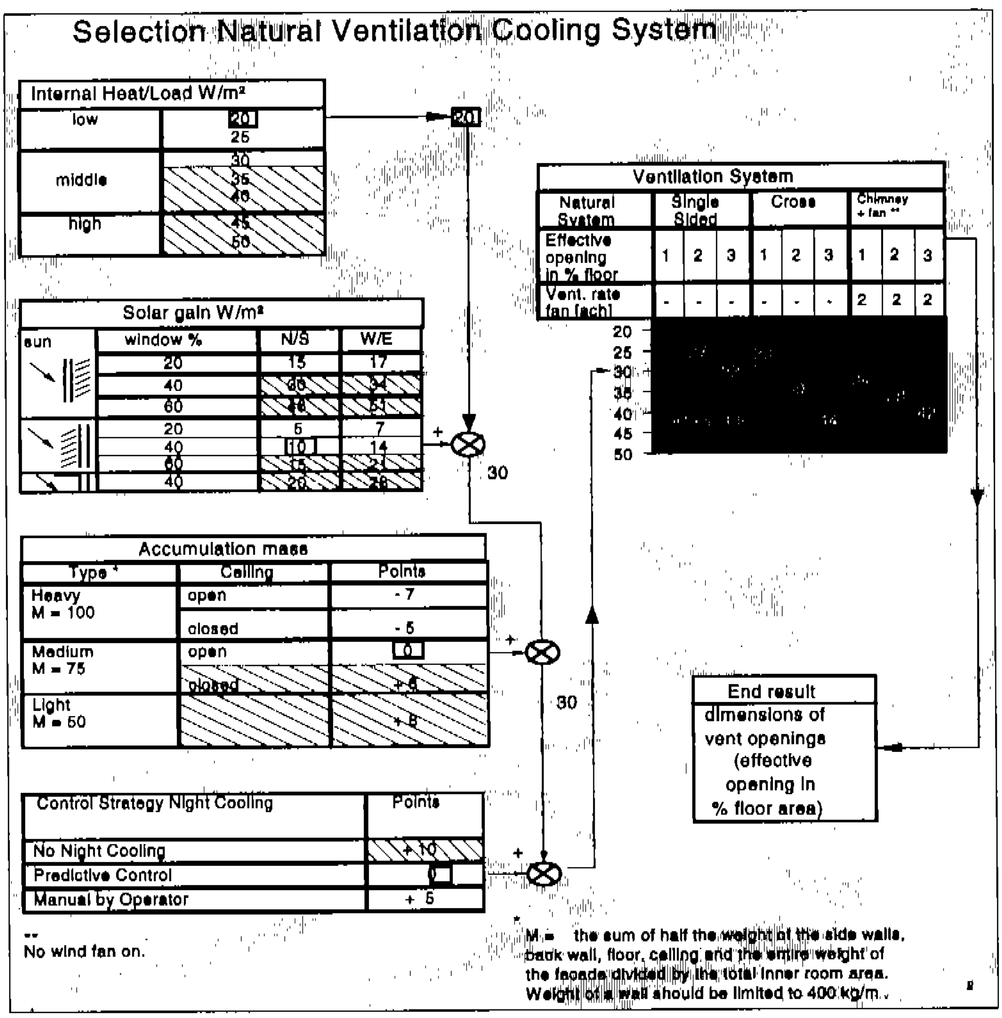


Figure 7. Graphical design tool for night cooling

Method based on simplified equations

The dimensions of the effective opening can be expressed as function of internal heat gain, solar protection system, mass of the building, control strategy and type of ventilation (single sided, cross, and fan assisted systems). The reference situation chosen is: cross ventilation, 40% glass area, outside shading, medium inertia building (M=75 kg/m^2).

Based on the simulation results for this particular situation, the required effective opening area, $A_{eff,ref}$, is determined as function of the heat gain (internal and solar), the comfort criterion, and the building inertia. These parameter values are the central point for further calculations. The coefficients are found by fitting the equations on the simulation output data. The total heat gain, Q_{im}, consists of two heat flows: the internal heat flow, Q_i, that consists of heat of people and electrical appliances and lights, and the transmitted solar heat that depends on the glazing type internal and external sun shades, denoted by the solar atmittance factor ZTA, and the window area, A_{window}. Q_{im} [W/m^2] is denoted by:

$$Q_{im} = Q_i + 10 \cdot (\frac{ZTA \cdot A_{window}}{0.585} - 1) \quad \text{with} \quad ZTA = \frac{\textit{nett solar heat entering the room}}{\textit{solar radiation falling on the window}}$$

The effective vent opening for the reference situation, $A_{eff,ref}$ (cross) [%], can be denoted by:

$$A_{eff,ref}(cross) = -0.172 + 0.124 \cdot Q_{im}$$

When the building inertia is different from: 75 [kg/m^3], this becomes

$$A_{eff}(cross) = \frac{75 - 0.41 \cdot (M - 75)}{M} \cdot A_{eff,ref}(cross)$$ with M [kg/m^2] is the building inertia.

The effective vent opening for single sided ventilation can be derived from:

$$A_{eff}(single\ sided) = 2 \cdot A_{eff}(cross)$$

Cross ventilation can be improved by using a duct system with exhaust fan. The power of the fan is designed to realise a ventilation rate of n ach (air changes per hour). It is only switched on when the wind speed is less than 2 m/s. The effective vent opening becomes:

$$A_{eff}(fan\ assisted, ventilation\ rate\ n) = (-0.1875 \cdot n + 1.375) \cdot A_{eff}(cross)$$

The equations derived above are found for a comfort criterion $N_{hour} = (N_{>25.5} \leq 100)$ (maximum 100 overheating hours above 25.5 [^{0}C] are allowed) with an internal heat gain Q_{im} of 20 [W/m^2]. In case more than 100 overheating hours are allowed $N_{hour} = (N_{>25.5} > 100)$, the reference effective vent opening, $A_{eff,ref}$, is given by

$$A_{eff,ref}(cross) = -0.172 + [0.124 + a_2 \cdot (100 - N_{hour})] \cdot Q_{im}$$

Where

$$a_2 = [3.8 + 0.09 \cdot (Q_{im} - 20)] \cdot \frac{N_{hour} - 50}{100} \cdot 10^{-4} + [1.24 + 0.009 \cdot (Q_{im} - 20)] \cdot (1 - \frac{N_{hour} - 50}{100}) \cdot 10^{-3}$$

Warning: Accuracy of the equations can be quaranteed for: $20 < Q_i < 30$; $50 < M < 100$; $0 < ZTA < 0.3$ and $50 < N_{hour} < 150$.

7. CONCLUSIONS

The optimal solution for a natural ventilated office building with night cooling would be a system with cross ventilation and an effective vent opening area of 2% net floor area. The allowable internal heat gain for a medium inertia building is 22-26 [W/m^2] and for a high inertia building 27-32 [W/m^2]. It performs better than single sided ventilation that works only in a high inertia building and is a lot cheaper than vertical ventilation (air duct). Windows with 1% vent opening are too small and with 3% effective opening too large (burglars). Although the split window of 3% is burglar free, its better performance does not match the higher costs.

All predictive control strategies mentioned in this paper show more or less the same performance. What really matters is the tuning of the night cooling strategy and the application of night cooling.

Vertical fan assisted ventilation with 2% effective vent openings improves the performance of cross ventilation provided a ventilation rate of at least 4 air changes per hour is applied

Based on the outputs of a large series of simulation runs, two user friendly design tools are developed. The first design tool is a graphical tool in the form of a chart. The second design tool consist of simplified equations and can be used in a spreadsheet. With these design tools it is easy to determine the required ventilation opening and control strategy at an early stage of the building design.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER 1998

PRACTICAL GUIDELINES FOR INTEGRATED NATURAL VENTILATION DESIGN

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ABSTRACT

Practical Guidelines for Integrated Natural Ventilation Design

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In many countries there is a turn towards natural ventilation as an alternative to energy and cost demanding mechanical ventilation systems. The objective is to save money and energy while maintaining an acceptable indoor air quality and thermal climate, or even to improve the indoor environment by reducing noise levels, giving the user more control over the indoor climate etc. The aim of the EC-JOULE project *NatVent*^{TM} is to investigate, develop and integrate "smart" components to provide good natural ventilation for office-type buildings. Hitherto, simple design tools and guidelines for integrated natural ventilation design have not been available for practitioners. Therefore, as part of the *NatVent*^{TM} project, a robust and easy-to-use computer simulation program has been developed, coupling an airflow calculation model with a thermal model. One of the most important objectives while developing the program has been to create a robust underlying theoretical model and an easy-to-use interface. The aim for the user interface is to facilitate the use of the program by any building designer, architect or engineer at an early design stage. Therefore the interface uses input that are simple to quantify and so the simulation tool can be used at any early design stage giving an indication of the suitability of natural ventilation in a specific building.
In addition to the development of the simulation tool, a large number of test-runs have been performed in order to identify the most significant parameters that influence the indoor air temperature and the outdoor air flow rates.
The test-runs have been statistically processed and thus a detailed picture of which parameters have the largest influence on the indoor environment can be presented. These are consequently generalised as guidelines for integrated natural ventilation design. The proposed paper will focus on these guidelines.

VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
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MODERN PASSIVE STACK VENTILATED SCHOOLS – EVALUATION OF VENTILATION AND MOISTURE CONTENT

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Synopsis

The aim has been to determine ventilation rates and risk of moisture damage in three modern schools with passive stack ventilation. The users are supposed to control the ventilation by using the lantern windows and the outdoor air is assumed to enter through an underground duct. The paper presents results, analysis and conclusions from the performed measurements and calculations.

The ventilation rates are sometimes low and vary with the use of the windows in the facade and the lantern. It is, however, always possible to arrive at a sufficient ventilation rate. The supply air flow through the underground duct can, without a supply fan, be low and even go backwards during warm weather. To obtain desired ventilation rates and energy conservation the building must have a good airtightness.

High relative humidities and even periods with condensation occur in the underground supply duct during spring and summer. Microbial growth has been found in two of the schools. Two important factors are choice of material and cleaning, where the knowledge is insufficient today. Moisture and microbial growth have been found in the roofs. The leakage paths, supply of moisture indoors and an interior pressurization have contributed. In order to reduce the risks the building must have a good level of airtightness.

1. INTRODUCTION

Natural ventilation (passive stack ventilation) is today a permitted ventilation system in new buildings according to the building codes in the nordic countries. The interest for passive stack ventilation is substantial in the nordic countries today. Apparently there is a resistance to mechanical ventilation systems among many architects and users. The arguments for and against are many. An important problem with passive stack ventilation is that its function is very dependent upon the current weather, which means that the user must agree to the fact that the indoor air quality and indoor comfort will very much depend upon the user's use i. e. opening and closing windows, controlling air inlets etc..

Passive stack ventilation can, but does not have to, mean risks. At low ventilation rates there is always a risk for insufficient indoor air quality and high moisture levels. If the upper part of a building is pressurized moist air can flow to the outside through air leakage paths and cause condensation and damage. This can also occur in buildings with mechanical ventilation.

During the last couple of years many passive stack ventilated schools have been built in Sweden. Most of them have a supply of outdoor air through underground ducts. In spite of the fact that this technique has been introduced in a number of projects the theoretical background is insufficient. The practical application of the technique does not seem to be founded on any extensive technical research and development work.

If the outdoor air enters the building through an underground concrete duct, the indoor air quality is at risk. Summer conditions mean that warm moist air can be cooled and condense in the duct. This condensation can cause microbial growth, which will impair the air quality.

The aim of this investigation has therefore been to determine ventilation, pressure conditions, CO_2-levels and risk of moisture damage (Blomsterberg 1997) for passive stack ventilated schools with underground supply ducts.

2. THE SCHOOLS TESTED

The investigation was carried out in three schools with similar design principles of building and ventilation system (see table 1). The differences are different designs of the underground supply duct, with or without supply fan, with or without vapour barrier of plastic foil.

Table 1 Description of investigated schools.

School	Year of construction	Location	Ventilation	Remarks
Å	1995/96	Western Sweden	Passive stack with lanterns, supply air through concrete ducts (crawl space) under the entire building	4 classrooms, no plastic vapour barrier
Y	1993/94	Southern Sweden	Passive stack with lanterns, supply air through concrete ducts (crawl space) under the building, supply fan for cooling, mechanical exhaust from WC	4 classrooms, no plastic vapour barrier
Z	1994/95	Western Sweden	Passive stack with lanterns, supply air through concrete ducts (with room height) under the building, supply fan for cooling, mechanical exhaust from kitchen and dining hall	8 classrooms

The outdoor air to the Z-school enters the building at the roof ridge through a vertical shaft. In the other schools the outdoor air enters through an underground concrete pipe, which ends up in the concrete duct under the building. The purpose of the underground duct is to be able to cool the supply air during summer and preheat during winter i. e. most of the air should go this way (Andersson 1995, Andersson 1996).

The ventilation system relies on that the users ventilate according to the actual need. The idea is that the user shall control the ventilation using the lantern windows. In some classrooms the control is automatic.

The room height is higher than in most modern schools i. e. the room volume per pupil is bigger. This larger volume acts as a ventilation buffer. The total air flow which has to be supplied is the same independent of room volume, the ventilation can however be postponed in time. This is true if the ventilation efficiency is not influenced by the room volume.

3. METHODS

The project was started with an ocular inspection of the building and the building services installations. Then measurements were carried out during a summer and a winter period:

- CO_2-concentration and air temperature in classrooms with lesson going on
- Airtightness combined with determination of the location of the leakage paths. This measurement was carried out at one occasion to document the quality of the airtightness.
- Continuous recording of the relative humidity in the underground duct, in the roof construction, in a classroom and outside
- Control of microbial growth in the underground ducts and pipes, and inside the roof
- Passive tracer gas for the determination of average ventilation rates for a couple of weeks

In order to generalize the measuring results numerical simulations of the ventilation rates were carried out, using a multi-zone air flow model, COMIS (Feustel 1990, 1995). A parametric study was made, where the following parameters were varied: temperature, wind, use of windows (in the facade and the lantern), and airtightness.

4. RESULTS AND DISCUSSION

4.1 Ventilation

The Å-school and the Z-school have a very varying ventilation rate depending upon the use of the windows in the facade and the lantern. Sometimes the ventilation is low (less than 3 l/(s and person). In the Swedish building code 7 l/(s and person) is recommended and 0.35 l/(s and m^2) is required. The requirement is usually fulfilled in the Z-school (see table 2), but not always in the Å-school (see table 3).

Table 2. Measured average ventilation rates in the Z-school, 21 May - 5 June, 1997. The average outdoor temperature was + 13 °C, with a minimum of 0 °C and a maximum of + 31 °C. The average indoor temperature was + 20 °C , with a minimum of + 18 °C and a maximum of + 22 °C. The wind speed (10 m above ground) varied between 0 m/s and 7.9 m/s, with an average value of 1.7 m/s. The classrooms have an floor area of 58 m^2.

Room	Day				Night + weekend	
	Air changes/h	l/(s m^2)	l/(s person)	l/s	Air changes/h	l/s
Cloak-room, A124						
Group-room, A129						
Classroom, A130	1.07	1.17	7	68	0.15	10
Meeting room, A131	1.29		7	75	0.26	15
Workshop, A132						
Meeting room, A133						
Cloak-room, A134						
Classroom, A135	0.43	0.47	3	27	0.24	15
Group-room, A136						
Total			171			40

Table 3. Measured average ventilation rates in the Å-school, 22 May - 6 June, 1997. The average outdoor temperature was + 14 °C, with a minimum of 4 °C and a maximum of + 29 °C. The average indoor temperature was + 21 °C , with a minimum of + 18 °C and a maximum of + 23 °C. The classrooms have an floor area of 76 m².

Rum	Day				Night + weekend	
	Air changes/h	l/(s m²)	l/(s person)	l/s	Air changes/h	l/s
Classroom, 4	0.18	0.24	2	18	-	-
Classroom, 12	0.34	0.46	4	35	0.22	23
Classroom, 17	0.36	0.49	4	37	0.07	7
Classroom, 23	0.17		2	17		
Summa exkl. 4				89		30

In the Z-school most of the time most of the outdoor air enters through the underground duct. At mild/warm weather the outdoor air enters through the underground duct thanks to the supply fan, the air flow depends upon the capacity of the fan. Without a fan air can even leave the building through the underground duct. In the Å-school half of the outdoor air enters through the underground duct (see table 4). This school has no supply fan. Besides at times a fairly big air flow leaves the building through the underground duct as well.

Table 4. Measured average outdoor air flow rates in the Å-school, 22 May - 6 June, 1997. The average outdoor temperature was + 14 °C, with a minimum of + 4 °C and a maximum of + 29 °C. The average indoor temperature was + 21 °C, with a minimum of + 18 °C and a maximum of + 23 °C.

	Diurnal	Day	Night + weekend
Air flow underground duct, l/s	89		
Total air flow, l/s	152	122	164

With this type of ventilation system the air flow can be too low if the windows in the lantern and the facade are closed. This is true above all when warm weather is prevailing. By opening windows in the lantern and the facade in a correct way there is always the possibility to obtain a sufficient ventilation rate. The user is to a great extent responsible for the ventilation, if there is no automatic control. The alternative to control the ventilation automatically often means a complicated control system, as there are many different modes of operation.

The ventilation is to a great extent influenced by wind when the weather is mild/warm. Above all the lantern must be designed and used in a correct way, so that wind and temperature driving forces do not counteract each other.

The air flow in the underground duct can be very low and even go in the wrong direction above all for warm weather. In order to guarantee that the air flows in the right direction a supply fan is needed for this kind of weather. If the supply fan is used during winter this means big risks for moisture damage in the roof construction.

If the windows in the facade and in the lantern are open, then there is a risk that the air flow through the underground duct will be very low (see table 5). With closed windows in the facade the air flow through the underground duct increases with the leakiness of the building envelope. The lowest adjustable ventilating air flow increases with increasing leakiness and can become too big during cold weather and with no one in the building i. e. nights and weekends. To obtain the correct air change rate and a low use of energy the building has to have a good level of airtightness.

Table 5. Calculated air flow rates with fan off, classroom A130, for an outdoor air temperature of + 20 °C and with a measured airtightness of 13.5 $m^3/(m^2h)$ @ 50 Pa. P-duct = percentage of outdoor air through the underground duct. The flows have been calculated for different wind speeds and directions.

	Lantern and window open			Lantern open and window closed			Lantern and window closed		
	Total, l/s	Duct, l/s	P-duct, %	Total, l/s, (l/sm^2)	Duct, l/s	P-duct, %	Totalt, l/s, (l/sm^2)	Duct, l/s	P-duct, %
NE 5 m/s	347	81	23	124	95	76	32 (0.55)	31	96
E 5 m/s	328	60	18	164	73	44	18 (0.31)	18	98
SE 5 m/s	371	66	18	296	43	14	28 (0.48)	24	85
S 5 m/s	454	47	10	339	0	0	22 (0.38)	13	59
SW 5 m/s	132	0	0	47 (0.81)	0	-1	12 (0.21)	-1	-7
No wind	153	0	0	35 (0.60)	18	51	0	0	0

The CO_2-level, which mainly is an odour indicator, can at times exceed the recommended level of 1000 ppm in the examined classrooms. In most cases this can be avoided by using the windows, manually or automatically.

An important condition for passive stack ventilated schools is that the building and its fittings and furniture does not emit any irritating substances, as there is a risk that the ventilation can be low. The main purpose of the ventilation is to ventilate for the users i. e. to remove pollutants caused by the users and of course provide the users with a sufficient amount of fresh air.

An indoor environment questionnaire in seven schools with similar passive stack ventilation systems (Hult 1997) shows that most of the seven schools 'have a relatively well perceived indoor environment and health, in spite of the fact that the air changes are lower and the CO_2-levels are higher than in schools with balanced mechanical ventilation'.

4.2 Moisture

The roofs of the three schools have in many measuring points been supplied with moisture, probably from the inside due to moisture diffusion combined with moisture convection. Air leakage of the upper parts of the building combined with a supply of moisture and an interior pressurization during certain periods will create risks for moisture damage, which also the measurements show. Damage as microbial growth have been discovered in some measuring points in the three schools. For the Z-school this probably occurred during the construction.

As an interior pressurization can occur during certain periods, the airtightness of the building envelope is crucial for avoiding damage caused by moisture convection. This kind of damage can occur also in buildings with other types or ventilation system. Buildings with mechanical supply and exhaust ventilation can experience interior pressurization due to partly clogged supply ducts. Buildings with only mechanical exhaust are best in this respect.

The values of relative humidity in the underground ducts during late spring and early summer as measured in this project show that 80 % is at times exceeded. Even periods with condensation have been measured in one of the schools. The risk of microbial growth on materials sensitive to moisture obviously exist. Damage as microbial growth has been discovered in two of the schools. The seriousness of this has to followed up during a longer period of time, a couple of years. Two important factors in this context is choice of material and cleaning. Regular cleaning must be possible and must be carried out. The knowledge of these measures is insufficiently known today.

5. CONCLUSIONS

- The ventilation and thereby the indoor air quality varies with the weather and the users use of the windows in the facade and in the lantern. It is usually possible to arrive at sufficient ventilation rate.
- As interior pressurization can occur during winter there is a risk of moisture damage in the roof construction caused by convection. In order to lower the risks the building has to have a good level of airtightness.
- The supply air flow through the underground duct can, without a supply fan, be low and even go backwards during warm weather.
- There is a risk of microbial growth in the underground supply duct during spring and summer. Two important factors are choice of material and cleaning, of which the knowledge today is however insufficient.
- An important prerequisite for obtaining a desired ventilation and energy conservation is that the building has a good level of airtightness.

During the last couple of years passive stack ventilated schools have been built and additional ones are being planned in Sweden. The technique is argued for by architects, but also engineers. The knowledge about all aspects of passive stack ventilation in modern well insulated school is today not sufficient, which this investigation has shown.

The following R&D-need exists:

- To clarify the function with respect to
  -ventilation efficiency
  -temperature and energy consequences
  -quality of supply air
  -design of outdoor air intake
- To analyze construction and life cycle costs (not commented upon in this investigation)
- To identify critical factors for good function and economy in combination
- To write an easily understandable handbook showing possibilities and risks with passive stack ventilation
- To provide design tools for the designing architect and engineer concerning passive stack ventilation

6. ACKNOWLEDGMENT

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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A WIND TUNNEL STUDY INTO THE LOCATION OF NATURAL VENTILATION AIR INTAKES IN URBAN AREAS

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A wind tunnel study into the location of natural ventilation air intakes in urban areas.

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Synopsis Ventilation of buildings in urban areas may result in high internal concentrations of traffic pollutants if air intakes are positioned where external concentrations are highest. This paper presents the results of a wind tunnel study into different wind-driven natural ventilation strategies for a building situated close to a busy road. Measurements of the concentration of a simulated traffic pollutant inside and around the building illustrate that noticeable reductions in internal concentrations can be achieved if air intakes are placed at roof level or on the leeward face of the building.

1. Introduction

Traffic pollution is of increasing concern in many of the world's cities. Occupants in buildings situated close to busy roads can expect to be exposed to a range of pollutants emitted by the motor vehicle such as oxides of nitrogen (NO_X), volatile organic compounds (VOC) and carbon monoxide (CO). On an urban scale, eighty-nine percent of all the CO in the atmosphere is due to the motor vehicle[1] and the concentration of CO in the atmosphere has been shown to be a good indicator of the concentration of other traffic related contaminants[2]. Although the health effects of these pollutants are not yet fully understood, asthma, other respiratory diseases and some cancers have all been linked to traffic emissions[3, 4].

When considering the ventilation of a building in the urban environment, either by natural or mechanical methods, the position of openings/air inlets is of great importance. If openings are placed where the pollutant concentration is highest then indoor concentrations can reach considerably high levels[5, 6]. The concentrations of traffic pollutants have been observed to decay with height and distance away from the road, which suggests that placing air intakes higher up or on the opposite side of the building to the traffic source should minimise the ingress of contaminated air into the building[7, 8].

It is difficult to assess the benefits of different air intake positions from measurements in the field. There are the obvious problems associated with constructing a full size prototype system or implementing a system into an existing building, and more significantly, the problem of analysing the data collected. When concentration measurements are taken in the field there is no control of parameters such as wind speed and direction, which strongly influence the dispersion of vehicle pollutants around buildings and therefore make the interpretation of data difficult.

Testing ventilation designs at model scale in a boundary layer wind tunnel enables some of the difficulties inherent in full-scale testing to be overcome. If the appropriate scaling conditions are satisfied then measurements made on the scale model will be representative of those that could be expected at full-scale. Several studies have illustrated the similarity between concentrations measured in the field and on scale models in a wind tunnel[9, 10], however at present little attention has been placed on testing ventilation air-intake positions.

This paper presents the results of a wind tunnel study of different natural ventilation air-intake positions for a building on the University of Nottingham campus situated close to a busy road. In a previous study by the authors the dispersion of a simulated traffic pollutant around this building had been investigated using a 1:100 scale model. Significantly lower concentrations of the pollutant were measured at the rooftop and at the non-roadside face and in general showed good agreement with field observations[11]. The present study goes further in that a model is also used to determine the ventilation rate and internal concentration.

2. Wind Tunnel Testing

The wind tunnel used in the tests was a small open-jet wind tunnel capable of delivering a maximum flow rate of 4.5 m/s. The working section has a width of 1 m, height 0.75 m and length 2.25 m[12]. The wind tunnel is relatively simple, and its use may be criticised on the grounds that is does not allow appropriate simulation of the turbulence structure. The mean velocity can be reasonably well simulated, but there is no real control over the generation of turbulence. However, the present concern is with time-averaged concentrations (rather than instantaneous values) and furthermore the pollutant is emitted from a line source (rather than a point source). Both of these factors will tend to reduce the importance of precise modelling of the upstream turbulence. Also the tunnel is an inexpensive facility in terms of capital and running costs. This can be an important factor when dealing with the design of buildings for which the development budgets are often likely to be small.

3. Scale Model Construction

A 1:50 scale model was constructed of the area shown in figure 1. The Institute of Building Technology (IBT) is housed in a naturally ventilated building in close proximity to a busy urban ring road (A52). Traffic flow figures supplied by Nottinghamshire City Council for this section of the A52 show that rush hour flows are typically in the region of 2500 vehicles in either direction and vehicles can be stationary for short periods during this time.

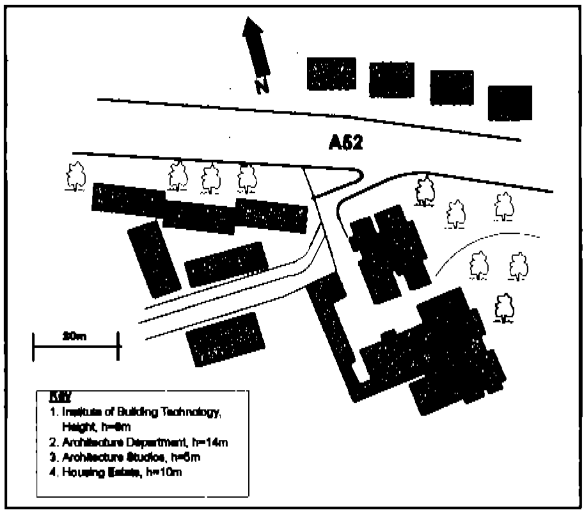

Figure 1 Plan of area modelled in wind tunnel.

A hollow model IBT building was constructed from 6 mm Perspex and the surrounding buildings and features constructed from medium density fibreboard. The detail on the building was kept deliberately simple and no attempt was made to model any surface roughness. On the roadside face of the IBT a circular, sharp-edged opening of 10 mm diameter was added at a height of 85 mm and on the opposite face an opening of the same dimension was placed at a height of 65 mm. On the roof openings of 25 mm diameter were added.

To simulate the exhaust emitted from a queue of stationary traffic 20 x 2 mm diameter holes were drilled in a length of plastic tubing (internal diameter = 18mm). The holes were separated by 40 mm to represent an average spacing of 2 m full-scale between

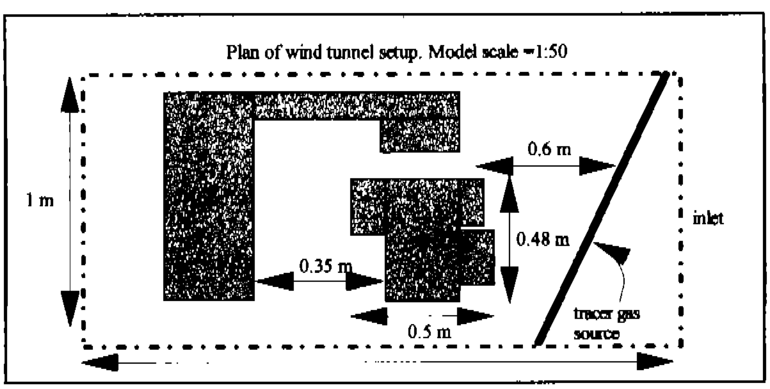

Figure 2 Wind tunnel arrangement of model building and tracer gas source.

successive vehicle exhausts in congested traffic. A tracer gas nitrous oxide (N_2O) was delivered to the tube at a controlled rate of 5 l/min and the tube aligned on the model to represent the actual position of the A52 with respect to the IBT (figure 2). Air was sampled from the interior of the modelled building and at the intake positions and the concentration of tracer analysed using a Binos 1000 analyser.

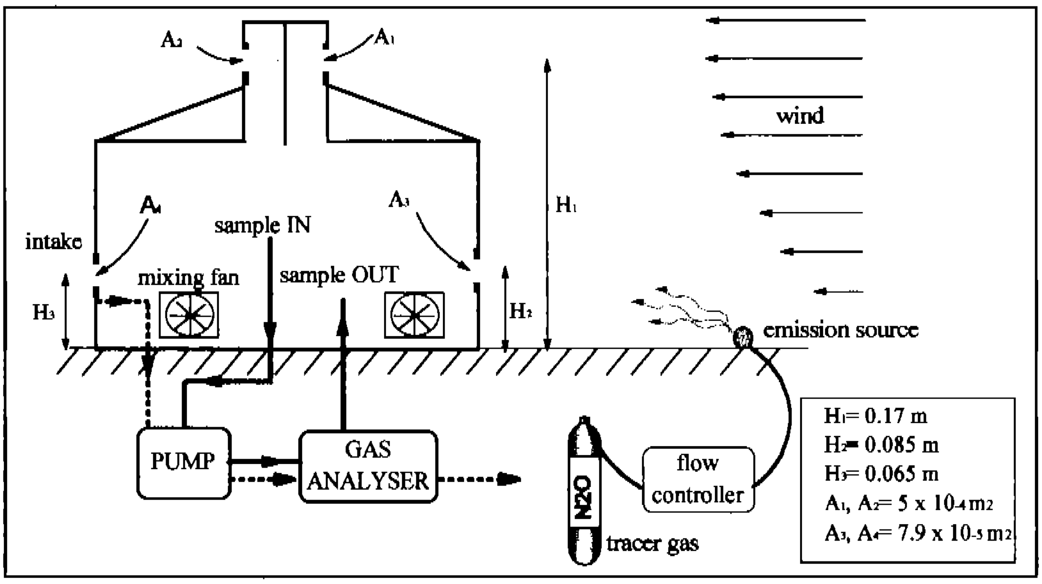

Figure 3 Schematic of model and tracer gas delivery and analysis system.

3. Scaling considerations.

In principle, dynamic similarity of the external flow fields in the model and full-scale requires equality of Reynolds number (*Re*). In practice it is not possible to achieve this equality in a normal wind tunnel using air as the working fluid and therefore at the range of wind speeds available *Re* is much smaller than the full-scale value. For sharp-edged (bluff-body) buildings, the influence of *Re* will be small if *Re* is greater than a critical value (typically quoted as 11,000).

The flow through openings (and hence ventilation rate) however has been shown to be strongly dependent on *Re*[13], in particular the discharge coefficient is a strong function of *Re* for low values of *Re*. For sharp-edged openings such as air vents, the concept of a critical *Re* can be applied, so the problem can be overcome by operating the wind tunnel at a velocity at which the ventilation rate, when expressed as a nondimensional coefficient, becomes independent of *Re*. Figure 4 is a plot of the nondimensional ventilation rate (determined using the tracer decay technique) against wind tunnel speed for each of the models tested (see §4).

From the plots it can be seen that independence of *Re* is obtained for tunnel speeds of 2.5 m/s and greater. A tunnel speed of 2.5 m/s was therefore selected for the experiments, giving $Re = 3.3 \times 10^4$ (for a characteristic model height of 0.16 m).

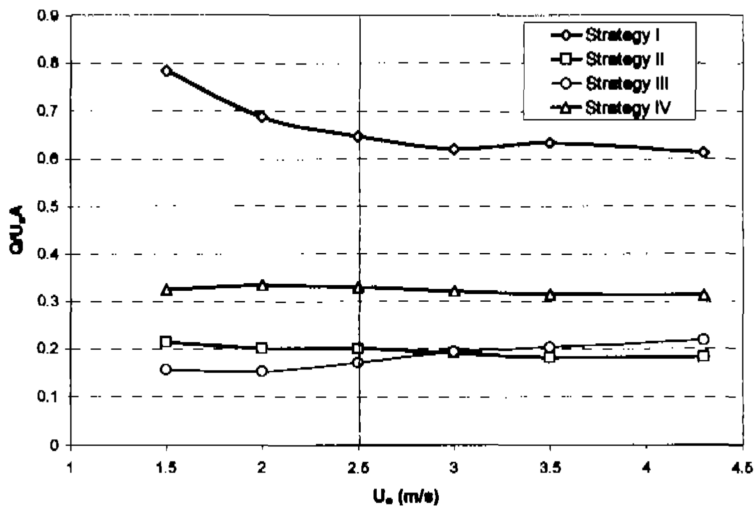

Figure 4. Non-dimensional ventilation rate against wind tunnel speed.

4. Experimental Procedure

Four different ventilation strategies were tested as illustrated in figure 5:

I) Air inlet on the roadside face, air outlet on the leeward face;

II) Air inlet on the leeward face, air outlet on the roof;

III) Air inlet on the roof, air outlet on the leeward face;

IV) Air inlet and outlet on the roof.

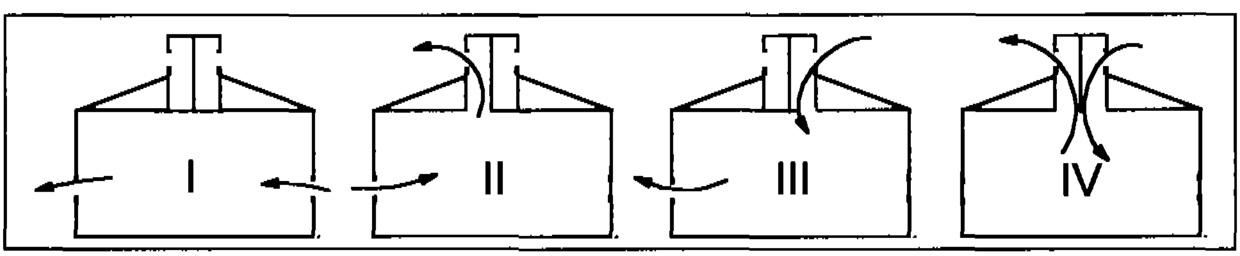

Figure 5 Four ventilation strategies tested in wind tunnel with constant wind speed of 2.5 m/s and constant wind direction (wind direction from right to left).

In each experiment the wind direction was held constant so that the roadside face of the IBT was always perpendicular to the direction of the wind and downwind of the pollutant source. The N_2O tracer gas was released and concentrations were recorded at the openings (inlet and outlet) and inside the model for each ventilation strategy once a steady state condition had occurred. The steady state was judged to have occurred at the time when the internal concentration would theoretically reach ninety-five percent of the concentration at the inlet (T_{95}) and was calculated from the air change rate. Steady state (or equilibrium) concentrations were logged as 2-minute averages on the gas analyser.

5. Results

Table 1 summarises the results obtained for the four strategies. The values of Q/UA range from about 0.2 to 0.65 with the largest value obtained with the low-level crossflow strategy. Q/UA is directly related to the coefficient of pressure difference ΔC_p across the two faces, which itself is dependent on wind direction. One would therefore expect strategy I to have a relatively large value and strategy II to have a small value and this is apparent in the results. The observation of similar values for strategies II and III is somewhat surprising but may be specific to the particular wind direction and roof geometry.

In principle the value of Q/UA has no influence on the equilibrium concentration inside the model, it will simply determine the time taken to reach that concentration. For the case of a single inlet and outlet the internal equilibrium concentration should also in principle be equal to the inlet and outlet concentrations. It is evident from table 1 that this is not precisely the case. The internal equilibrium is dependent on the average concentration at the opening and since the concentration at the intake was sampled at one point only, the difference between equilibrium concentrations is likely to be due to this point being unrepresentative of the average concentration. Another possible cause is some correlation between the instantaneous flow rate and concentration at the inlet.

The variation of the internal concentrations between the different strategies is due to the different location of the inlets. Strategy I has the highest concentration with a low-level intake close to the road. Strategy II, with the intake furthest away from the road, has a concentration about one-third lower. The two strategies with high level intakes give reductions of about twenty percent.

Strategy	Q/UA	Position	Concentration (ppm)
I	0.65	Inlet	285
		Outlet	189
		Inside	**240**
II	≅0.2	Inlet	127
		Outlet	167
		Inside	**159**
III	≅0.2	Inlet	212
		Outlet	143
		Inside	**194**
IV	0.32	Inlet	196
		Outlet	189
		Inside	**193**

Table 1 Wind tunnel results (U=2.5 m/s).

6. Conclusions

Wind tunnel tests indicate that small but significant reductions in internal pollution concentrations arising from road traffic can be achieved by judicious siting of air inlets.

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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OSLO, NORWAY, 28-30 SEPTEMBER 1998

Active Envelopes – Essential in Urban Areas?

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Keywords: active envelopes, double façades, ventilation technologies, energy-efficiency.

Abstract

Today, the development of new technologies to improve building-envelopes performances are highly encouraged and provide a clear challenge for designers and researchers. In this context several typologies of active envelopes are becoming very popular amongst designers and architects. They are favorite choice in offices and many advantages are claimed in the professional literature. Especially in urban areas, designers choose for active envelopes because of the good sheltering from the high external pollution and noise load. Reading some one-sided articles a designer could get the idea that if he uses an active envelope, he automatically gets an energy-efficient and highly comfortable building.

To improve energy-efficiency, active envelopes should act as an active solar collector in summer, decreasing the cooling loads. In winter they should behave like an air-air heat exchanger, recovering heat losses and gather the solar energy to use this energy in the HVAC-system. A higher glass to wall ratio provides more daylight and the extra pane improves the sound insulation. Although active envelopes might offer high potential in improving energy efficiency and in thermal and acoustical comfort performance, the expectations are not nearly always achieved.

The performances of the airflow window of the DVV-headquarters building in Brussels where the subject of a performance based assessment. The paper presents the experimental data and the model used to quantify the thermal and acoustical properties. The attention is drawn on correctly dividing the radiation and convection balances, the impact of a bad workmanship, the unclear meaning of the equivalent U-value and the Solar Coefficient and the importance of absorbing the short wave solar radiation to realize a good solar coefficient.

Concluding, one could say that the high expectations (i.e. the performances) are not always fulfilled due to a wrong choice of typology, doubtful models and bad workmanship. Secondly, designers should be aware that even huge efforts could lead to disappointing small results and that the overall energetic, economical and ecological performance could be very discouraging. Even if, in some cases, the performances are achieved one could ask whether the extra costs count for the, sometimes small, benefits.

Active Envelopes – Essential in Urban Areas?

Dirk Saelens and Hugo Hens

Synopsis

Today, the development of new technologies to improve building envelope performances is highly encouraged and provides a clear challenge for designers and researchers. In this context several typologies of active envelopes have become very popular. The paper starts with an overview of the history and the performances of active envelopes in the context of urban design. With these considerations in mind we will analyse the performance of the active window system of the DVV-headquarters downtown Brussels.

This paper presents experimental data and a model to quantify the physical properties of active envelopes. The experiments clearly demonstrate that workmanship and design have an important impact on the results: the performances, predicted at the design stage, are not always achieved after construction. For instance the DVV Case Study reveals that airtightness and the use of frames with thermal break are extremely important. The attention is also drawn to the importance of a correct split between the convection and radiation when predicting thermal performance, the unclear meaning of the equivalent U-value and the Solar Heat Gain Coefficient and the importance of the absorption of short wave radiation in the cavity.

List of symbols

Lower and upper cases

- c specific heat capacity (J/(kg K))
- d cavity width (m)
- e emissivity (-)
- h heat transfer coefficient (W/(m²K))
- z height (m)
- E absorbed solar energy (W/m²)
- E_{st} total solar radiation (W/m²)
- G airflow rate per unit width (m³/(hm))
- H window height (m)
- R thermal resistance (m²K/W)
- SHGC Solar Heat Gain Coefficient (-)
- U thermal transmittance (W/(m²K))

Greek symbols

- α absorption coefficient (-)
- θ temperature (°C)
- λ thermal conductivity (W/(mK))
- ρ reflection coefficient (-)
- τ transmission coefficient (-)
- ϕ heat flux (W/m²)

Subscripts

- a air
- c convection
- cav cavity
- di direct
- e exterior
- i interior
- in indirect
- r radiation
- s shading device

1. An introduction to active envelopes

1.1 Definition and history of the active envelope

Active envelopes are facade systems, which are designed to act as air-air heat exchangers. They typically consist of two panes, with a cavity in between, which commonly incorporates the shading device. Through the cavity air is drawn by means of natural or forced convection.

In 1849 Jean-Batiste Jobard described the first active envelope concept: warm air in winter and cooled air in summer flowed between two glazed panes. Approximately 65 years later, Paul Scheerbaert describes a similar idea, and in 1930, Le Corbusier develops his so-called

“Le mur neutralisant”: a double skin system for La Cité de Refuge. The first studies on airflow-windows were published in the fifties in Scandinavia. The issue was to improve the energy efficiency and the comfort performance of windows in dwellings. In 1957 the first patent related to airflow-windows was filed in Sweden. In 1967 the EKONO Company built the first office building equipped with airflow-windows in Helsinki. [1]
The spirit for further development is to be found in the energy crises of 1973 and 1979 and the growing environmental awareness. Suddenly energy-efficiency and thermal comfort was no longer an exclusive issue for Scandinavian countries only.

1.2 Active envelopes in urban areas

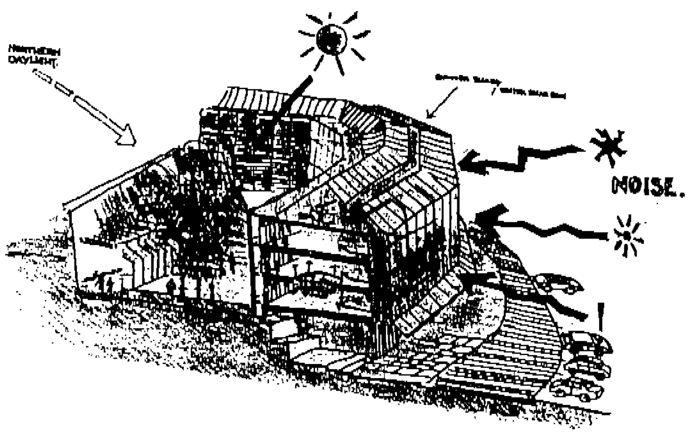


Figure 1: Briarcliff house in Farnborough

Project developers and architects prefer the major urban areas to locate the headquarters. Easy access to public services, a well-developed public transport system and the accessibility from major traffic roads justify this choice. Another, and perhaps more important, motivation is the influence on the project’s prestige and the company’s reputation. Therefore it is not surprising that glass and active envelopes are increasingly applied in prominent office buildings. The building and the company get an attractive, high-tech image and there is no need for an external shading device to ruin that appearance.
The city is a hostile environment. The high noise and pollution loads threaten the comfort in the building. Again active envelopes seem to be the ideal solution: the double skin forms an adequate protection against the aggressive surroundings. Furthermore, active envelopes claim to be energy-efficient and highly comfortable: in winter they collect the solar energy and act as air-air heat exchangers while in summer they protect against overheating by removing the heated air from the cavity. After reading some one-sided articles the designer could get the idea that active envelopes are always the perfect choice.

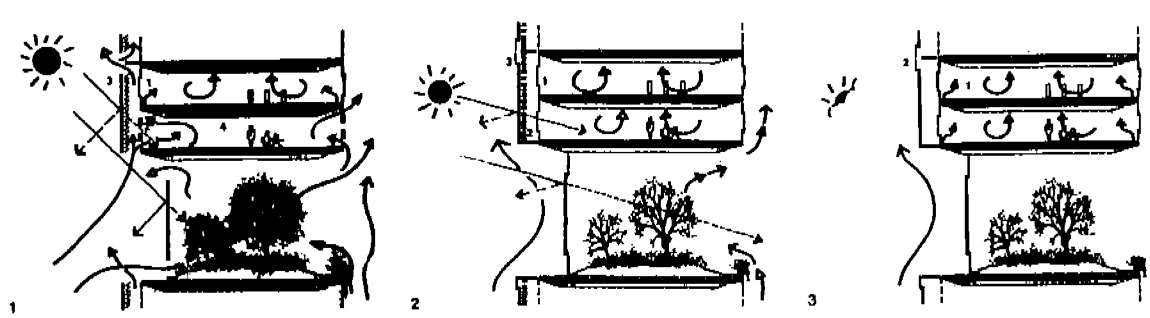


Figure 2: RWE-Tower in Essen, Germany

The first active envelope that catches the attention of the architectural literature was the Briarcliff house [2] (Figure 1). At Farnboroughs, Arup’s designers had to design an office building that could deal with a severe acoustical load: high noise levels from the nearby airport, service roads and the roundabouts the building looks over. From the experience Ove Arup had at the British Sugar Company (1975), the engineers had learned that the double skin idea shelters the building from the sun and the traffic noise. More recently, designers pay attention to the integration of natural ventilation in office buildings by means of active envelopes. Examples are the Commerzbank A.G. in Frankfurt am Main (Germany) (Figure 2) [3] [4] and the Dienstleistungszentrum Stern RWE A.G. in Essen, Germany [5]. Since the eighties, especially airflow windows and airflow facades attracted a lot of attention. The Brussimmo Building in Brussels and the DVV-headquarters in Brussels (Belgium) are two examples. [6]

2. The DVV case study

2.1 Introduction to DVV-headquarters

***Figure 3**: DVV-Headquarters*

The previous paragraph showed that there are numerous reasons for choosing active envelopes in urban areas. Now, we focus on a case study and find out to what extent the claimed advantages are met in reality. The object of this case study is the new extension of the DVV-Insurance Company office building downtown Brussels. (Figure 3) After further expansion of DVV, the existing building became too small and an extension was planned. Construction started in 1993 and the building was finished in April 1995. The new extension contains five office floors, wrapped around an atrium. The active windows form an integral part of the HVAC-system and each window acts as a return-duct for the HVAC-system. The choice for an active envelope was ruled by the wish to achieve optimal energy efficiency, excellent thermal and acoustical comfort and sheltering from the external pollution.
In the DVV-building the conditioned air is injected at floor level. Exhaust is realised through the light armatures, which are connected to the active windows. The exhaust air flows top-down through the cavity and returns to the air conditioning system, where it either is re-used or expelled.

2.2 Modeling and measuring

The model focuses on performances at the envelope level: the U-value and the Solar Heat Gain Coefficient. In the Annex 32 terminology the performances at element level are called level two performances. [7] It is clear that the level two performances are closely related to the building level performances (level one). If, for example, energy efficiency is formulated at level one, requirements on the thermal insulation values are a level two consequence.
In cavities, conduction, convection, radiation and the enthalpy transport through the cavity define the combined heat-mass transport. (Figure 4) Starting point are the heat balances for the three panes and the roller blind and the heat balances describing the enthalpy-transport in the cavities (eq. 1-6). When we calculate the U-value, the coefficients E are set zero to eliminate the solar influence. The U-value is calculated from the heat flux through the inner

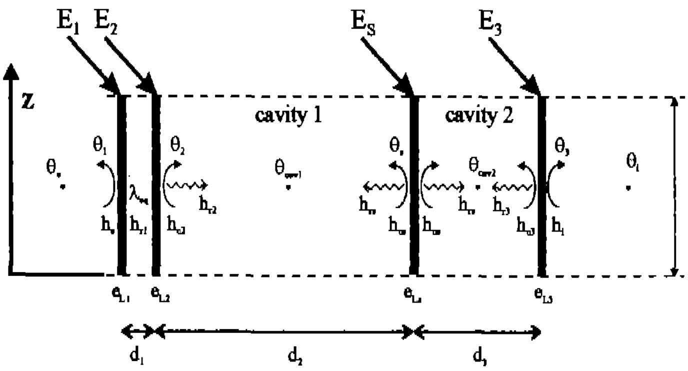


Figure 4: Model diagram

surface (eq. 7 and 8). The SHGC is defined as the ratio of the total energy flux (ϕ_{di} + ϕ_{in}) into the building to the total solar radiation (E_{ST}) (eq 9,10 and 11). The direct gains are also calculated from the heat flux through the inner pane (eq. 10). When calculating the SHGC, only the effect of the solar radiation should be rated: therefore the inner and outer temperatures are set zero ($\theta_i = \theta_e = 0$). The shading device is supposed to be airtight, not allowing any air-exchange between the cavities. The airflow rates in the cavities are supposed to be proportional to the third power of the cavity width.

pane 1	$E_1 = h_e \cdot (\theta_1 - \theta_e) + \frac{\lambda_1}{d_1}(\theta_1 - \theta_2) + h_{r1} \cdot (\theta_1 - \theta_2)$		(1)
pane 2	$E_2 = h_{c2} \cdot (\theta_2 - \theta_{cav1}) + \frac{\lambda_{eq}}{d_2}(\theta_2 - \theta_1) + h_{r1} \cdot (\theta_2 - \theta_1) + h_{r2} \cdot (\theta_2 - \theta_s)$		(2)
shading device	$E_s = h_{r2} \cdot (\theta_s - \theta_2) + h_{cs} \cdot (\theta_s - \theta_{cav1}) + h_{r3} \cdot (\theta_s - \theta_3) + h_{cs} \cdot (\theta_s - \theta_{cav2})$		(3)
pane 3	$E_3 = h_i \cdot (\theta_3 - \theta_i) + h_{c3} \cdot (\theta_3 - \theta_{cav2}) + h_{r3} \cdot (\theta_3 - \theta_s)$		(4)
cavity 1	$\left[h_{c2} \cdot (\theta_2 - \theta_{cav1}) + h_{cs} \cdot (\theta_s - \theta_{cav1})\right] \cdot dz = G_{in} \cdot c_a \cdot d\theta_{cav1}$		(5)
cavity 2	$\left[h_{c3} \cdot (\theta_3 - \theta_{cav2}) + h_{cs} \cdot (\theta_s - \theta_{cav2})\right] \cdot dz = G_{in} \cdot c_a \cdot d\theta_{cav2}$		(6)
heat flux and U-value	$\phi_u = \frac{1}{H}\int_0^H \frac{\theta_i - \theta_3}{R_2}$	$U = \frac{\phi_u}{(\theta_i - \theta_e)}$	(7,8)
direct and indirect gains	$\phi_{di} = \tau_1 \cdot \tau_2 \cdot \tau_s \cdot \tau_3 \cdot E_{st}$	$\phi_{in} = \frac{1}{H}\int_0^H h_i \theta_3 dz$	(9,10)
solar heat gain coefficient	$SHGC = \frac{\phi_{in} + \phi_{di}}{E_{st}}$		(11)

***Table 1**: governing equations*

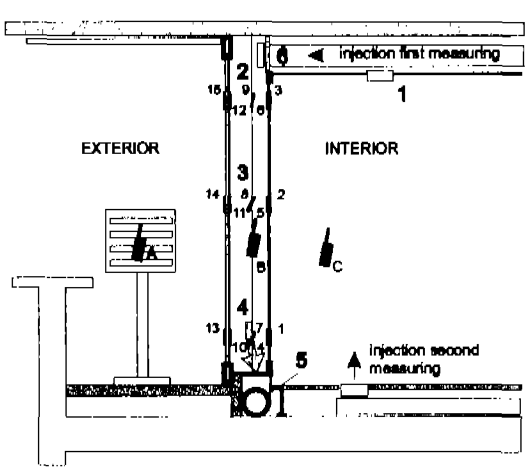


Figure 5: Experiments diagram

The temperature profiles can easily be found by writing an expression for the pane and roller blind temperatures (θ_2, θ_s and θ_3). After substitution of these expressions into the cavity heat-balances (eq 5 and 6), we obtain a system of two coupled differential equations from which we can compute the temperature profiles. A detailed description of the model can be found in the references [6 and 8]. To check the performance of the active window system experimentally, the following measurements were set up on a south-west orientated window:

(1) Temperature profile analysis. Data from 15 thermocouples, shielded from direct solar radiation with aluminium foil, were logged every 15 minutes. *(2) IR- thermography.* Infrared thermography measures the surface temperatures to

reveal thermal bridging and air leakages of the building's envelope *(3) Measurement of the airtightness*. To check the airtightness of the active windows at DVV's, three experiments were set up: (1) airflow-visualisation with smoke-sticks, (2) pressure difference measurement and (3) tracergas measurement. On figure 5, the tracergas measuring points are indicated with bold numbers, the thermocouples with small numbers.

2.3 Main results

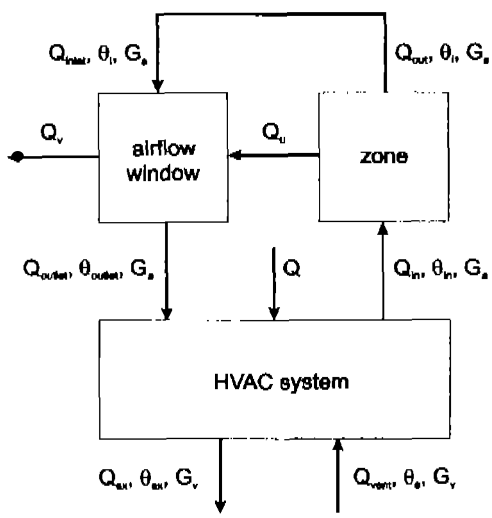
2.3.1 Interpretation of the equivalent U-value and SHGC

Table 2 shows **the importance of dividing the convection and radiation balance**. Models that use a combined surface coefficient underestimate the U-value. [7, 8 and 9]
Under normal conditions the U-value and SHGC depend on the material properties only. The U-value and SHGC of active envelopes, however, are no longer single values, depending on a set of thermal conductivities, thicknesses, solar properties, et cetera. Both quantities are strongly related with the system properties. [7, 8 and 9]

airflow rate	*U-value: use of combined surface coefficients*	*U-value: division between radiation and convection, without shading device*	*U-value: division between radiation and convection, with shading device*
$m^3/(hm)$	*$W/(m^2 K)$*	*$W/(m^2 K)$*	*$W/(m^2 K)$*
40	0.91	1.25	0.86
90	0.54	0.94	0.60
140	0.39	0.82	0.49

***Table 2**: Effect of the model and the shading device [7]*

Correct calculation of the U-value and SHGC is one thing. When discussing performances of active envelopes, **the equivalent U-value and SHGC should be interpreted cautiously**. A low equivalent U-value, for instance, does not automatically stand for a low energy demand. If we consider for instance, the following simplified model (Figure 5) we notice that despite the decrease of the U-value with increasing airflow rate, the energy consumption of the HVAC-system will increase. This is due to the fact that the air, which is reused in the HVAC-system, cools down when passing through the airflow window.



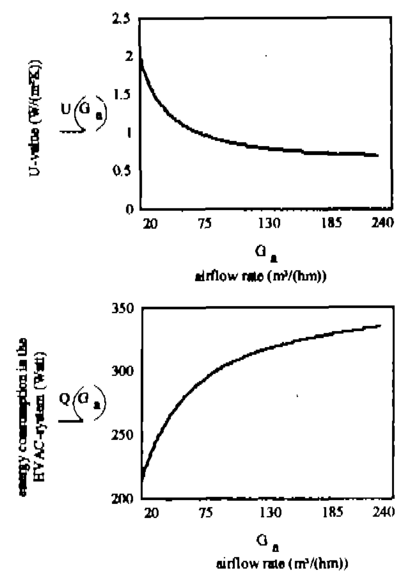


***Figure 5**: Comparison between the U-value and the energy added to the HVAC-system (Q).*

2.3.2 The influence of bad workmanship

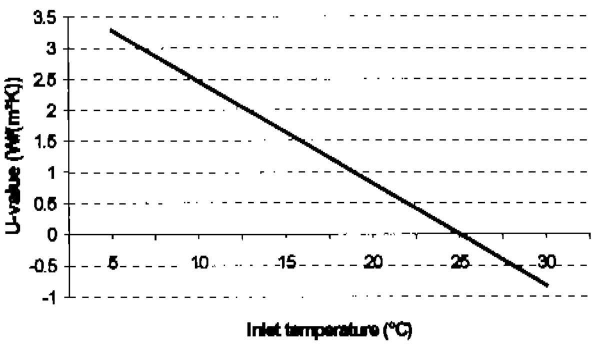


***Figure 7**: The U-value as a function of the inlet temperature*

In practice the calculated performances are not always achieved due to bad workmanship. We can illustrate the influence of bad design and bad workmanship by the influence of the inlet temperature on the U-value. Figure 7 shows that the U-value changes linearly with the initial temperature. Theoretically, even negative U-values are possible for initial temperatures higher then 25 °C. This figure shows that the infiltration of cold air has a disastrous influence on the U-value and stresses the importance of profiles with a thermal break.

At DVV's, the assumption that the inlet temperature equals the room temperature proved to be completely wrong. Passing the lights, the air warms up: so, one could expect a higher temperature at the inlet. Surprisingly, the measured inlet temperatures lay far below room temperatures (Figure 9a). Two hypotheses were made: air from the outside is sucked into the cavity, or thermal bridging may be present.

A tracergas measurement with constant emission rate could not demonstrate a lack of airtightness of the outer pane, so the thermal-bridging hypothesis was more likely. Measurement of the temperature profile in the duct revealed that the air cooled down with 3°C while passing through the duct. This cooling is probably caused by a wrong insulation concept. We suppose the thermal insulation is simultaneously used as acoustical insulation and therefore placed just on top of the acoustical ceiling, not covering the ducts at all.

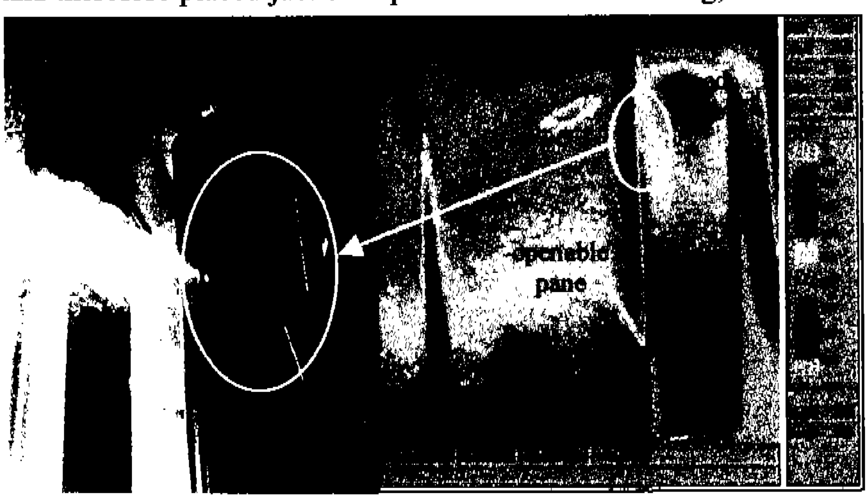


***Figure 8**: Photograph of the smoke-stick experiment and the IR-thermography*

Another design weakness is demonstrated by the bad sealing of the inner pane. Both the smoke-stick experiment and the IR-thermograph reveal this phenome-non. Figure 8 clearly shows how the smoke is sucked into the cavity (a pressure measurement indicated an under-pressure of 3 Pa in the cavity). Figure 8 shows an infrared photograph of the window and also illustrates the effect of bad sealing: the temperature along the right edge of the openable one is much lower then the temperature in the middle of the pane. The air sucked from the room into the cavity is colder and causes a local drop in temperature. Compared to the fixed pane right from the openable pane we do not detect a temperature drop along the edge. The locks were the cause of the bad sealing: they do not properly press the inner pane against the edge.

2.3.3 The active envelope as a solar collector

Solar heat storage and the excellent SHGC without exterior shading device are pointed out as an important feature of active envelopes. In winter the collected solar energy should be used to heat the building. In summer the solar energy is expelled to diminish the cooling loads and the low SHGC should prevent against overheating.

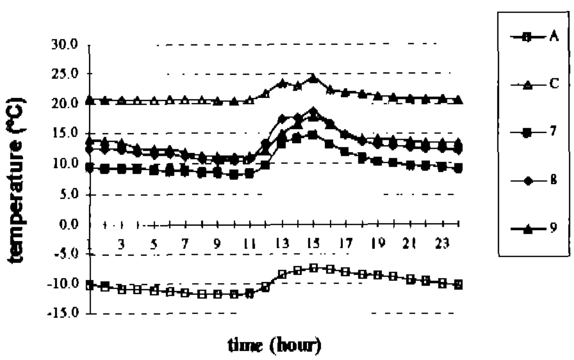
The use as a solar collector is not possible without absorption of solar energy in the cavity. SHGC without sunshading or special glazing in the active system doesn't exceed the SHGC of absorbing double glazing (SHGC = 0.45 to 0.66). Table 3 proves the importance of the airflow rate and the absorption of solar energy in the cavity. The SHGC without sunshading is largely independent of the airflow rate, as glass and air are to a large extent transparent for the short-wave solar radiation.

SHGC	airflow rate $m^3/(hm)$ 0	140	difference
no sunshading, no absorption	0.63	0.56	10%
no sunshading, absorption at pane 2	0.45	0.29	35%
sunshading, no absorption	0.36	0.21	40%

***Table 3**: The importance of the shading device and the absorption in the cavity.*

From the above we understand that absorption of solar energy in the cavity is necessary to heat the air. With the outlet temperature we can describe the solar collector efficiency. The higher the temperature the higher the amount of energy absorbed by the air. To have an idea what the solar heat storage capacity is we will analyse the temperature profiles measured on the first of January 1997. (Figure 9a) (position of the thermocouples: Figure 5) Due to the low solar inclination in winter the sun penetrates deep into the building, not being obstructed by the overhanging roof. The shading system is not in use. To know if the active envelope is suitable to act as a solar collector, we compared the interior and the outlet temperature (point 7 and 9 on Figure 5) (Figure 9b). We discover that the interior air always is hotter then the outlet air. As the area above the base line is a measure for the energy loss in the cavity, in this case the active window always loses energy. This shows that heating the air while passing through the cavity in this case is fiction.

A second point of attention is the temperature drop when the sun does no longer heat the air. Once the air passed the sunlit part, temperature quickly decreases: the air hardly can hold the absorbed heat because of its low capacity.



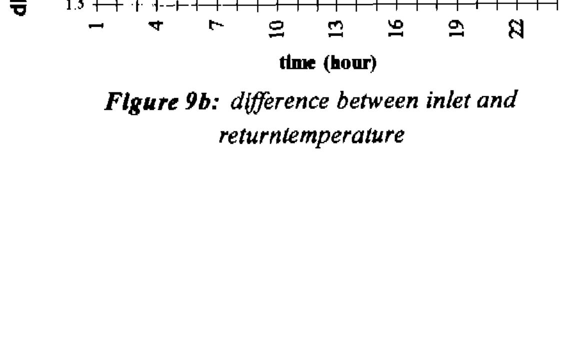


***Figure 9b:** difference between inlet and returntemperature*

3. Conclusions

In this paper we defined the goals of the DVV project in an urban context: the active window with forced convection was designed to shelter the building from the high noise and pollution load. Second reason to apply the double skin concept was energy-efficiency.

In the case study we focused on the envelope level performances. While analysing the model and comparing the results with experimental data we highlighted some misconceptions about active envelopes and stressed that good design and workmanship are essential to fulfil the claimed performances.

We proved that a model that does not divide convection and radiation strongly underestimates the U-value. With a simplified example, we indicated that the U-value and SHGC have to be interpreted cautiously! Low equivalent U-values do not automatically imply low energy demands.

Bad workmanship and poor design have a severe influence on the active envelope performance. Infiltration of cold air and thermal bridging of the window frames dramatically raises the U-value. In the DVV case study we first demonstrated the poor design and the bad workmanship of the duct between the lightarmature and the air inlet. Secondly, we illustrated the inferior design of the locks of the inner pane.

Regarding sunshading and the use of active envelopes as solar collector, we stressed the importance of absorption of the solar radiation in the cavity. Active envelopes without absorption do not collect solar energy nor have a good SHGC. The reason is that glass and air badly absorb the short wave solar radiation and that air has a very low capacity for heat.

4. Acknowledgement

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VENTILATION TECHNOLOGIES IN URBAN AREAS

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OUTLINE OF VENTILATION STANDARD FOR ACCEPTABLE INDOOR AIR QUALITY OF SHASE, JAPAN

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SYNOPSIS

The Ventilation Standard HASS-102 of The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan (SHASE Japan) was revised in November, 1997. The title of the revised standard is Ventilation Standard for Acceptable Indoor Air Quality. So far, the old Ventilation Standard, which was issued in 1939, had been used for a long time. The task for revision was undertaken by the Sub-Committee on Ventilation Effectiveness and Standard (chaired by Murakami) of SHASE Japan. This paper focuses on the frame work of the Standard, design criteria for acceptable concentration of indoor air pollutants, calculation method for ventilation requirement, technical principles for construction of ventilation equipment and so on.

Main points for revision are as follows;

1) The amount of ventilation requirement is obtained by the emission rate and the design criteria for acceptable concentration of indoor air pollutants. In other words, the amount of ventilation requirement is calculated in consideration of the situation of space usage and the conditions of air pollutant generation.
2) The kinds of indoor air pollutant prescribed for in this standard are CO_2, CO, suspended particulate, NO_2, SO_2, HCHO, radon, asbestos and TVOC.
3) Design criteria for acceptable concentration of CO_2 is provided by a general indoor quality index (1000ppm) as well as one of pollutants influencing occupant's health (3500ppm). It describes how to properly use these two indexes for each pollutant source.
4) When pollutants are not perfectly mixed with the room air, ventilation effectiveness is taken into account for calculation of the amount of ventilation requirement.
5) Also prescribed are technical principles for construction of ventilation equipment and test methods of ventilation performance after the construction.

INTRODUCTION

The Ventilation Standard HASS-102 of The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan (SHASE Japan) was revised in November,1997. The title of the revised standard is Ventilation Standard for Acceptable Indoor Air Quality. So far, the old Ventilation Standard, which was issued in 1939, had been used for a long time. The task for revision was undertaken by the Sub-Committee on Ventilation Effectiveness and Standard (chaired by Murakami) of SHASE Japan and started in April, 1994. The final draft was prepared in January, 1997 and opened to public review. After the some modification based on the comments from the public review, the standard was approved in November, 1997.

This paper focuses on the frame work of the Standard, the design criteria for acceptable concentration of indoor air pollutants, calculation method for the amount of ventilation requirement, technical principles for construction of ventilation equipment and so on.

Main points for revision are as follows;

1)The amount of ventilation requirement is obtained by the emission rate and the design criteria for acceptable concentration of indoor air pollutants. In other words, the amount of ventilation requirement is calculated in consideration of the situation of space usage and the conditions of air pollutant generation.

2)The kinds of indoor air pollutant prescribed for in this standard are CO_2, CO, suspended particulate, NO_2, SO_2, HCHO, radon, asbestos and TVOC.

3)Design criteria for acceptable concentration of CO_2 is provided by a general indoor quality index (1000ppm) as well as one of pollutants influencing occupant's health (3500ppm). It describes how to properly use these two indexes for each pollutant source.

4)When pollutants are not completely mixed with the indoor air, ventilation effectiveness is taken into account for calculation of the amount of ventilation requirement.

5)Also prescribed are technical principles for construction of ventilation equipment and test methods of ventilation performance after the construction.

Due to revision of 1), 2) and 4), building service engineers or designers are requested to collect the data such as the emission rate of air pollutants and to judge the mixing condition of air pollutants, etc. On the other hand, it becomes easier for designers to select alternative ventilation equipment based on detailed investigation and it can be expected that the more appropriate ventilation design will be performed for various conditions of building indoor environment.

Member lists of the Sub-Committee on Ventilation Effectiveness and of SHASE Japan and the task group for revision of the Standard are following,

Sub-Committee on Ventilation Effectiveness:

Chair: Shuzo Murakami

Secretary: Shin Hayakawa, Shinsuke Kato and Hiroshi Yoshino

Member: Shin-ichi Akabayashi, Masamichi Enai, Hitoshi Fukao, Koichi Ikeda, Jun Kagawa, Motoyasu Kamata, Hikaru Kobayashi, Nobuyuki Kobayashi, Yasushi Kondo, Takashi Kurabuchi, Hideaki Matsuki, Hiroshi Matsumoto, Hiroyuki Matsuura, Futoshi Mihashi, Kunio Mizutani, Shin-ichiro Nagano, Yoshiaki Nagasawa, Satoshi Nakamura, Toshiaki Ohmori, Shigeki Ohnishi, Hiroyasu Okuyama, Takao Sawachi, Yukinori Shibuya, Yoshimi Suyama, Toshihiko Tanaka, Kakuo Ueda, Yasuo Utsumi and Toshio Yamanaka (Alphabetical order)

Task Group for revision of the Standard:

Chair: Nobuyuki Kobayashi

Co-chair: Koichi Ikeda

Secretary: Shin-ichi Akabayashi and Yasushi Kondo

Member: Jyun Kagawa, Kunio Mizutani, Shuzo Murakami, Toshiaki Ohmori, Shieki Ohnishi, Hiroyasu Okuyama, Kakuo Ueda, Yasuo Utsumi and Hiroshi Yoshino (Alphabitical order)

1. PURPOSE AND SCOPE

This standard prescribes the amount of ventilation airflow volume required for maintaining acceptable indoor air quality and the technical criteria on mechanical ventilation equipment for providing this airflow volume. The standard applies to the ordinary indoor environment mechanically ventilated such as habitable rooms, office spaces, attached spaces to these rooms and the spaces for various facilities.

2. CONCEPT AND CALCULATION METHOD FOR BASIC VENTILATION REQUIREMENT AND DESIGN VENTILATION REQUIREMENT

2.1 Principles of calculation method

The amount of required ventilation airflow volume is obtained by two steps in order to keep indoor air pollutant concentration less than the acceptable level. Firstly, the amount of required ventilation airflow volume (basic ventilation requirement) is calculated by Equation (1), assuming that the pollutants are completely mixed with indoor air and the situation is steady state.

$$Qp = \frac{M}{Ci - Co} \quad (1)$$

Qp: Basic ventilation requirement [m^3/h]
M: Emission rate of indoor air pollutant [m^3/h]
Co: Pollutant concentration of supplied outdoor air [m^3/m^3]
Ci: Design criteria for acceptable concentration of indoor air pollutant [m^3/m^3]

When several pollutants are generated in a space, the kind of pollutant and it's emission rate for each pollutant source should be investigated. Then the amount of required ventilation airflow volume for each kind of pollutant shown in Table 1 is calculated by Equation (1). The maximum among these values should be the basic ventilation requirement.

However, there is another idea that the basic ventilation requirement should be the sum of ventilation airflow volume required for each pollutant. Because it is expected that the health effect by each pollutant will accumulate in the body. But there is no consistent medical knowledge that the several low level pollutants found in ordinary indoor environment give accumulative health effect. Also, there is no criteria internationally agreed for this problem. Therefore it was decided that the maximum value among required ventilation airflow volume obtained for each pollutant is the basic ventilation requirement.

Secondly, the amount of modified required ventilation airflow volume (design ventilation requirement) is obtained by correcting the basic ventilation requirement, considering the actual mixing condition of the pollutants. In the case of the complete mixing condition, the design ventilation requirement is equal to the basic ventilation requirement. In the other cases, the basic ventilation requirement is increased or decreased, taking account of ventilation effectiveness such as the collection efficiency of a kitchen hood for polluted air from a gas stove.

2.2 Design criteria for acceptable concentration of indoor air pollutants

Table 1 shows the list of air pollutants found in habitable rooms, office spaces and so on, as well as the design criteria for acceptable concentration of these pollutants. When there will be a pollutant not indicated in Table 1, it is necessary to examine the emission rate and concentration in the outdoor air, and then to determine the amount of ventilation airflow volume to keep the pollutant concentration less than one-tenth as much as the standard value recommended as the acceptable concentration in a working environment by the Japanese Society for Occupational Health.

2.3 Design Criteria for Acceptable concentration of CO_2

For a general indoor environment, CO_2 has been used for a long time as an index of indoor air quality. The law for Building Sanitation Management Standards by The Ministry of Health and Welfare states that the CO_2 concentration should be under 1000ppm. This is not based on the health effect of CO_2 alone but on the environmental experience that, when CO_2 concentration is higher than 1000 ppm, other pollutant concentrations are also higher. As a result, it was decided that the design criteria for an acceptable concentration of CO_2 should be 1000ppm as a general indoor quality index.

On the other hand, literature[1)] shows that health effects appear when the CO_2 concentration is above 7000 ppm. It is prescribed in Canada[1)] that the acceptable concentration of CO_2 is 3500 ppm for residential buildings. Then, the value of 3500 ppm is adapted provisionally for the design criteria for an acceptable CO_2 concentration.

3. CALCULATION METHOD FOR VENTILATION REQUIREMENT IN OCCUPIED SPACE

The calculation method for the required amount of ventilation is described as follows. The flow chart is shown in Figure 1.

(1) The first step is to investigate what kind of air pollution is generated in a space.

(2) The second step is to make clear whether there is pollutant generation from source other than occupants.

(3) If there is no pollutant generation other than the occupants, the required ventilation airflow volume can be calculated on the basis of the design criteria for acceptable concentration of CO_2 (1000 ppm) as a general indoor quality index. In this case, building materials are assumed not to emit any kind of pollutants.

(4) If there are other pollutant sources, the next step is to investigate the kind and the emission rate of pollutants from each source. After that, the pollutant sources, which are identified in terms of the kind and the emission rate of all pollutants, shall be separated from the pollutant sources without complete identification for kind and emission rate. The words of "pollutant sources which are identified in terms of the kind and the emission rate of all pollutants" means those sources which should satisfy that the emission rate of all pollutants indicated in Table 1 are identified and there is no other pollutant or, if there are other pollutants, the kind and the emission rate of those are identified.

(5) For the pollutant source which is not identified in kind and emission rate, the amount of required ventilation airflow volume is calculated, based on the design criteria for acceptable concentration of CO_2 (1000ppm) which is provided for the general indoor quality index. That means it is expected that the ventilation requirement based on CO_2 of 1000 ppm can dilute the air pollutants from these sources to the level of no health effects.

(6) For each kind of air pollutant identified in emission rate from all sources, the amount of ventilation airflow volume is calculated, based on the design criteria of acceptable concentration which is provided for a single indoor quality index. In this case, the design criteria for an acceptable concentration of CO_2 is 3500ppm.

(7) The basic ventilation requirement is then the maximum value among the required ventilation air flow volumes obtained in (5) and (6).

(8) The normalized concentration in an occupied zone is calculated in consideration of the mixing condition of indoor air.

(9) The design ventilation requirement is obtained by multiplying the basic ventilation requirement by the normalized concentration.

4 . DESIGN VENTILATION REQUIREMENT UNDER THE INCOMPLETELY MIXED CONDITION.

4.1 Calculation method for normalized concentration in occupied zone

The normalized concentration in an occupied zone is the ratio of the difference between the average pollutant concentration in an occupied zone and that of the supplied air to the pollutant concentration difference if the indoor air is completely mixed and at steady state.

Namely, the normalized concentration in an occupied zone, Cn, can be calculated by the following equation.

$$Cn = \frac{Ca - Co}{Cp - Co} \qquad (2)$$

Cn: Normalized concentration in an occupied zone[-]

Ca: Average pollutant concentration in an occupied space [m^3/m^3]

Co: Pollutant concentration of the supplied outdoor air [m^3/m^3]

Cp: Pollutant concentration of the completely mixed indoor air $Cp = M / Qp + Co$ [m^3/m^3]

M: Emission rate of indoor pollutant [m^3/h]

Qp: Basic ventilation requirement [m^3/h]

The average pollutant concentration in an occupied zone, Ca, is calculated by either experiments or the method of CFD(Computational Fluid Dynamics).

4.2 Calculation of design ventilation requirement.

The design ventilation requirement, which is required for maintaining the average pollutant concentration in an occupied zone under the design criteria can be calculated by the following equation.

$$Q = Qp \times Cn \qquad (3)$$

5. DESIGN CRITERIA OF VENTILATION REQUIREMENT FOR ATTACHED SPACES AND THE SPACES FOR VARIOUS FACILITIES

In principle ventilation airflow volume required for attached spaces and the spaces for various facilities should be satisfied in order to keep pollutant concentration under the design criteria. It is, however, sometimes difficult to know the emission rate of the air pollutants because there exists various kinds of indoor air pollutants in these spaces. In such cases, the value shown in Table 2 can be used as the approximate ventilation requirement (indicated as air change rate), considering the purposes for the ventilation and the generating condition of indoor air pollutants in those spaces.

6. CONCLUSIONS

This paper described the revised Ventilation Standard HASS-102. The most important point is to calculate the required ventilation airflow volume by two kinds of CO_2 concentration. One is obtained by the design criteria of acceptable concentration for the general indoor quality index of CO_2, and the other one is obtained by the design criteria for the single index taking into account the health effects of CO_2. This way leads to the prevention of an occupant's health suffering from air pollution, even if the kind and emission rate of all indoor air pollutants are unknown. The other important point is to indicate the calculation method to take into account the ventilation effectiveness for the incompletely mixed condition.

In future, discussion will be continued to improve the revised Standard on the basis of accumulative products of research works and technology development.

REFERENCE

1) Federal-Provincial Advisory Committee on Environmental and Occupational Health: Exposure Guidelines for residential Indoor Air Quality, (1989), p.8

Table 1 Acceptable concentration of indoor air pollutants

(a)Pollutant as a general index for indoor air quality and acceptable concentration

Pollutant	Acceptable concentration	References
CO_2*1	1000 ppm	Building Sanitation Management Standards in Japan

(b)Pollutants as independent index and acceptable concentration

Pollutants	Acceptable concentration*3	References
CO_2	3500 ppm	Criteria in Canada
CO	10 ppm	Building Sanitation Management Standards in Japan
Airborne particle	0.15 mg/m^3	Same as above
NO_2	210 ppb	One hour value by WHO*2
SO_2	130 ppb	Same as above
HCHO	80 ppb	Half hour value by WHO
Radon	150 Bq/m^3	Criteria by EPA*2
Asbestos	10 piece/l	The Air Pollution Control Act of the Environment Agency in Japan
TVOC	300 $\mu g/m^3$	Criteria by WHO

**1. Acceptable concentration of 1000ppm for CO_2 is not based on health effects of CO_2 itself but on a general index for indoor air quality. Namely, the value is applied when the emission rate of individual pollutant cannot be estimated, assuming that the pollutant level increases in proportion to the increase of CO_2 concentration. In the case that all the emission rates of indoor pollutants and their acceptable concentration are known, it is not necessary to apply the value of 1000ppm for CO_2. The value of 3500ppm based on health effects can be used in this case.*

**2. The abbreviations in this table are shown in full spelling as follows;*

WHO: World Health Organization

EPA: Environmental Protection Agency

**3. The values of volume concentration expressed in units of ppm and ppb are converted under the conditions of 25℃ and 1 atm from the mass concentration.*

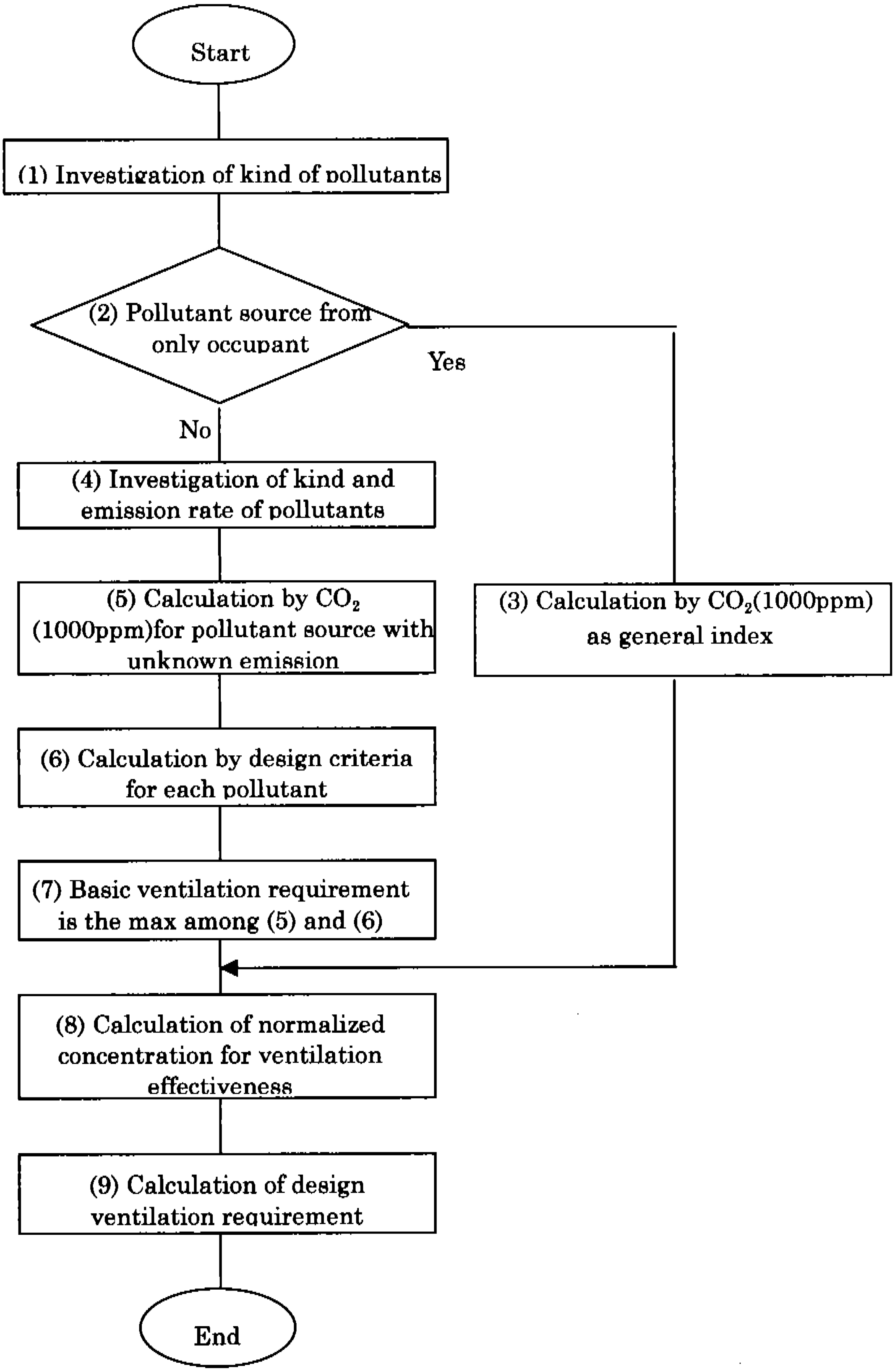

Fig.1 Flow chart for calculation method of ventilation requirement

Table 2 Ventilation requirement for attached spaces and the spaces for facilities

Name of space	Purpose for ventilation					Ventilation systems [*4]			Air change rate (1/h)
	Odor	Heat	Humidity	Noxious gas	Supply of O_2	A	B	C	
Rest room	●						×		5~15
Dressing-room	●						×		5
Kettle room		●	●		●		×		40~20kQ [*1]
Stack room / Storehouse	●	●	●				×		5
Dark room	●	●					×		10
Copy room	●	●		●			×		10
Bath room			●				×		3~7
Kitchen		●	●		●		×		40~30kQ [*1]
Kitchen for business	●	●	●	●	●		×		40~20kQ [*1]
Boiler room		●			●			×	(10~)
Refrigerator room				●			×		(5~)
Electric room		●					×		(10~15)
Generator room		●			●		×		(30~50) [*2]
Elevator machine room		●					×		(10~30)
Parking area				●			×		(10) [*3]

●*: Factors to be taken into consideration*

×*: Not used in general*

**1: Type I hood often used for home kitchen: 30KQ*

Type II hood often used for the kitchen for business: 20KQ

K: Burned gas volume (m^3/kcal)

Q: Heat generation of a stove (kcal/h)

**2: Five times per hour in the case of not operating*

**3: According to the regulation by law air change rate shown by parenthesis should be used in the initial design stage. In the final design stage, ventilation requirement should be calculated on the basis of heat generation, allowable temperature rise and regulation.*

**4: A: Mechanical supply and exhaust, B: Mechanical Supply System, C: Mechanical Exhaust system*

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The Air Infiltration and Ventilation Centre provides technical support in air infiltration and ventilation research and application. The aim is to provide an understanding of the complex behaviour of the air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

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Discussion Papers
and
Additional Presentations

Contents

Section 1: Questions and Answers from Papers during Discussion Sessions

SESSION 1: Modelling & Indoor Air Quality

[redacted]

Question from: G Hunt
For the radial diffuser, does the directionality of the flow depend upon the Archimedes (or Froude) Number? (The Froude number $Fr = \frac{u}{\sqrt{g\frac{\Delta T}{T}H}}$ *) ie. does flow spread radially for Fr<<1 and have directionality for Fr>>1?*

Answer:
The Froude number for the outflow was for all tests less than 0,5. The directionality was not however, correlated to a Froude number based on the height of the diffuser, but on the quantity parameter $\frac{g\beta\Delta T_0}{\left(\frac{q_v}{L}\right)^2}$

Question From: D Etheridge
Are you saying that a non dimensional parameter depends on a dimensional quantity (ΔT) rather than on another non dimensional parameter (Archimedes number)? To obtain a general relationship it is preferable to use non dimensional parameters.

Answer:
The reason that ΔT_0 seems not to be a non dimensional parameter at first glance is that the thermal expansion coefficient β is left out. Formally the parameter should be written $(\Delta T_0\beta)$. The result is, however, the same because $\beta = \frac{1}{T}$ does not vary much in the tests. The Archimedes number was tried but the test results did not correlate with this parameter.

[redacted]

Question from: J Axley
(i) Was the solid domain modelled microscopically?
(ii) How was convergence evaluated for the non stationary computation considered?
(iii) Have you compared your results to any measured data?

Answer:
(i) Yes
(ii) Convergence determination in this kind of problem is more an art than a science. The simple answer is that we ran the simulation long enough to assure us that it had reached a stationary state. We used a variety of "tricks" to speed up the process.

(iii) We did compare our data to laboratory data taken by Texas A&M. Some scaling was required to make the comparison, and the ranges of their data and ours were not the same. For the overlap range,(geometry 1 for us) there was reasonable agreement. We have no full scale data with which to compare, but we would like to undertake such a full scale experiment.

Question from: K T Andersen

Could you in your CFD calculations take into account that, as time goes by, the insulation layer is cooled down by the infiltration air resulting in an increased heat loss through the wall and a decrease in heat recovery?

Answer:

Yes we could and we did. It is important to keep a correct accounting system to get the correct total of conductive and convective loads. You must select a control plan and reference temperature consistently for both effects and for both infiltration and exfiltration. Cooling the insulation may or may not have a negative impact on the energy use.

Question from: H J Poh

(i) What type of CFD commercial package is used to solve the problem?
(ii) Is the software installed in a PC or a Mainframe?
(iii) What are the boundary conditions and domain?

Answer:

(i) We used the commercial code "FLUENT".
(ii) We used it both on a PC and a Supercomputer. We did some runs with large numbers of nodes on the Supercomputer but most all of our reported data came from PC runs. For the 30,000 grid points we typically use, the PC was actually faster than the Supercomputer.
(iii) We sought to apply pressure boundary conditions. The limitations of conventional CFD codes made this difficult to apply in practice without using some tricks. The analysis was done separately, for each wall and then combined. To do so it was necessary to use compatible boundary conditions.

Question from: D Stevens

What is the dimension of the crack/opening compared to the wall?

Answer:

The cracks were sized to give a leakage rate similar to that seen in a typical (leaky) US house. The crack height was in the order of 1cm.

It should also be noted that the analysis was 2-dimensional so that a crack is modelled as a slot.

Question from: T H Dokka

There seems to be no temperature gradient in the lower part of the wall (CFD-computation). Is this right?

Answer:
The effect you are seeing is due to the fact that in some cases the cold incoming air falls along the inside face of the wall. In such a case most of the inside-outside temperature drop happens through this air layer and the gradient through the wall is reduced.

This effect occurs for an undisturbed flow and could be significantly altered in a real condition if there is a radiator or forced air movement disturbing the boundary.

Normally, we would not expect much heat recovery from a "straight-through" leakage path and we were a bit surprised to see it in the results. I believe, however, that a significant part of the calculated effect is due to this boundary layer effect.

Question from: H Yoshino
I'm interested in ventilation requiremenst. How do you decide the ventilation requirement in the three cases?

Answer:
The ventilation requirement used in the three designs is taken from the Norwegian Building Code.

Question from: S Aggerholm
How does the programme calculate the additional cost for the improvement of the building or the systems?

Answer:
The user has to estimate the investment cost and eventual increased maintenance level of the measurements since this is specific for each project.

Question from: D Saelens
(i) When discussing the energy efficiency of buildings, you only focus on the system performances. Isn't it important to also assess the envelope performance?
(ii) When removing the acoustical ceiling to use the thermal mass of the building, isn't it possible that you would have acoustical problems? Does the program make an overall assessment or does it only focus on the energy performance?

Answer:
(i) In Norway there has been a lot of attention to reduce the envelope losses resulting in rather rigid requirements for the U-values (see paper). Further reduction in envelope losses is seldom cost effective.
(ii) The program has no assessment of the acoustic environment but my experience is that, in normal office rooms, it is possible to achieve a good acoustic environment without an acoustic ceiling.

[redacted]

Question from: G Hunt
Can you comment on whether the case of a diffuser with no swirl would improve/reduce air exchange efficiency/local age of air?

Answer:
The paper relates to the performance of a commercially manufactured class of unit designed to impart swirl into the supply air. I have no first hand knowledge of how distribution would change if used without swirl vanes. We have observed that at relatively large θ, the angle made with the normal to the surface, $(40° < \theta < 60°)$ equilibrium / Coanda effects cause capture of the air by the horizontal surface. This fundamentally changes the distribution of the air and hence local values of mean age as well as O/A air change efficiency.

I would expect similar results with no swirl. However, for small angles, θ, the swirl may impart some stability into the column of fresh air prior to mixing taking place.

[redacted]

Question from: M Santamouris
I found the method very interesting. Can you explain how the method works with large openings and in particular in cases where there is a bidirectional flow?

Answer:
I have yet to consider large openings. However, given a macroscopic model of flow through large openings loop equations may directly be formed and included in the design evaluation of component sizes. Such macroscopic models are available. When a large opening is located through the building envelope modelling wind effects presents some special problems.

Question from: A Binamu
It seems that in the considered system only wind moving from one direction is considered. How can the system be used in a case where wind is moving from different directions?

Answer:
The example used was based on a similar example presented in the CIBSE applications manual AM10:1997 "Design of Natural Ventilation Systems in Non-Domestic Buildings". It is the position of the CIBSE manual – a position I share – that designers should consider a small number of environmental conditions, or "design days", representative of critical conditions early in the design of a building. As design progresses, the designer should consider using annual simulation to evaluate system performance more completely. With that

said, if a number of wind directions appear to be critical – the designer should consider a number of global geometry and system topology strategies that will provide natural ventilation that is relatively insensitive to wind direction (e.g. employing stacks with terminals located above the highest point in the building, using directionally insensitive stack terminals such as H-POT configurations, and linking air inlets on all elevations to a common plenum or low resistance duct network in the interstitial space between building levels).

With these strategies in hand, a building idealization may be developed, loop equations may be formulated for each of the relevant wind directions as well as still air conditions, and the limiting asymptotes determined. At this point the designer would proceed as before. Of course, if the basic strategies used to avoid directional sensitivity are not effective, the resulting loop equations may not admit a design solution.

SESSION 3: Ventilation Performance & Building Airtightness

[redacted]

Comment from: J Axley
I feel it is important to note – as often noted in the ventilation field – that of the five pollutants considered for IAQ assessment, only one pollutant is directly and unambiguously associated with health (ie. as distinct from comfort) and that pollutant is passive smoking which, within the residential context, is directly under the control of the "masters" of the house. Importantly, bioaerosols were not considered.

Response:
The other pollutants can also be related to health even if it is difficult to quantify:
1. Emission from room walls and furniture (even if not taken into account directly).
2. Due to human metabolism. CO_2 is here only a well known tracer, but it represents all human bioeffluents. It can be assumed that a high level of CO_2 will increase the risks of illness transmission between the members of the family.
3. Cooking activities can be related to CO or NO_x emission with known impact on health.
4. Air dryness is more related to comfort but can have an impact also on eyes.
5. (Passive smoking).

Another point not mentioned is the length of the period with low humidity for rooms, which can be related to house dust mites survival (no clear quantification of that was found).

[redacted]

Question from: D Stevens
Did you consider any HRV's with treated paper cores to allow for latent heat transfer?

Answer:
No we didn't. All HRV we tested were aluminium and plastic heat exchangers, common on the French market. Condensation tests were run on a plastic cross-plate heat exchanger only.

Question from P Op't Veld
What is the influence of leakage in heat recovery? Is it taken into account?
Note: EN308 only gives limitations of leakage under test conditions. A better approach is correction for leakage (like the Dutch standard NEN5138). In practice pressures in compartments of HRV can differ (ie. in case of imbalance) and give other situations than in test conditions. Moreover, EN308 is not written for domestic appliances.

Answer:
EN308 is written for industrial heat exchangers but the European draft of TC156/WG2/AH7 on residential HRV refers to EN308 for test conditions.
It is indeed right that test conditions cancel the influence of leakage but artificially because, of course, leakage does happen when running. Instead of correcting it, we could recommend to improve leakage characteristics. The CEN draft allows quite low values of external and internal leakage but the HRV we have tested hardly fulfills them mainly for pressure levels requested in the draft.

[redacted]

Question from: P Ajiboye
You ranked the order of the ratio of the type of problems with ventilation systems. Is this rank the same for all the ventilation systems examined?

Answer:
The frequency of most common faults of the inspection positions are given for the different systems in the table below. The answer to your question is that it is not the same for all systems. But highest for all systems are the "Air flow" inspection position.

Most common faults in the different system types:

Rank all	Inspection position	MSE-X	MSE	MEO	PSV
1	Air flow (function)	1	1	1	1
2	Other functions	2	2	3	2
3	Handling instruction	3	3	2	4
4	Air intake (supply)	8	9	5	7
5	Device,duct (funct)	-	16	4	3
6	Drawings	9	4	7	9
7	Ducts (exhaust)	4	12	9	8
8	Fan (function)	-	7	6	6
9	Control	7	8	9	14
10	Duct (supply)	-	6	9	11
11	Device (exhaust)	6	13	14	12
12	Device (supply)	10	17	8	5
13	Filter (supply)	10	5	12	-
14	Fan (exhaust)	4	14	14	15
15	User opinion	12	18	16	9
16	Others (exhaust)	-	15	13	13
17	Fan (supply)	-	10	17	-
18	Filter (exhaust)	12	11	17	-

Key: a dash (-) indicates no faults found for this inspection position

Question from: J Axley
The overall objective of the compulsory checking program was to improve public health. Was any correlation between health problems and failure to pass the ventilation system check observed?

Answer:
The overall objective was not evaluated. The program will need to be in force for some considerable time (10 years) before being able to see a measurable change. Around 1990 the ELIB study was carried out and measurement was made before the bill was passed in Parliament. However, the study was only carried out in dwellings.

This study was concentrating on the technical side, so that at least the systems were working according to the requirements at the construction time or at major renovations. A study of the relationship between the health and technical performance should be carried out in the near future. So, in short, the answer is no, as it was not the objective of this study.

[redacted]

Question from: J Andersson
What is the main reason for not using plastic film as a vapour barrier any more? It seems to be an excellent solution as you are comparing all the other alternatives to it.

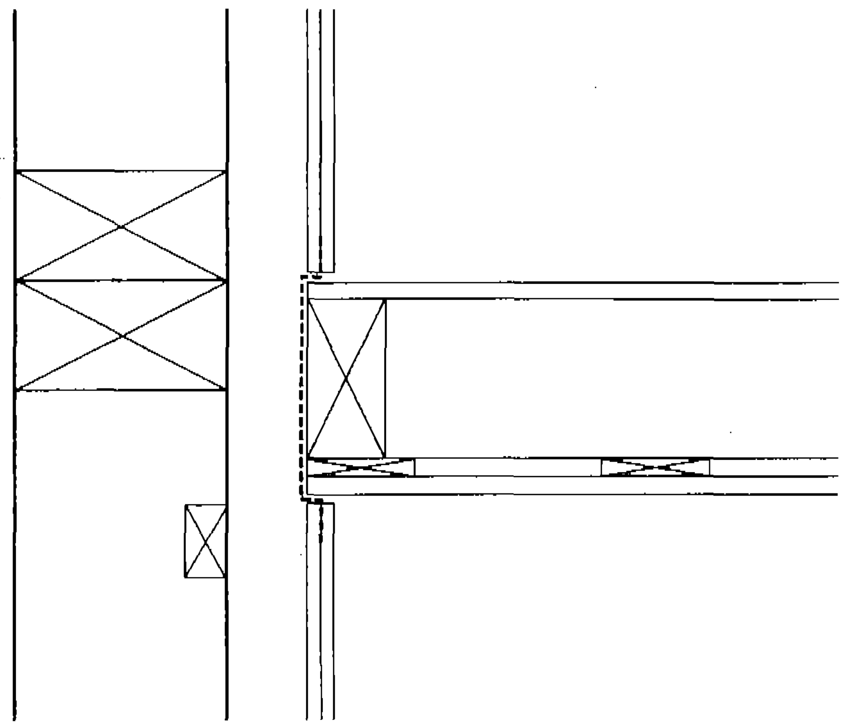
Answer:
Yes it is an excellent solution both for airtightness and as a vapour barrier. We cannot see any good reason not to use plastic film. Although there are people in Sweden asking for solutions where plastic film is not used, probably because of a belief that it is an environmental problem.

Question from: B Borresen
Does your evaluation differ between inside and outside tightness or just the overall tightness? A high overall tightness will be a disaster if the inside tightness is bad.

Answer:
In the field study the measured airtightness in the finished building includes the building envelope airtightness. That means that the airtightness of the whole structure is measured. In the laboratory tests the airtightness was measured on the inner sealing layer.

Comment from: J Axley
In the large element test rig, one particular framing detail (i.e. the lack of top-plate at the top of the lower level wall to which the gypboard could be connected) would lead to joint instability and thus leakage. This single observation is made to make the point that framing strategies, in addition to sheathing and sealing strategies, may have a significant impact on airtightness.

Reply:
A "wooden stud" is used behind the gypsum board in the structure used in the laboratory. See diagram below:



SESSION 5: NatVent™ – Overcoming Technical Barriers

Question from: J Andersson
Could you describe how the cost comparison between natural and mechanical ventilation systems was made? Was it only based on the systems themselves or did they also include, e.g. increased room heights eventually needed for the natural ventilation? Will it be commented upon in the report?

Answer:
The costs were the interviewees perception in general of the cost for natural versus mechanical ventilation, so no calculations are involved. Several of the interviewees also emphasised that the installation cost (for natural ventilation systems) could be high if it includes extra space e.g. extra room height, large ducts, atria etc. These viewpoints are also in the report on barriers from the NatVent™ project.

Comment from: B Borresen
You are referring to good indoor air quality as approximately 1 ach. Norwegian governmental guidelines, when low emitting materials are used, is typically 2.5-4 ach (1l/s.m^2 floor area + 7l/s per person). Governmental working environment inspectors are critical if 3ach minimum are not achieved.

Reply:
The value given of 1ach was just as an illustration. It can be higher. However, it is a building weighted average including corridors and toilets.

Question from: Johnny Andersson
(i) *During periods with high external pollution levels you showed that "closed air intakes during rush hours" was an alternative. How is ventilation achieved for those rooms during that period?*

(ii) *How much maintenance is needed for cleaning or exchanging the filters for the air intakes you showed as a prototype?*

Answer:

(i) When natural ventilation systems are shut down ventilation can be provided by 'back-up'mechanical systems.

(ii) This is an ongoing, but important issue. Experiments are being conducted to determine the 'life-time' of units.

[redacted]

Question from: D Etheridge

In the example you showed, when you switched to controlled inlets the direction of flow changed for the leeward vent. Is there something else going on in the building e.g. extract fans?

Answer:

Yes, in that particular case, extract fans are installed in the corridor.

Question from: J Axley

Have you had sufficient time to evaluate the performance of the TNO device over time? Has there been any loss of performance over time due to fouling, corrosion etc.?

Answer:

We did some field evaluation tests in our own office for a period of three months and in some houses over a period of some years. Although not actually measured after this period, the users did not see any negative effect on the performance. Fouling and corrosion does not seem to be a problem. The control unit can easily be removed and cleaned.

The only problem we became aware of was a minor noise problem. For wind directions parallel to the façade and controlled air inlet the outside shape works as a flute of an organ. This sometimes gave some noise problems even when the inlet was closed manually.

Question from: D Stevens

Given there is no filter, is there a problem with plating of particulates on the inside of the face of the TNO pressure controlled inlet?

Answer:

Up until now, we have not seen any problems of this kind. The control unit can easily be removed and cleaned by the user. I have had two of these inlets in my own house for the past two years and they have never been cleaned. I have not observed any negative effects on the performance. To be honest, I did not check the performance by measurements.

SESSION 6: Cooling & Indoor Air Quality in Commercial & Public Buildings

Question from: H M Mathisen
In the presentation you concluded that it was "relatively well perceived air quality" (or something similar) in the school. My question: Relative to what (mechanically ventilated schools, old schools etc)?

Answer:
Marie Hult, in charge of the referred to questionnaire survey, was also involved in a similar study of 17 healthy schools and day care centres, all with balanced ventilation.

Question from: P Ajiboye
How serious was the microbial growth problem and over what time scales?

Answer:
There is a risk of spread of spores and smell of mould into the classrooms when there is microbial growth in the underground supply duct. The risk of problems due to mould growth and the timescale is dependent on the choice of materials in the underground duct and on the cleaning of it. If the duct is insufficiently cleaned there will be more and more particles and "dirt" on the surfaces where microbial growth can occur. The moisture level will most probably be high enough for microbial growth every spring/summer. We would like to follow up what will happen in those underground ducts after some years of use.

Question from: B Borresen
I have been involved with several measurements covering ventilation efficiency etc for schools. Your presentation indicated low air flows compared to Norwegian guidelines. However, the temperature control seemed well, even on hot days. Investigations referred to recording how satisfied the students/teachers were, seems very positive compared with mechanically ventilated schools. Could it mainly be caused by the fact that the natural vented schools were small, were new, look nice and that the local community and parents were very active?

Answer:
You've got a point there. Many of the modern Swedish passive stack ventilated schools have a very pleasant interior: high ceiling, well chosen materials, nice colours, good daylight, good artificial lighting etc. The schools are often located in quiet areas. These factors might influence the perception of the indoor environment. It would be interesting to repeat the questionnaires after a couple of years, as most of the referred questionnaire surveys were carried out when the schools were fairly new.

The school labelled Z was using night cooling through the underground duct, which contributed to the reasonable indoor temperature during the spring monitoring period.

Question from: P Ajiboye
How relevant are the results for particles up to PM_{10} size?

Answer:
I guess that sub-micron particles can be assumed to follow the flow but I cannot remember at what size this approximation becomes invalid. Our concern was primarily with gaseous pollutants.

Comment from: B Borresen
The understanding of air flow movement and not least concentration movement is important. We use CFD and at times wind tunnels. Why open the model? Why use inlets/outlets? Your experiments: would, it not be relevant actually just to measure concentrations at different air inlet points? That is as long as you are just relating to concentrations and not air exchange rates etc. Of course, inlet, outlet and inside concentrations should be <u>the same</u>.

Reply:
We agree that it should normally be sufficient to measure external concentrations. We used an open model for two reasons:-

(i) To determine the ventilation rate. This has an important bearing on the time taken to reach equilibrium. With low ventilation rates the external concentration may not be representative of the internal concentrations which are experienced in practice.

(ii) For future use in the measurement and time-varying internal concentrations arising from unsteady emission sources.

We compared the external and internal concentrations with the expectation that they would agree. That they did not always do so was a surprise to us and some possible explanations are given in the paper.

Question from: V Kukadia
You said the traffic was simulated by a line source. How was the line source generated?

Answer:
The source was not strictly a line source in that it consisted of a tube with small holes drilled in it at discrete intervals. It was therefore not a continuous source. The aim was to simulate the emissions from lines of stationary traffic. The flow rate was chosen to reproduce the non-dimensional emission characteristics used for the 1:100 scale model described in Ref.11.

Question from: Unknown
What about the cleaning properties of the cavity?

Answer:
It is obvious that active envelopes require more maintenance but in this case the inner panes can be opened for cleaning purposes. More important is the airtightness of the envelope. An openable pane is more sensitive for air leakage than a fixed one. In this case study we demonstrated the bad sealing of the inner pane with an IR-thermography and with the smoke stick experiment. A lack of airtightness in the inner or outer pane causes energy losses and can cause a disturbance of the airflow and therefore airtightness is a very important performance check.

Question from: Y Li
It appears that the airflow rate through the cavity is kept as a constant in your tests. Could you please comment on the interaction of forced convection and natural convection and its effect on the envelope performance in particular when used as solar collector.

Answer:
The model assumes a constant airflow rate through each cavity and a constant velocity over the cross-section. The airflow rates in the cavity are supposed to be proportional to the third power of the cavity width.

It should be clear that the downward forced airflow could be disturbed by the upward thermal stack effect. The effect of the bouyancy depends on the thermal conditions and the downward airflow. In case of high temperatures (on sunny days, the temperature of the pane can rise up to 55°C) and a moderate downward airflow, an upward flow will occur along the heated pane (and the shading device) disturbing the downward forced flow and affecting the heat-flux. It seems more appropriate to use an upward airflow if the active envelope is used as a solar collector. Further research and measurements will be performed on this topic.

[redacted]

Question from: T H Dokka
TVOC is a rather poor measure of the emission from building materials (statement). It is also difficult to measure. How do you measure emissions, and how do you calculate the concentration?

Answer:
The standard does not describe how to measure emission rate but there are some measurement methods shown in literature which could be useable in the near future.

Measurement of emission rates from different kinds of building materials is now possible. With this kind of data available, we can calculate the concentration of VOC in indoor air.

Question from: G Richardson
The Japan standard of $0.15mg/m^3$ for particulates. How is the figure arrived at $x\%\ PM_{10}$; $y\%\ PM_5$ etc?

Answer:
The standard of $0.15mg/m^3$ for particulates is prescribed as a daily mean value. Actually, measurement is taken three times a day and these values are averaged. Therefore, it doesn't take into account the value of $x\%\ PM_{10}$, for example.

Section 2: Additional Presentations not included in Main Proceedings

VENTILATION TECHNOLOGIES IN URBAN AREAS

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Ventilation Reliability – an Evaluation Tool for Domestic Ventilation

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Synopsis

Pre-assessing the reliability of ventilation systems is a difficult task and no simple methods have existed. This paper presents a tool for estimating the reliability of domestic ventilation systems. In general, ventilation reliability means the probability that the chosen ventilation system performs in an acceptable way for a certain building, in a certain climate, between scheduled maintenance measures. The tool presented consists of two different sub-tools: one for reliability as indicated by air flow rate stability as a function of situational factors and another for reliability as indicated by performance over time i. e. systems and components reliability.

Four different ventilation strategies have been analysed; natural window airing, passive stack ventilation, mechanical exhaust ventilation and mechanical exhaust and supply ventilation. The use of the tool involves specifications of the dwelling, the occupancy, the ventilation system etc. and the tool presents the consequences of the choices made, in a qualitative way on a five-grade scale from excellent to very bad reliability.

Introduction

Basic principles of a method for estimating the reliability of mechanical ventilation systems have been outlined (Kronvall, 1996) and further developed (Kronvall & Ruud, 1997). The concept of these analyses has been based on general methods for system safety analysis. The reliability analyses have now also been extended to other ventilation systems than mechanical ones, and another anlalysis component – flow rate stability has been added.

This final tool estimates the ventilation reliability for a dwelling. In general, the ventilation reliability means the probability that the chosen ventilation system performs in an acceptable way for a certain building, in a certain climate, between scheduled maintenance measures. Of real concern is the reliability of the indoor air quality. For practical reasons the tool presented consists of two different tools:

1. for reliability as indicated by air flow rate stability as a function of situational factors
2. for reliability as indicated by performance over time i. e. systems and components reliability.

Furthermore, the reliability as indicated by perceived indoor air quality can be analysed using the IAQ tool prepared in another part of the annex work. (Månsson ed., 1998).

The tool for airflow rate stability is based on the assumption that the ventilation flow rate in the bedrooms should exceed 4 l/s per person.

The tool for systems and components reliability is based on certain assumptions concerning mean life times, standard deviation of mean life times and maintenance intervals. If other assumptions are of interest, then an advanced computer based tool is available. In this advanced tool, the assumptions, mentioned above could be changed, see Main background report.

The paper forms part of the Swedish contribution to the work of IEA-Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems". The final report from the whole annex work (Månsson,ed., 1998) will be available by the end of 1998.

Input Parameters

Building parameters

Three climates are taken into account: Nice, London, and Ottawa corresponding to mild, moderate and cold situations.

Three types of dwellings are considered:
- A four main rooms (three bedrooms) apartment located on ground floor in a four-storey building (called D4a ground).
- A four main rooms apartment located on the top floor in a four-storey building (called D4a top).
- A four main rooms detached single family house (called D4c)

Two different leakage values. n_{50} can have the following values:
- 1 or 5 air changes per hour at 50 Pa, for the apartments D4a
- 2.5 or 10 air changes per hour at 50 Pa, for the house D4c

Ventilation system description

The ventilation systems are identified by four basic systems. (Natural window airing **NWA**, passive stack ventilation **PSV**, mechanical exhaust only **MEO**, mechanical exhaust and supply **MSE**). Those systems can then be combined with local fans in bathroom/toilet and/or kitchen and window opening patterns (closed, or climate depending).

WINDOWS OPENING (AIRING)

Bedroom windows can be opened during weekdays from 8 to 12 o'clock, depending on the weather conditions.

Supply air

NWA 2 cases of purpose provided openings:

- **no :** **0 cm^2**
- **yes :** **410 cm^2** (80 cm^2 in each habitable room. 30 cm^2 in each of the toilet, bath, kitchen)

PSV 2 cases of purpose provided openings:

- **no :** **100 cm^2**
- **yes :** **400 cm^2** (80 cm^2 in each bedroom and 160 cm^2 in the living-room)

MEO 2 cases of purpose provided openings:

- **no :** **0 cm^2**
- **yes :** **100 cm^2** (20 cm^2 in each bedroom and 40 cm^2 in the living-room)

MSE 3 cases of supply flow rates:

- **15 l/s** (3 l/s in each bedroom and 6 l/s in the living-room)
- **30 l/s** (6 l/s in each bedroom and 12 l/s in the living-room)
- **45 l/s** (9 l/s in each bedroom and 18 l/s in the living-room)

EXHAUST FLOW RATES AND STACKS

NWA No vertical duct

PSV A passive stack ducted ventilation system is installed in the kitchen, bathroom and toilet.

MEO and MSE The mechanical exhaust flow rate is given for three levels.

- 15 l/s (7.5 l/s in the kitchen 5 l/s in the bath and 2.5 l/s in the toilet)
- 30 l/s (15 l/s in the kitchen 10 l/s in the bath and 5 l/s in the toilet)
- 45 l/s (22.5 l/s in the kitchen 15 l/s in the bath and 7.5 l/s in the toilet)

Heat exchanger efficiency for MSE is 50 %

LOCAL ADDITIONAL EXHAUST FANS

No no local fan

Yes 2 local fans as follows:

- Kitchen hood: Running time 1 h/day, at 17.00 - 18.00 o'clock. Flow rate 100 l/s
- Bathroom fan: Running time 2 h/day. **Weekdays** 6.00 - 8.00 o'clock and weekends at 9.00 - 11.00 Flow rate 25 l/s.

TECHNICAL QUALITY OF VENTILATION SYSTEMS

Poor system: Low cost equipment is chosen in order to minimise the initial cost. Low attention to future maintenance and life cycle cost.

Average system: Relatively good quality standard equipment chosen according to good engineering practice. Some attention on minimising future maintenance and life cycle cost, but still rather high attention on minimising the investment cost.

Best Practice: The best available high quality equipment is carefully chosen. High attention on minimising future maintenance and life cycle cost. Less attention on reducing the investment cost.

Note: All the components of each system are assumed to be equally poor or good, i.e. there are no single especially weak (or strong) component. (To see the influence of an unevenly designed system we refer to the use of the advanced tool, which has been used when developing the simplified tool. See "Main background report".)

Maintenance levels

High: Maintenance is performed approximately 50-100 % more intense (often) than normal practice for the actual type of system.

Medium: Maintenance is performed with an intensity according to normal practice for the actual type of system.

Low: Maintenance is performed approximately 30-50 % less intense (often) than normal practice for the actual type of system.

Note: For each maintenance level, the maintenance interval of each component is assumed to be equally intense, i.e. there is no single maintenance interval for any component that is especially short or long. (To see the influence of an uneven maintenance scheme we refer to the use of the advanced tool. See "Main background report".) It should also be noted that if the chosen maintenance level does not correspond to the suggested level in the output of the LCC-tool, then the calculated LCC does not apply for the chosen level.

Other situations

The tool was produced based on a four room dwelling. For other cases, it is possible to use the tool if it is considered that one bedroom is added or deleted (nevertheless, a one main room dwelling should not be taken into account). The results are then less precise, but the ranking would remain quite the same. If these changes must be made, the following adjustments could be applied:

- n_{50} : no change, as n_{50} is related to the dwelling volume
- Airing : no change
- Supply air devices : the bedroom(s) added or deleted are equipped as defined in 2.2.2
- Mechanical airflow: add or subtract 3, 6 or 9 l/s for each bedroom. Split the total exhaust airflow in kitchen, bathroom and toilet using weights of 3:2:1.
- Additional fan : no change

Output Data

The output from the tool is given as qualitative ratings with the following interpretations:

+ +	**Excellent reliability**
+	**Good reliability**
0	**Fair reliability**
-	**Poor reliability**
- -	**Very poor reliability**

Evaluation Tool

In Tables 1, 2 and 3 are given the reliability as indicated by situational factors (flow rate stability)

Table 1 Reliability flow rate stability. Climate ***Nice***										
System	**Airing**	**Inlet**	**Exhaust**	**Extra**	**Dwelling**					
		area	**fan flow**	**fan**	**Apartment**				**House**	
			rate		**D4a top**		**D4a ground**		**D4c**	
					n50		**n50**		**n50**	
					1	**5**	**1**	**5**	**2.5**	**10**
NWA	No	0		N	- -	-	- -	-	- -	- -
				Y	- -	-	- -	-	- -	- -
		410		N	**0**	**0**	**0**	**0**	-	**0**
				Y	**0**	+	**0**	**0**	-	**0**
	Yes	0		N	-	**0**	-	**0**	-	-
				Y	-	**0**	-	**0**	- -	-
		410		N	+	+	**0**	**0**	**0**	**0**
				Y	**0**	+	**0**	+	**0**	**0**
PSV	No	100		N	+	+	**0**	+	**0**	**0**
				Y	+	+	**0**	+	**0**	**0**
		400		N	+	+	+	+	+	+
				Y	+	+	+	+	+	+
MEO		0	15	N	- -	**0**	- -	**0**	- -	**0**
				Y	- -	**0**	- -	**0**	- -	**0**
			45	N	**0**	+	**0**	+	-	**0**
				Y	**0**	+	**0**	+	-	**0**
		100	15	N	-	**0**	**0**	**0**	**0**	**0**
				Y	**0**	+	**0**	**0**	**0**	**0**
			45	N	+	+	**0**	+	**0**	**0**
				Y	+	+	+	+	**0**	**0**
MSE			15	N	**0**	+	**0**	+	**0**	+
				Y	**0**	+	**0**	+	**0**	+
			30	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++
			45	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++

Table 2 Reliability flow rate stability. Climate **London**

System	Airing	Inlet area	Exhaust fan flow rate	Extra fan	Dwelling					
					Apartment				House	
					D4a top		D4a ground		D4c	
					n50		n50		n50	
					1	5	1	5	2.5	10
NWA	No	0		N	-	-	-	-	- -	-
				Y	- -	-	- -	-	- -	-
		410		N	0	+	0	0	0	0
				Y	0	+	0	0	0	0
	Yes	0		N	0	0	0	0	-	0
				Y	-	0	-	0	-	0
		410		N	+	+	0	+	0	0
				Y	+	+	0	+	0	+
PSV	No	100		N	+	+	0	0	-	0
				Y	+	+	0	0	-	0
		400		N	+	+	+	+	0	0
				Y	+	+	+	+	0	0
MEO		0	15	N	- -	0	- -	0	- -	0
				Y	- -	0	- -	0	-	0
			45	N	0	+	0	+	-	0
				Y	0	+	0	+	-	0
		100	15	N	0	+	0	0	0	0
				Y	0	+	0	0	0	0
			45	N	+	+	+	+	0	0
				Y	+	+	+	+	0	0
MSE			15	N	0	+	0	+	0	+
				Y	0	+	0	+	0	+
			30	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++
			45	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++

Table 3 Reliability flow rate stability. Climate ***Ottawa***

System	Airing	Inlet area	Exhaust fan flow rate	Extra fan	Dwelling					
					Apartment				House	
					D4a top		D4a ground		D4c	
					n50		n50		n50	
					1	5	1	5	2.5	10
NWA	No	0		N	-	0	-	0	-	0
				Y	-	0	-	0	-	0
		410		N	0	+	0	+	0	0
				Y	0	+	0	+	0	0
	Yes	0		N	0	0	0	0	0	0
				Y	0	0	0	0	0	0
		410		N	+	+	+	+	0	+
				Y	+	+	+	+	0	+
PSV	No	100		N	0	0	0	0	-	0
				Y	0	0	0	0	-	0
		400		N	+	+	0	0	0	0
				Y	+	+	0	0	0	0
MEO		0	15	N	- -	0	- -	0	-	0
				Y	- -	+	-	0	-	0
			45	N	0	+	0	+	-	0
				Y	+	+	0	+	0	0
		100	15	N	0	+	0	0	0	0
				Y	0	+	0	+	0	0
			45	N	+	+	+	+	0	0
				Y	+	++	+	+	0	0
MSE			15	N	0	+	0	+	0	+
				Y	0	+	0	+	0	+
			30	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++
			45	N	++	++	++	++	++	++
				Y	++	++	++	++	++	++

Table 4 Reliability as indicated by performance over time is shown in the set of tables on this page

Apartments

Passive stack ventilation system			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	++	++	-
Average system	++	++	+
Best practice	++	++	++

Central exhaust ventilation			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	++	-	--
Average system	++	++	-
Best practice	++	++	+

Central supply and exhaust ventilation			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	+	--	--
Average system	++	+	--
Best practice	++	++	0

For natural window airing ventilation strategy, only openable windows, and sometimes natural supply air devices in the facades, constitute the ventilation system. For this
case, the score "++" could be used.

Single family houses

Passive stack ventilation system			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	++	+	-
Average system	++	++	+
Best practice	++	++	+

Central exhaust ventilation			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	++	-	--
Average system	++	++	-
Best practice	++	++	+

Central supply and exhaust ventilation			
Technical quality of system	Maintenance level		
	High	Medium	Low
Poor system	+	--	--
Average system	++	0	-
Best practice	++	++	-

For natural window airing ventilation strategy, only openable windows, and sometimes natural supply air devices in the facades, constitute the ventilation system. For this
case, the score "++" could be used.

Further Information

Reliability as indicated by situational factors

Computer simulations of outdoor air flow rates to individual rooms have been performed by means of a multi-zone air flow model.

The reliability as indicated by situational factors study is focused on the fraction of time of the total heating season when the **ventilation airflow rate is at or above a certain ventilation requirement.** The target value chosen for outdoor air ventilation flow rate is 4 litres per second per person in the bedrooms. This airflow rate chosen represents an internationally commonly used figure.

In order to establish quantitative relationships between building characteristics, ventilation strategies, climates etc. and the responding parameter, which is the air flow rates to the bedrooms, a procedure using fractional factorial design (174 combinations) and a statistical analysis was used.

In the simplified tool a qualitative value is given depending on the fraction of time when the air flow rates at or above the target value. This is 4 litres per second and person in the bedrooms, as an average for the bedrooms in the house/dwelling.

In order to assess the performance of different ventilation strategies etc., from a flow rate stability point of view, it is necessary to set up intervals for different classes, see table 5.

Table 5 Transformation from quantitative to qualitative values
Flow rate stability

Fraction of time with acceptable flow rates	**0.50-1.00**	**0.25-0.50**	**0.12-0.25**	**0.06-0.12**	**0.00-0.06**
Assessment	**++**	**+**	**0**	**-**	**--**
Reliability Quality	**Excellent**	**Good**	**Fair**	**Poor**	**Very poor**

Reliability as indicated by performance over time

Mechanical ventilation systems are built up by a number of mechanical and electrical components, such as fan(s), electrical motor(s), damper(s), silencer(s), air terminal devices, system(s) for automatic control etc. The way that these components influence the performance of the system can of course be described in a fault-tree analysis. There are principally three different kinds of probabilities to estimate individual events.

Fixed probabilities These probabilities are, in principle, not depending on time. Power failure for example can be estimated if you can acquire data on how many hours per year you can expect power failure from the Electricity Company.

Time-depending probabilities These are depending on in which state, i.e. at what time you analyse the problem. The failure intensity for mechanical components, for example, is not the same as long as the component is fairly new compared to when it grows older. Another example is duct fouling with its consequences of gradually lower flow rates.

More or less unknown probabilities In the context of ventilation performance, typical examples are events based on user influence. These probabilities are very little known, not only because people are different, but also that the design of the ventilation system influences the behaviour.

By connecting component models in a fault-tree scheme, the analysis can be extended to a system level. An example of a fault-tree for a mechanical exhaust ventilation system is given in figure 1.

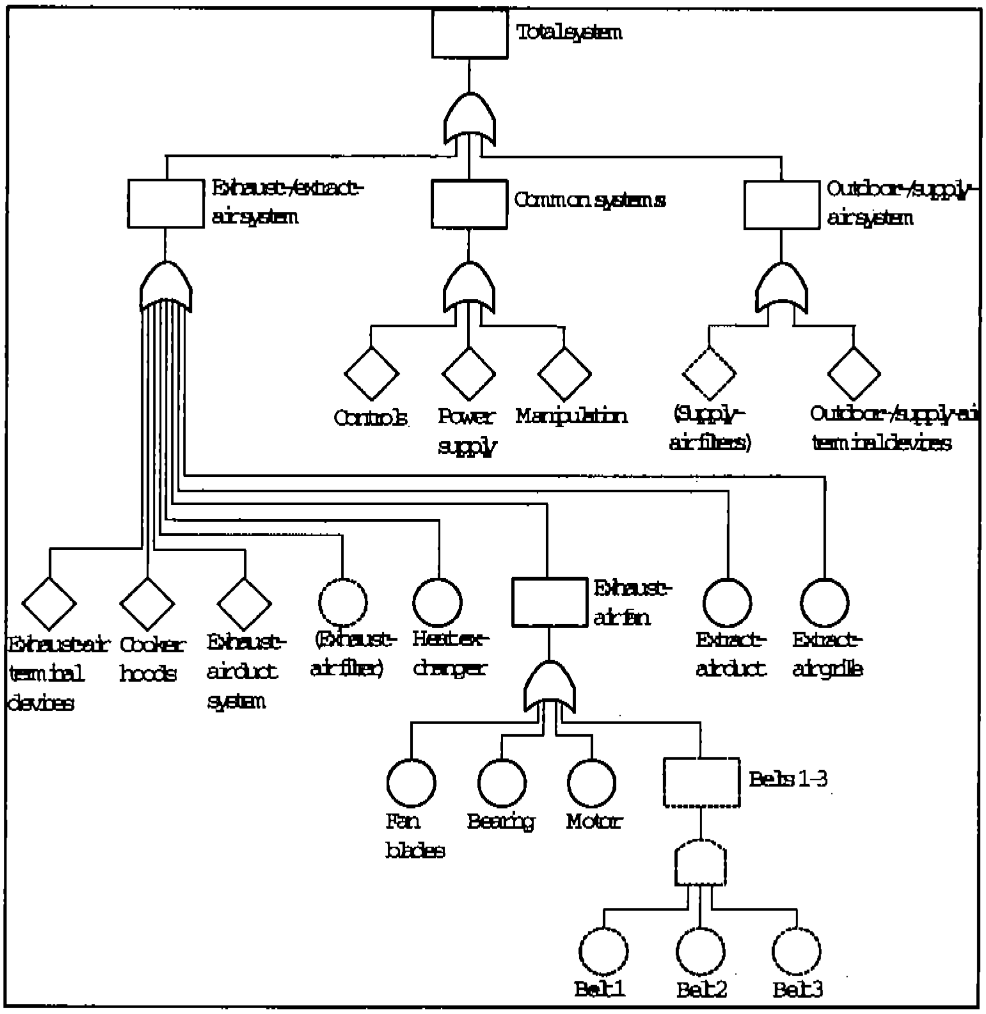

Figure 1 Fault tree for a central mechanical exhaust ventilation system

A simplified model for describing the influence of maintenance has also been incorporated into each component model. It assumes that after each maintenance occasion, the component is "as good as new".
For each type of system, three different quality standards have been defined and combined with three different levels of maintenance level. This means a matrix with a total of nine different combinations for each ventilation system. Relevant data have been collected from published and orally transferred empirical experiences from maintenance people and other researchers working with reliability or related matters.

The result of each combination in the matrixes with life time and maintenance intervals can be presented in figures showing the estimated reliability for the system as a function of time. The result is further evaluated by calculating the mean and minimum value of the reliability for a time span of 30 years. An example is shown in figure 2. The result for each system applied in both single-family houses and multi family buildings can be summarised in a matrix.

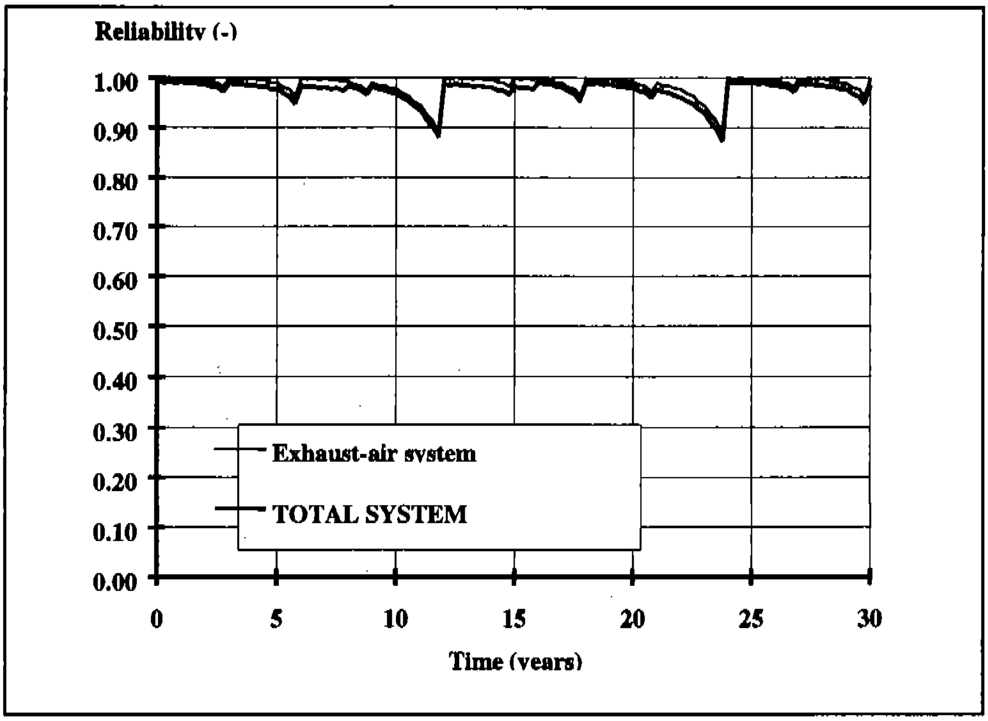

Figure 2 ***The variation of reliability over a 30 year period for a multi-family building with exhaust ventilation at an average system performance with medium maintenance***

The resulting mean and minimum values (of the reliability for a time span of thirty years) given in the matrixes is then finally evaluated and transferred into a single classification system, the qualitative tool. The interpretation from mean and minimum values to qualitative values are given in table 6.

Table 6 Transformation from quantitative to qualitative values Performance over time

	Minimum reliability									
Mean reliability	**0.90-1.00**	**0.80-0.89**	**0.70-0.79**	**0.60-0.69**	**0.50-0.59**	**0.40-0.49**	**0.30-0.39**	**0.20-0.29**	**0.10-0.19**	**0.00-0.09**
0.90-1.00	++	++	+	+	0	0	-	-	- -	- -
0.80-0.89		+	+	0	0	-	-	- -	- -	- -
0.70-0.79			0	0	-	-	- -	- -	- -	- -
0.60-0.69				-	-	- -	- -	- -	- -	- -
0.50-0.59					- -	- -	- -	- -	- -	- -
0.40-0.49						- -	- -	- -	- -	- -
0.30-0.39							- -	- -	- -	- -
0.20-0.29								- -	- -	- -
0.10-0.19									- -	- -

0.00-0.09										- -

The use of the advanced tool for systems and components factors involves the use of a computerised tool (spreadsheet) for MS-Excel, Version 5.0 or 7.0. The tool is available electronically (by e-mail with attached *.XLS files, or on diskette) from:

Mr. Svein Ruud, Swedish National Testing and Research Institute,
Mail: Box 857, SE-501 15 BORAAS, Sweden,
E-mail: svein.ruud@sp.se

Or

Dr. Johnny Kronvall, J&W BAS,
Mail: Slagthuset, SE-211 20 MALMOE, Sweden,
E-mail: johnny.kronvall@malmo.jacwid.se

In this advanced tool, it is possible to change the figures and use your own figures.

Conclusions

From the work performed, the following conclusions can be drawn:

Regarding flow rate stability

- The flow rate stability is very sensitive to the ventilation system chosen.
- Generally, the flow rate stability is better for mechanical systems than natural and passive ones
- It is easier to maintain stable flow rates in colder climates than in mild ones

Regarding performance over time

- The general concept of system safety analyses can be used in order to assess the performance over time for mechanical ventilation systems.
- There is a great lack of knowledge on basic reliability data for ventilation system components, e.g. life-times; only more or less "best guesses" can be used with the present knowledge.
- The reliability of a mechanical ventilation system is determined both by the technical quality of the system and the maintenance intensity.

Acknowledgements

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VENTILATION AND COOLING

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The principles of a homogeneous tracer pulse technique for measurement of ventilation and air distribution in buildings.

by

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THE PRINCIPLES OF A HOMOGENEOUS TRACER PULSE TECHNIQUE FOR MEASUREMENT OF VENTILATION AND AIR DISTRIBUTION IN BUILDINGS.

SYNOPSIS

The principles of a new tracer gas technique is described in the paper. The new technique involves pulse injection of tracer gas and has the same advantages as the previously known homogeneous emission technique. It can for example advantageously be used in large buildings and buildings with many rooms and yields information on the distribution of ventilation air within the building. However, contrary to the homogeneous emission technique, yielding the average ventilation performance during an extended time, the new technique allows measurement during short term periods.

The new technique is based on homogeneous pulse injection, which means that pulses of tracer gas are injected in each zone in a zone-divided building, with amounts which are proportional to the zone volumes and integrating sampling of tracer gas concentration. Theoretical and practical aspects of the technique are described.

1. INTRODUCTION

1.1 Existing tracer gas techniques

There are essential four different main principles for tracer gas injection for measuring ventilation characteristics in buildings.

- Step change techniques
- Pulse techniques
- Constant emission techniques
- Constant concentration techniques

These four principles all have their advantages and disadvantages for field measurement of ventilation. They will shortly be discussed below, especially with respect to their usefulness for measuring the local mean age of air in real buildings. Note that here exist several variants of the techniques, which however are outside the scope of this paper to discuss. A comprehensive treatise of tracer gas techniques can be found in a book by Etheridge and Sandberg (1).

1.1.1 Step change techniques

Step change techniques involve the step-down technique (usually called the decay technique) and step-up technique.

1.1.1.1 Step-down technique

When using the *step-down technique* for measuring local mean ages of air, tracer gas is mixed into the whole ventilated system* to a uniform initial concentration (C_0), after which no more tracer gas is supplied and the decay of the local concentration is integrated until no tracer gas is left in the system.

* note. The concept "ventilated system" involves all building spaces, that are somehow connected in a ventilation sense, i. e. all spaces to which air can be transferred from other parts of the system. The only air which can be supplied over the boundary of a ventilated system is ambient air.

The local mean age of air is obtained from the concentration integral divided with the initial concentration.

$$\bar{\tau}_p = \frac{\int_0^\infty C dt}{C_0} \qquad (1)$$

1.1.1.2 Step-up technique

In the step-up technique tracer gas is injected into the supply air at a constant concentration at time t=0 and the local concentration history at a point p is recorded as a function of time. To obtain the local mean age of air, the concentration difference between the equilibrium concentration C_∞ and the concentration at time t is integrated from 0 until equilibrium is attained.

$$\bar{\tau}_p = \frac{\int_0^\infty (C_\infty - C) dt}{C_\infty} \qquad (2)$$

1.1.2 Pulse techniques

In the pulse technique a pulse of tracer gas is injected into the supply air and the local time response of concentration recorded at point p. The local mean age of air is computed from the first moment of the concentration divided with the zeroth moment.

$$\bar{\tau}_p = \frac{\int_0^\infty t \cdot C dt}{\int_0^\infty C dt} \qquad (3)$$

1.1.3 Constant emission techniques

These technique rely on a continuous release of tracer gas at a constant rate in the ventilated system. If the emission rate is so distributed in the system that, in each part (zone) of the system, it is proportional to the zone volume, the emission is said to be homogeneous. If the tracer emission is made homogeneously, the local mean age of air can be calculated from the quotient between the local steady state concentration C_p and the emission rate per volume unit $(\dot{m} / V)$ (e.g. 1, 2, 3).

$$\bar{\tau}_p = \frac{C_p}{\left(\frac{\dot{m}}{V}\right)} \qquad (4)$$

1.1.4 Constant concentration technique

In this technique the injection rate of tracer gas in the different zones of the system is adjusted so that the resulting steady state concentrations are equal in every zone. This technique requires a real time analysis of tracer gas and an injection control device with feedback from the analysis equipment.

The constant concentration technique has not been considered for evaluating local mean ages of air. It is used in order to determine the direct inflow (q_p) of ventilation air into the zones.

$$q_p = \frac{\dot{m}_p}{C} \tag{5}$$

where $\dot{m}_p$ is the injection rate in zone p and C is the target concentration.

1.2 Field measurement requirements

In order to be useful for routine field measurement of ventilation, the used technique should be simple, quick and inexpensive. Expensive equipment and expertise should not be tied up during a long measurement. The necessary initial conditions should be possible to establish.

Tracer gas methods are the only ones, which can be used to measure how the air is actually distributed in the building space. The air distribution patterns are usually described using the concept of "local mean age of air" or its inverted value "local ACH". Other possible alternatives are "local purging flow rate" and "ventilation effectiveness".

Methods which allow air sampling in bags or in adsorption tubes and subsequent analysis in a laboratory are generally preferred over on-site real time analysis. Diffusive sampling in adsorption tubes is commonly used with the passive tracer gas technique (e. g. homogeneous emission technique).

The decay technique with bag sampling at intervals is sometimes used. The decay technique is however critically depending on the necessary uniform initial concentration of tracer gas, which can be very difficult to achieve in large buildings and buildings with many rooms.

Pulse techniques are seldom utilised for mapping the distribution of local mean ages of air, because it requires the first moment of tracer concentration to be calculated in several positions. This requires on site monitoring of concentration versus time, during the whole pulse response.

In this paper a new pulse technique (the homogeneous pulse technique) is presented, which is useful for routine field measurement of ventilation. It relies on a homogeneous pulse injection pattern and allows integrating sampling, for example with passive diffusive samplers.

2. THEORETICAL CONSIDERATIONS OF THE HOMOGENEOUS PULSE TECHNIQUE

2.1 The analogy between steady state concentration and integrated pulse response.

The principle of the homogeneous pulse technique can most easily be explained through an analogy with the constant emission technique in multi-zone systems.

The reason is the well-known fact that the steady state concentration C_∞ of a tracer with constant emission rate $\dot{m}$ has a direct relationship to the integral from zero to infinity of the concentration response (dosage) from a short pulse with a released amount of *m*, if the air flow patterns are stable within the ventilated system (ref. 4, 5).

Thus, if the time response at a location **i** from a short pulse with a released amount of $\mathbf{m_j}$ injected at a location **j**, is represented by

$$C_{ij}(t) = m_j \cdot f_{ij}(t) \tag{1}$$

where f_{ij} is a characteristic response function, a continuous release of tracer at location **j** would yield a steady state concentration at location **i** of

$$C_{ij}^{\infty} = \dot{m}_j \cdot \int_0^{\infty} f_{ij}(t)dt \tag{2}$$

We can therefore conclude that the integral of a short pulse response

$$I_{ij} = \int_0^{\infty} C_{ij}(t)dt = \frac{m_j}{\dot{m}_j} C_{ij}^{\infty} \tag{3}$$

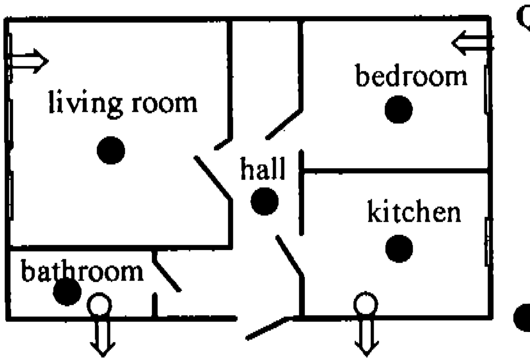
where C(t) is the concentration response from the pulse with amount m and C_{∞} is the corresponding steady state concentration of a tracer with constant emission rate of $\dot{m}$. In this discussion we will use this analogy throughout.

2.2 Determination of the flow matrix in multi-zone systems.

The flow matrix **Q** in a system consisting of **n** zones, each of which is fully mixed can be determined by injection of tracer gas with a constant rate in one zone at a time (or more general in **n** different linearly independent patterns) and measure the equilibrium concentrations in each zone for each pattern. The equilibrium concentrations are given by the matrix equation system:

$$\mathbf{C}^{\infty} = \mathbf{Q}^{-1}\dot{\mathbf{m}} \tag{4}$$

where $\mathbf{C}^{\infty}$ is the quadratic matrix of steady state concentrations C_{ij}^{∞} in zone **i** resulting from the **j**th release pattern and $\dot{\mathbf{m}}$ is the quadratic release pattern matrix. This equation system has been used in ref. (6) to determine the flow matrix in a 5 room indoor experimental house displayed in figure 1, where the tracer gas injection was made in one room after the other. Mixing fans were used in each room.



Q-matrix obtained from constant emission [m^3/h]

Livingroom	287	-193	-70	3	4
Hall	-297	479	-8	-91	-45
Bedroom	-15	-70	118	-5	-1
Kitchen	5	-108	-22	138	-14
Bathroom	6	-74	-15	1	80

Figure 1. *The indoor test house and the* **Q**-*matrix obtained from constant emission tracer gas experiments.*

Using the analogy between steady state concentrations and integrated pulse responses, a similar equation corresponding to eq. (4) can be set up for pulse injection:

$$\mathbf{I} = \mathbf{Q}^{-1}\mathbf{m} \tag{5}$$

where **I** is the quadratic matrix of integrated pulse responses I_{ij} in zone **i** resulting from the **j**th pulse release pattern and **m** is the quadratic pulse release pattern matrix.

These matrices can easily be transformed to the τ-matrix through multiplication from the right by $\mathbf{m}^{-1}\mathbf{V}$, where **V** is the diagonal zone volume matrix.

$$\mathbf{C}^{\infty}\dot{\mathbf{m}}^{-1}\mathbf{V} = \mathbf{Q}^{-1}\mathbf{V} = \boldsymbol{\tau} \tag{6}$$

$$\mathbf{I}\mathbf{m}^{-1}\mathbf{V} = \mathbf{Q}^{-1}\mathbf{V} = \boldsymbol{\tau} \tag{7}$$

This means that the τ-matrix can be determined either from the steady state concentrations and the emission rates per unit volume, or from the integrated pulse responses and the injected amounts per volume unit.

In order to illustrate this, the expected pulse responses with the flow matrix obtained in ref. (6) and displayed in figure 1 is computed from the numerical solution of the differential equation system:

$$\mathbf{V}\frac{d\mathbf{C}}{dt} + \mathbf{QC} = \mathbf{0} \text{ with } \mathbf{C}(0) = \mathbf{1} \tag{8}$$

which describes the concentration versus time in the different zones when a unit pulse amount per volume unit is injected in one zone at a time and the mixing within each zone is instantaneous. The result is displayed in figure 2.

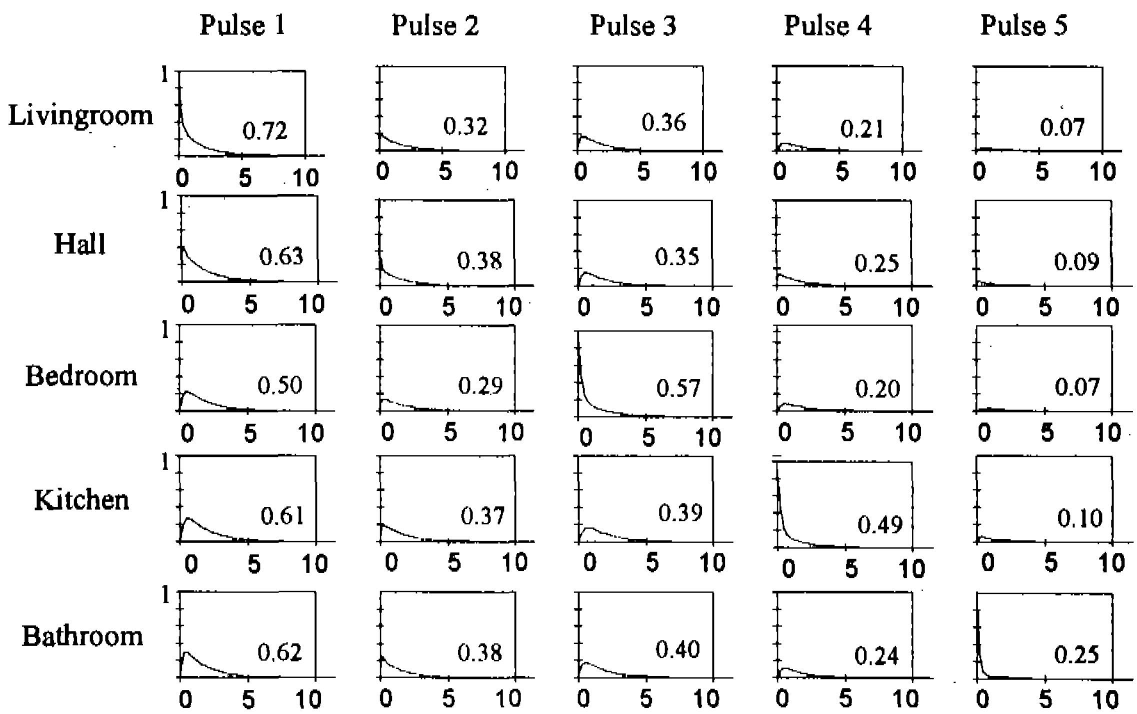


Figure 2. *Diagrams displaying the simulated concentration responses in the different rooms as a function of time (hours) when pulses of tracer gas are injected in the room marked with an asterisk with a unit amount per room volume. The values enclosed in the diagrams represent the computed pulse response integrals and hence also the entries in the τ-matrix.*

2.3 Determination of the local mean age of air in multi-zone systems.

The τ-matrix obtained from the numerical integration of the pulse responses displayed in figure 2 is shown below.

$$\tau = \begin{pmatrix} 0.72 & 0.32 & 0.36 & 0.21 & 0.07 \\ 0.63 & 0.38 & 0.35 & 0.25 & 0.09 \\ 0.50 & 0.29 & 0.57 & 0.20 & 0.07 \\ 0.61 & 0.37 & 0.39 & 0.49 & 0.10 \\ 0.62 & 0.38 & 0.40 & 0.24 & 0.25 \end{pmatrix} \tag{9}$$

The local mean ages of air in a multi-zone system can be obtained from the row sums of the τ-matrix:

$$\bar{\tau} = \tau \mathbf{1} \tag{10}$$

where **1** denotes the unit vector

$$\mathbf{1} = \begin{pmatrix} 1 \\ 1 \\ 1 \\ 1 \\ 1 \end{pmatrix}; \qquad \bar{\tau} = \begin{pmatrix} 1.68 \\ 1.70 \\ 1.62 \\ 1.97 \\ 1.89 \end{pmatrix} \tag{11}$$

This result is in close agreement with that obtained in the original paper (6).

2.4 Determination of the local mean age of air without knowledge of flow matrix.

In order to determine the flow matrix and the full τ-matrix as many independent experiments as there are zones must be performed. However, to determine the local mean age $\bar{\tau}$ -vector, only a single experiment is necessary. From eq. 7 and eq. 10 the $\bar{\tau}$ -vector can be obtained directly:

$$\bar{\tau} = \mathbf{Im}^{-1}\mathbf{V}\mathbf{1} \tag{12}$$

If pulses are injected in the different zones with amounts which are proportional to the zone-volumes $\mathbf{m} = k\mathbf{V}$ then eq. 12 reduces to:

$$\bar{\tau} = \frac{1}{k}\mathbf{I}\mathbf{1} \tag{13}$$

which means that the local mean age in a zone is equal to the sum of all individual integrated pulse responses in that zone divided with the injected amount per volume unit.

$$\bar{\tau}_i = \frac{1}{k}\sum_j \int_0^\infty C_{ij}(t)dt \tag{14}$$

Because there is a finite number of pulses and the individual pulse responses are independent of each other, the tracer concentration is the sum of all pulse responses

$$\bar{\tau}_i = \frac{1}{k}\sum_j \int_0^\infty C_{ij}(t)dt = \frac{1}{k}\int_0^\infty \sum_j C_{ij}(t)dt = \frac{1}{k}\int_0^\infty C_i(t)dt \tag{15}$$

This holds true even if the pulses in the different zones are not injected simultaneously, but at different times as long as the flow patterns in the system is not changing. Eq. (15) tells that the local mean ages of air in a zone-divided system can be determined with integrating sampling, if pulses of tracer gas are injected into the different zones with amounts which are proportional to the zone volumes and the sampling is made from the moment of the first injection and continued until all tracer gas is ventilated out of the system.

In figure 3 the concentration versus time in the different zones are calculated from the individual pulse responses displayed in figure 2 under the assumption of successive injections into the different zone with 0.1 hours intervals. Instantaneous mixing within a zone is assumed. The calculated integrals closely agree with the local mean ages of air.

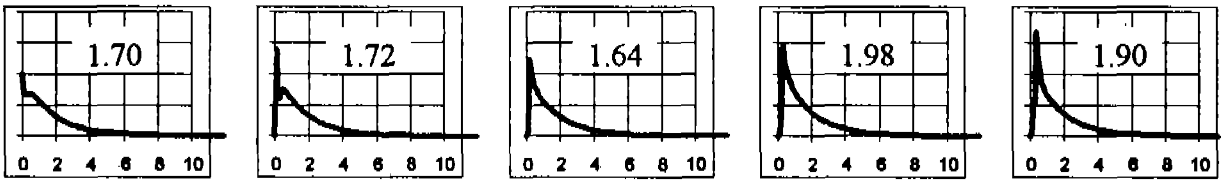


Figure 3. *Tracer concentration as a function of time in the different rooms, when pulses are injected in one zone after the other with 5 minutes interval, with amounts proportional to the zone volumes. The values enclosed in the diagrams represent the computed pulse response integrals and hence also the local mean ages of air.*

3. PRACTICAL ASPECTS

3.1 Measurement of the integrated pulse response.

At every location where the local mean age of air is to be measured, the local concentration must be integrated from the moment that tracer gas injection is started in the system, until all injections have been performed and all tracer gas has been removed by the ventilation air. Such integration can be performed either using real time analysing instrument, recording the time history and making the integration afterwards, or using integrating sampling and making the analysis afterwards in the laboratory. In most cases, integrating sampling is more convenient and less expensive. For determination of the mean ages of air, the details of the time history is namely not interesting, only the integrated value is relevant.

Integrating sampling can be performed actively or passively. For active sampling a pump is used, which draws air either continuously or discontinuously through an adsorbent bed which adsorbs the tracer gas or alternatively collects the sampled air in a sample bag for later analysis. Passive sampling relies on diffusion of tracer gas towards an adsorbent, contained in a sampling equipment (diffusion tube or sampler badge) with a well defined geometry , in which it is adsorbed and retained for later analysis. Ideally, for specimen that are completely adsorbed the rate of collection in a well designed diffusion sampling equipment is proportional to the air concentration, according to Fick's first law of diffusion. Therefore, the sampled amount is proportional to the desired integrated concentration history.

3.2 Injection of tracer gas

In the homogeneous pulse technique, pulses of tracer gas must be injected in all zones of the building, which forms the ventilated system. The amount of tracer gas injected in a zone is to be proportional to the zone volume. The injection equipment must therefore be constructed so that the injected amount of tracer can be easily controlled.

There are several possible principles for controlling the injection, which are however outside the scope of this paper to discuss.

The division of the building space into zones is a crucial matter. The smaller the zones, the better will the injection approach the ideal homogeneous case. However, the air in a room of ordinary size is usually well mixed and such a room may be treated as a single zone, unless special effects like air short-circuiting is suspected. In all cases, regardless of the zone volume, the injected tracer should be mixed into the air in the zone to yield a uniform concentration immediately after injection. The smaller the zones, the easier is the mixing, so it may be worthwhile to make several injections at evenly distributed different positions even in rooms of normal size.

4. CONCLUSIONS

A new tracer gas technique - the homogeneous pulse technique - for measuring the local mean ages of air in buildings is proposed and discussed from theoretical and practical points of view. In this technique, pulses are homogeneously distributed within the building space. It is not necessary to inject all pulses simultaneously. Especially in large buildings and buildings with many rooms, it is valuable that the pulse distribution can be made in any pace (as long as the air flow patterns in the building do not change). The local mean ages of air can be simply evaluated from the amount of tracer collected in integrating samplers, which may be of the passive diffusive types, commonly used in passive tracer gas technique. The samplers must be active from the moment that tracer gas injection begins until essentially all tracer gas is ventilated out from the building space.

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VENTILATION AND COOLING

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SPATIAL VARIABILITY OF POLLUTION INDUCED BY TRAFFIC IN STREET CANYON

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1. Introduction

Concentration of pollutants produced by car traffic in a street below the roof level has large spatial variations. In a street, pollutants are diluted by the turbulent air flow which is induced by the wind speed above the roof level, and also produced by car displacement. The airflow structure is in relation with street size and building shape. Particularly strong gradients of concentrations can be observed vertically and also horizontally in front and along buildings where are set up ventilation inlets and windows. So it is necessary to take into account this variability to consider the influence of outdoor air upon indoor air quality.

The street canyon case has been studied extensively last years by C.S.T.B. and many other authors. A bibliographical synthesis is presented, including results from field measurements, wind tunnel experiments and numerical simulation.

2. General pattern of pollution in urban area

At the level of a town, background pollution and local pollution must be considered. Industrial sources contribute mainly to create a background level of pollution covering uniformly at least quarters of towns. Chimney exhausts from domestic heating come in addition to this background concentration as the sources are also generally regularly disseminated on the roofs. Car traffic at the opposite acts locally as the source is usually confined in town at the ground level in the streets between buildings. This leads to great differences in spatial distribution of pollutant concentration both between different streets or inside a street. Concentration field in a street will be the result of pollutant dispersion by the mean air flow and by turbulence diffusion from the source. The mean air flow is mainly resulting from general wind conditions above street and turbulence is principally created by the traffic.

3. Airflow structure in the street, influence of wind characteristics

The structure of air flow in the street is function of street geometry and wind direction above roofs. When passing above with a given angle from street axis there is creation of a vortex occupying more or less the volume between building as it is illustrated in the figure n° 1 , taken from Hertel and Berkowicz (1989). This vortex is easily reproduced in wind tunnel and by numerical simulation but fields observations shows important fluctuations in form and intensity of this vortex due to instationary characteristics of incident wind. This vortex allows, through the vertical wind component created, the elimination of pollutant by convection above the roofs. The figure n°2 gives an example of measured vortex (Baranger 1986). At the opposite when the wind blows in the axis of the street the decrease of pollution above the roofs is due only to turbulent diffusion, which may lead to situation of higher concentration in the street. (Hertel, Berkowicz 1989)

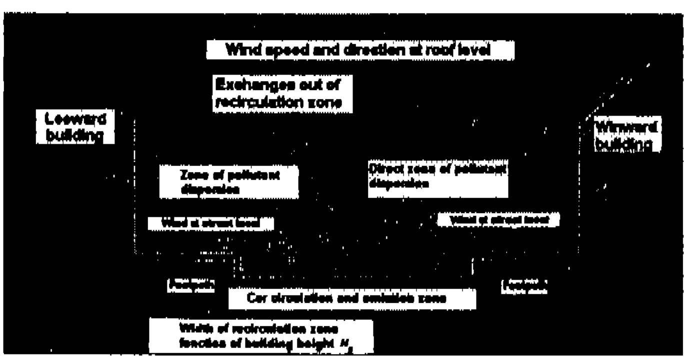

Figure n° 1 – Geometry of wind structure in canyon street according the OSPM model of Hertel and Berkowicz

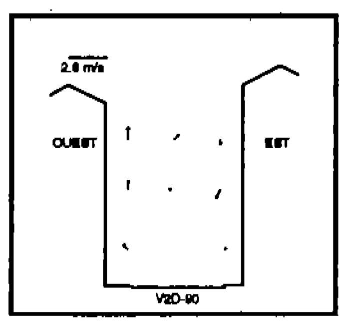

Figure n° 2 – Measured vortex in a canyon street for a wind perpendicular to the street (Baranger, 1986)

4. Concentration field and influence of street geometry

The concentration field in the street is dependent of the form of the vortex, which is dependent of geometrical characteristics of the streets. For example the roof geometry may change considerably the form of the vortex. For similar street dimension, changing from flat terrace roof to normal inclined tile roof changes the simple vortex to a double vortex in the street. The second one will be turning opposite to the first one. The ratio between height and width will determine if the vortex will take all the volume of the street or not and thus the general level of concentration which will be lower. Wind tunnel measurements shows that the critical value where there is a notorious change in street concentration level when the ratio height/width is greater than one (Meroney and Al. 1994). The figure n°3 reproduce numerically an example of pollutant concentration distribution under vortex condition in a canyon street (Delaunay 1997).

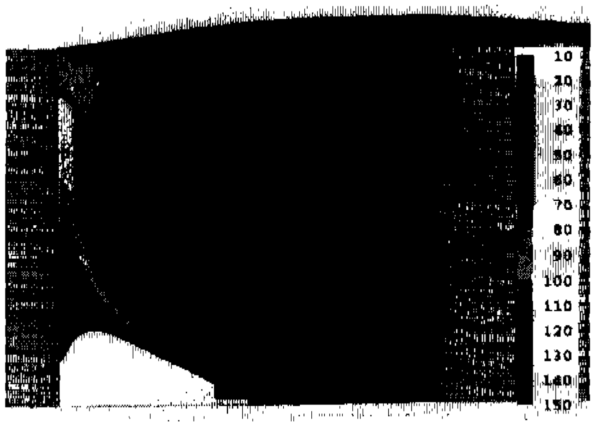

Figure n° 3 – Pollutant concentration repartition under vortex conditions in a canyon street obtained by numerical modelling (Delaunay, 1997)

5. Vertical Variability of concentration level

So when the structure of the wind flow in the street presents the form of a vortex, especially when the wind arrives perpendicular to the street, it may leads to concentration pattern and level different from one side to the other and which is illustrated on the following figure n°4 issued from measurements (Baranger, 1986).

As the source is at ground level, there is a vertical decreasing gradient of the concentration level along the building facades. The order of magnitude of the difference in concentration between bottom and top of the building can be up to a factor 15 in severe configuration, but turns usually around 5.

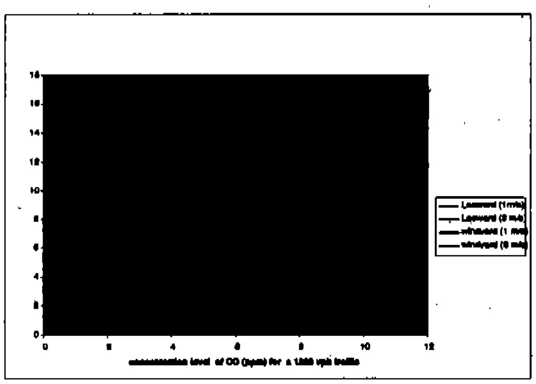

Figure n° 4 – Mean pollution concentration gradient on the leeward and the wind ward side of a canyon street for two classes of wind speed and wind direction perpendicular to the street (Baranger, 1986)

6. Lateral variability of concentration level

Along facades inside the street, differences of concentration level also occur. due to lateral streets which permit evacuation of pollutant if the traffic in lateral street is reduced compared to main street the figure n° 5 presents example of measurements made in Wind Tunnel (Hoydysh, Dabbert 1986). Differences also appear in the case of existence of a nearby particular source as a tunnel or ventilation outlet diffusing concentrated exhaust gases from a confined volume. figure n°6 presents the site of measurements and figure n°7 the results of measurements compared to numerical simulation (Flori and al 1995).

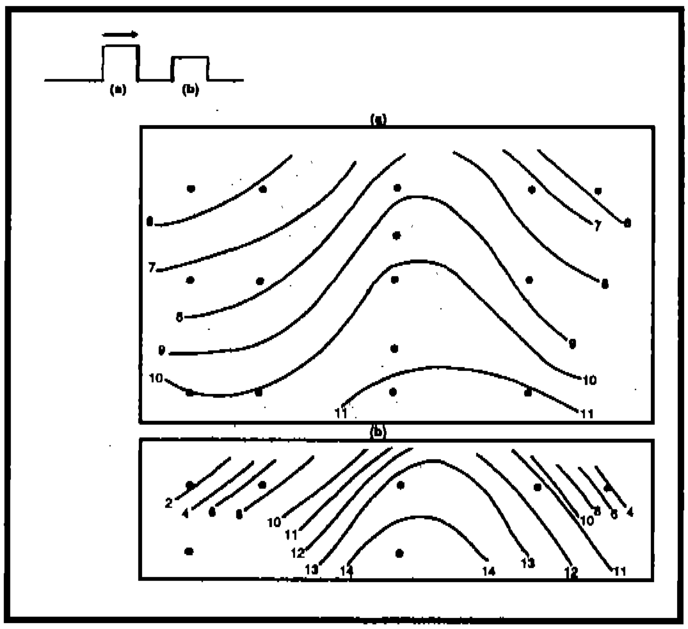

Figure n° 5 – Concentrations level on windward (a) and leeward (b) facades of two buildings with lateral crossing streets (Hoydysh, Dabbert, 1986)

Figure n° 6 – Measurement site for determination of the influence of a tunnel outlet in a canyon street (Flori and al, 1995)

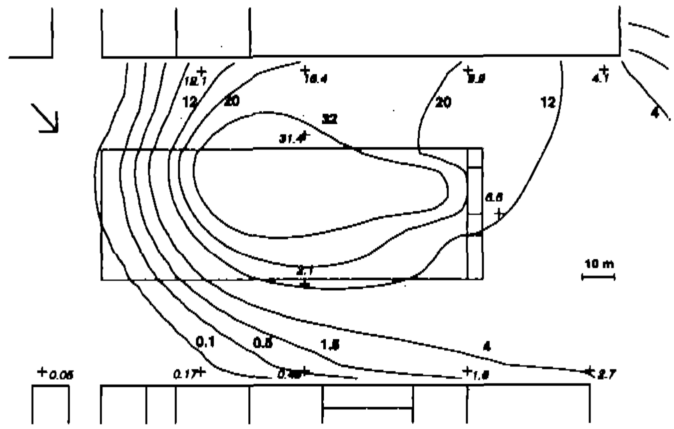

Figure n° 7 – Comparison of numerical simulation and measurements in a canyon street with a tunnel outlet (Flori and al, 1995)

7. Conclusion

Spatial variability of concentration level in a street canyon is strongly dependent of wind characteristics (speed and direction) which can induce particular distribution of concentration along building facades.

Other factors, as geometrical characteristics of the street, can be of influence, in the spatial variability of pollution concentration in a street canyon. Sometimes these factors may have correcting effects on the above mentioned action of wind.

Following factors can be mentioned: the street height/width ratio and the form of roofs but also the localisation and periodicity of traffic lights.

All these factors should be taken in account or at least kept in mind during the ventilation system conception phase.

Acknowledgements

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VENTILATION TECHNOLOGIES IN URBAN AREAS

19TH ANNUAL AIVC CONFERENCE
OSLO, NORWAY, 28-30 SEPTEMBER, 1998

Practical Guidelines for Integrated Natural Ventilation Design

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Synopsis

Natural ventilation in office buildings can sometimes offer other advantages than traditional mechanical ventilation systems. Often natural ventilation systems are promoted at an early stage by an architect, but perceived difficulties, e.g. to pre-determine the function of a natural ventilation system, can serve as a barrier and a mechanical system is often chosen instead. In this paper the difficulty to determine the indoor climate achieved by a natural ventilation system is addressed, both by presenting a computer simulation tool coupling a thermal model with an air flow model and by a statistical analysis of test runs made with the simulation tool to determine the most important parameters for a natural ventilation system. Results from the statistical analyses for evaluating the expected indoor climate determine the influence of different parameters as well as their interactions. It can generally be concluded that natural ventilation design is enhanced by a number of factors, the most important ones being large facade vent areas, limited internal heat loads, possibility for night ventilation and window use during daytime and use of well designed solar shading devices.

Introduction

In many countries, there is a turn towards natural ventilation as an alternative to what many consider energy and cost demanding mechanical ventilation systems. The objective is to save money and energy while maintaining an acceptable indoor air quality and thermal climate, or even to improve the indoor environment by reducing noise levels, giving the user more control over the indoor climate etc.

Often this process is promoted by architects while engineers have a more restricted view of natural ventilation. In an in-depth study (Aggerholm 1998, Svensson & Kronvall 1998) the perceived barriers to natural ventilation held by e.g. architects, consultants, contractors and developers, are identified and one of the major perceived barriers to natural ventilation is the lack of means to predict the indoor climate that can be achieved with a natural ventilation system. Furthermore, in the same study, it was found that pre-design tools for determining the performance of natural ventilation systems were desired.

In this piece of work the need for abilities to determine the indoor air temperature and the ventilation rate have been addressed, with the development of a computer simulation tool and with statistical analysis of systematical simulations of the performance of naturally ventilated office buildings. From this study conclusions of the suitability of natural ventilation in office buildings with different prerequisites can be drawn.

Methods

One of the main objectives of this work is to identify the parameters that have the largest influence, as well as what significance interaction between different parameters have, on the performance of natural ventilation. To achieve this, a large number of prerequisites had to be studied. Thus a computer simulation tool was developed and by changing input parameters in a systematic way, different conditions were studied. These test runs were then statistically analysed and from the statistical analysis, conclusions were drawn on the importance and influence of different parameters. In this chapter, the different steps in the work are described.

Computer Simulation Tool - The *NatVent™* program

In order to be able to predict natural ventilation air flow rates and indoor air temperatures at the design stage, a computer model has been developed within the *NatVent™* -project. The program is an integrated model with a thermal and an air flow model coupled together. It can be used early in the design process to determine possibilities and restrictions in the use of natural ventilation in an office building.

Features

The *NatVent*™ program is set in a typical Windows environment. As a platform, a main window is created. Within this main window, input and output forms may be opened and adjusted to fit the specific building.
The aim for the user interface is to facilitate the use of the program by any building designer, architect or engineer at an early stage. Therefore the interface uses input that are simple to quantify, even at an early stage in the design process.
The input is given by the user step-by-step in four forms describing the building. Under these headings, the relevant input for the different topics are found.

The Location
describes the surroundings and the climate.
The Building
describes the geometry of the building as well as the construction.
The Ventilation Strategy
describes the vents and other ventilation devices, if any.
The Windows
section describes input about the windows.
As the input is given, it is easy to go back to review and to change the input. When the user is satisfied with the input given, the calculation is started with the "Run project" button. For every hour in the studied period values for temperature and ventilation rate are calculated.

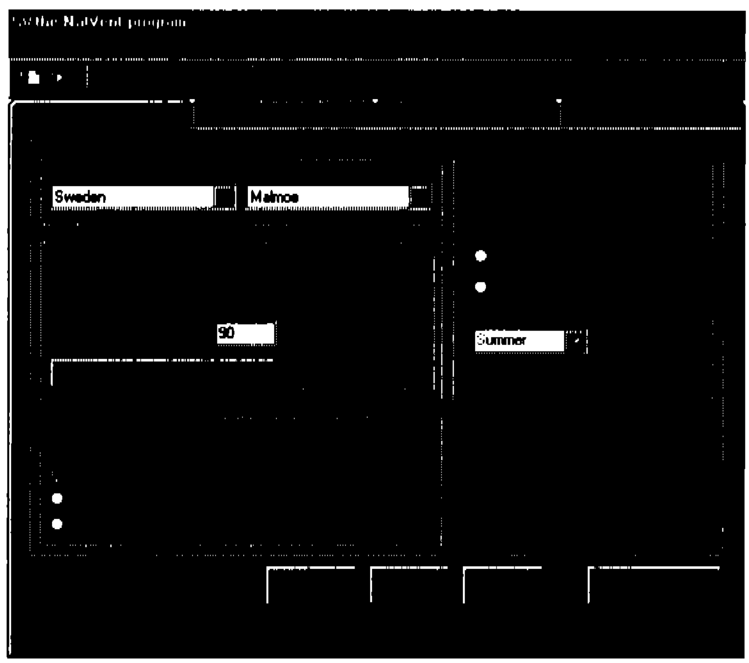

Figure 1 One of the input forms

The Results

A number of different results are extracted and shown in an illustrative way. Depending on the period chosen for the calculations, temperature and air flow rates are presented as duration diagrams or as graphs with hourly values. If the results indicate e.g. an air flow rate too low, it is easy to go back to the input form to change some input to give a higher air flow rate.
In the User's guide criteria for acceptable indoor air temperatures and air flow rates and CO_2 levels are found, as well as suggestions of parameters to change to affect the temperature or ventilation rate in the desired way.

Theory

The program uses a single-zone model. Thus, the entire building or a selected part is represented by only one single zone. The selected part can be either one of the floors or a part of a single floor. The single zone has one temperature and one internal pressure at floor level. The zone is influenced in many ways by the weather, the occupants and maybe by a mechanical ventilation system. To visually illustrate these factors, Figure 2 shows a picture of the thermal paths and the air flow paths that create the temperature and ventilation flows in the zone.

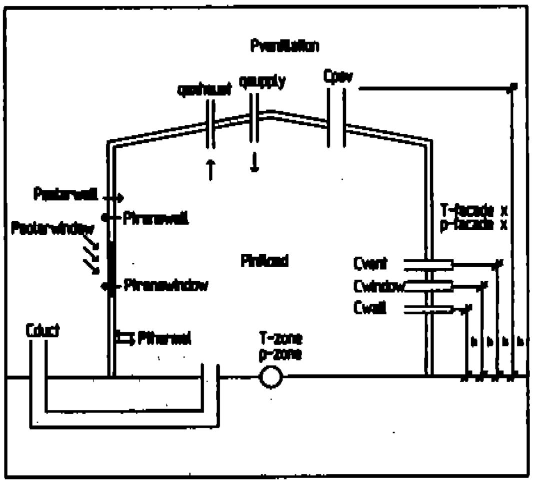

Figure 2: Factors effecting the air flow

To calculate the temperature and the air flow pattern a temperature model, coupled to an air flow, model is needed. To simplify the calculations the indoor temperature of the previous time step is used in the air flow model. Below those two models are introduced.

Thermal model

The solar radiation on external walls and the roof and the insolation through windows and skylight are calculated either from actual weather data e.g. with a test or design reference year or from summer design weather data generated for a stationary hot, calm, clear sky period. The thermal model is a single zone, one time constant model. In the model, it is assumed that all internal structures and surfaces have the same temperature and that the internal air temperature can be averaged to one air temperature representing the hole building or zone. The thermal model is not applied during winter periods and instead a constant temperature is used.

Air flow model

Due to wind, thermal buoyancy and fans, if any, a pressure difference over the building envelope will be created. As a pressure difference occurs over the building envelope, the air is bound to flow from higher pressure to lower pressure and thus an air flow to and from the building will arise.

Air flow to and from the building can take many paths. Air flow through small cracks and imperfections in walls and ceiling, through vents in the façade, through window airing, through ducts for supply air or passive stacks and through a forced flow from fans - if any. The equations for each one of these links between the outside and the building are determined and by an iteration process, the internal pressure is found when the mass flow rates to and from the building are equal. These flows are then used in the thermal model.

Parametric study

The Test Runs

The significance of different parameters and the different parameters' interaction, with respect to the indoor climate, was studied. In order to study the influence of different parameters on the performance of natural ventilation, systematic test runs were made. A "typical" office building was chosen and then different input parameters were systematically altered.

The Building

The building is chosen to be a typical office building, with a general layout of the building as in Figure 3. It is symmetric with a length of 21 meters and a width of 10 meters. The room height is 2.5 meters and the height of the intermediate floor is 0.3 meters. The offices are cellular offices along the two parallel longer façades. Each office room is 3 meters wide and 4 meters deep and the building is divided by a two meters wide corridor. There are a total of 14 office rooms on each floor. Each office room has a window and a vent (if vents are used). With this basic design, different parameters e.g. thermal mass, window area etc. can be changed. Half of the office rooms are south facing and half of them are north facing.

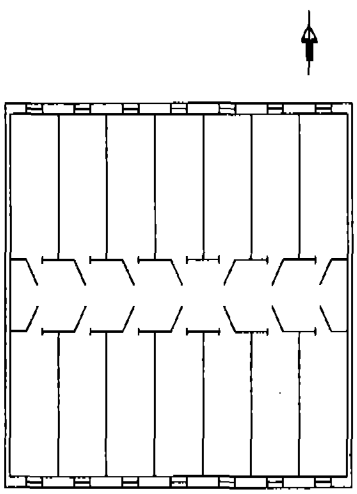

Figure 3 A general lay-out of the studied building.

Climate

The computer simulation tool contains test or design reference years for seven locations in six different countries in Europe. The climates used for the test runs were from Switzerland, the Netherlands and Norway.

Ventilation Strategies

For the building, five different ventilation strategies were studied, see Table 1. These five cases represent ventilation with passive stacks and ducted air supply; with passive stacks; with ducted air supply and skylight; with skylight and with cross ventilation, i.e. ventilation with windows and vents. All five cases are being addressed in the same way as described.

Table 1 Five different ventilation strategies.

Case	1	2	3	4	5 *)
Passive Stacks	x	x	-	-	-

Ducted Air Supply	x	-	x	-	-
Skylight	-	-	x	x	-

*) For case 5 cross ventilation, i.e. ventilation by means of vents and windows, was used.

Important Parameters

For the test runs, a number of important parameters were studied. By doing some initial test runs in a simplified scheme a number of parameters were determined to have a high significance on the indoor temperature and/or the ventilation rate. Some of the parameters were then grouped, i.e. as it is likely to have the same degree of insulation in the roof as in the walls those two parameters were treated as one. The final parameters were climate (i.e. temperature, wind conditions and solar radiation); number of storeys; air tightness of the building envelope; U-value (average value of walls and roof); thermal mass; vent size; internal heat loads; if there was night ventilation or not; fenestration; window opening; solar shading (i.e. external or internal solar shading and overhang over the window); and the type of windows (i.e. the U-value and transmittance of the glazing). The values of the parameters can be found in Table 2.

Table 2 Parameters for the test runs. There are three levels for all parameters except for night ventilation, which either is applied or not.

Parameter	**-1**	**0**	**1**	
Climate	Norway	The Netherlands	Switzerland	
Number of Storeys	2	6	12	-
Air Leakage	1	5.5	10	l/sm^2 at 50Pa
U-Value	0.2	0.4	0.8	$W/m^2/K$
Thermal Mass	40	100	160	$Wh/K/m^2$
Equiv. Vent Size Facade 1 & 3	0	420	840	cm^2
Internal Heat Loads	15	27.5	40	W/m^2
Night ventilation	None		2	$l/s/m^2$
Fenestration	1	3	5	m^2 per office room
Windows open	0	0.25	0.50	m^2 per office room
Solar Shading ***Solar shading*** ***Overhang***	 None (100) None (0)	 Internal (60) Medium (40)	 External (20) Large (60)	 (% of radiation through) Angle of overhang (°)
Type of windows ***U-Value*** ***Transmittance***	1 pane 5 0.85	2 panes 2.7 0.75	Energy glazing 1.6 0.65	 $W/m^2/K$

Criteria of critical performance

The criteria of critical performance that were studied were the outdoor air ventilation rate below 0.7 $l/s/m^2$, the indoor air temperature exceeding 25°C and the indoor air temperature exceeding 28°C. The number of hours the ventilation rate was below the limit during work hours were registered as well as the numbers of work hours the temperature did exceed the lower and/or the higher of the two temperature limits. The number of work hours were calculated from a year that holds approximately 261 weekdays of 8 work hours + a lunch hour. This gives 2349 work hours per year.

Statistical Analysis

With twelve parameters and with three levels ("low", "medium" and "high") for each parameter, except for the night ventilation with only two levels ("applied" and "not applied") the number of test runs should be 3^{11}x2 (= 354 294) in order to try all combinations. Instead of this time consuming work fractional factorial design of the experiment has been used. The test runs were planned, and the outputs from them were analysed statistically. By using this method, the number of simulations has been reduced to 90. As totally five ventilation strategies are studied, the total number of simulations has been reduced from 1 771 470 to 450.

The Statistical Analysis Tool

The results have been examined statistically by using a PC-Windows program called MODDE (Umetri, 1997). This is a program for the generation and evaluation of statistical experimental designs. A simulation plan has been made in an attempt for extracting the maximum amount of information from the fewest number of simulations. In this study screening design has been used, using fractional factorial design method and partial least square model (Fisher, 1990). In a set of test runs the parameters are varied simultaneously and with a mathematical model the results are combined.

The output from the statistical analysis tool is in the form of normalised coefficients for the different parameters. At the same time parameters, having significant interactions, can be found. By selecting the parameters and interactions that have significant value, as well as erasing the parameters where the coefficients are not statistically verified, a regression equation can be created.

Output from the statistical analysis

From the statistical analyses, normalised coefficients showing the different parameters' influence on the number of hours with the outdoor air ventilation (vent < 0.7 l/s per m^2), or the indoor air temperatures (temp > 25 and temp > 28 degC) exceeding the limits. The input parameters are as in Table 2.

For both the ventilation criteria and the two temperature criteria, constants ($a_1 - a_n$) are determined by erasing the parameters where the standard errors are too large compared to the actual value of the coefficient. With these constants, three different equations, one for each criterion, can be determined. The equation is written on the form e.g.:

$$\text{Work hours with limit exceeded} = 10^{(a0\ +\ a1*\text{parameter1}\ +\ a2*\text{parameter2}\ +\ \ldots an*\text{parametern})}$$

where the parameters (parameter_1 to parameter_n) are presented as -1, 0 or 1, representing "low", "medium" and "high". The actual values of the parameters are found in Table 2, while the resulting coefficients from the statistical analyses are shown in table 3.

Table 3 *Results of the statistical analyses*

	Case 1 Passive stacks Ducted air supply			Case 2 Passive stacks			Case 3 Ducted air supply Skylight			Case 4 Skylight			Case 5 No vent devices		
	VENT	T25	T28	VENT	T25	T28	VENT	T25	T28	VENT	T25	T28	VENT	T25	T28
Parameter	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**	**Coeff.**
Climate Netherlands chosen	-0.81	+0.03		-0.66	-0.10		-0.82			-0.31	-0.06		-0.28		
Number of storeys Two chosen		-0.15	-0.22	-0.78		-0.14		-0.07	-0.12	-0.79		-0.11	-0.80	-0.07	-0.11
Air leakage Low chosen				-0.78	-0.07	-0.12	-1.10	-0.06	-0.08	-0.66	-0.06		-0.61	-0.06	
U-value		+0.05	+0.08					+0.03	+0.07						
Thermal mass		-0.10	-0.17		-0.06	-0.12		-0.07	-0.14		-0.04	-0.10		-0.04	-0.10
Eq. Vent size	-0.88			-1.13	-0.05	-0.07	-0.71		-0.06	-0.50			-0.55		
Int. heat loads	-1.54	+0.16	+0.22	-1.05	+0.17	+0.26	-0.90	+0.16	+0.24	-0.49	+0.17	+0.25		+0.17	+0.25
Night ventilation					-0.07	-0.10		-0.07	-0.11		-0.10	-0.15		-0.10	-0.16
Fenestration		+0.07	+0.12		+0.04	+0.10		+0.05	+0.11			+0.08			+0.07
Windows open	-1.04	-0.06	-0.10	-0.67	-0.09	-0.15	-0.39	-0.66	-0.10		-0.09	-0.15	-0.55	-0.09	-0.15
Solar shading	+1.52	-0.19	-0.29	+0.82	-0.17	-0.27	+0.97	-0.18	-0.28	+0.61	-0.16	-0.27	+0.61	-0.17	-0.28
Type of windows															
Unit	h	h	H	h	h	h	h	h	h	h	h	h	h	h	H
Base case	0	339	160	3	644	382	19	516	272	1 294	706	366	1 552	762	397
Optimise VENT	0	661	412	0	885	670	0	845	526	190	1 197	993	336	912	467
Optimise T25	1	86	28	15	158	20	665	83	19	541	99	21	1 622	124	19
Optimise total	0	86	28	15	158	20	46	83	17	173	99	21	24	93	19
Measure(s) taken for total optimisation	Increased vent size			None			Increased vent size			Increased vent size			Increased vent size Increased leakage		

Optimisation

Different parameters influences on indoor climate performance – ventilation rates and indoor temperatures – are shown in table 4.

Table 4 Generalised influences of different parameters on indoor climate performance.

Parameter	Influence on ventilation rate		Influence on indoor temperatures	
	In-flu-ence	**Comments**	**In-flu-ence**	**Comments**
CLIMATE				
- cold	+	Cool / windy summer conditions beneficial	+	Cool / windy summer conditions beneficial
- warm	-	Less thermal forces for natural ventilation	--	High thermal impact on indoor temperatures
NUMBER OF STORIES				
- low rise	--	Decreases flow rates	0	
- high rise	++	Increases flow rates	0	
AIR LEAKAGE OF ENVELOPE *)				
- low leakage	(--)	Decreases flow rates, but necessary for flow control!	(-)	Increases peak temperatures.
- high leakage	(++)	Increases flow rates But energy wasting etc. To be avoided!	(+)	Decreases peak temperatures. But energy wasting etc. To be avoided!
U-VALUE				
- low	0		+	Decreases peak temperatures
- high	0		-	Increases peak temperatures
THERMAL MASS				
- light building	0		-	Increases peak temperatures
- heavy building	0		++	Beneficial
FACADE VENTS				
- no vents	--	Decreases flow rates substantially	-	Increases peak temperatures
- large vent area	++	Increases flow rates substantially.	+	Decreases peak temperatures
INTERNAL HEAT LOADS				
- low	(--)	Decreases flow rates	++	Decreases peak temperatures substantially
- high	(++)	Increases flow rates	--	Increases peak temperatures substantially
NIGHT VENTILATION				
- no night ventilation	0	(Only working hours ventilation considered)	--	Increases peak temperatures substantially
- night vent. in use	0		++	Decreases peak temperatures substantially
FENESTRATION				
- few / small windows	0		+	Decreases peak temperatures
- many / large windows	0		-	Increases peak temperatures
WINDOWS OPEN				
- never	--	Decreases flow rates substantially	-	Increases peak temperatures
- a lot	++	Increases flow rates substantially	+	Decreases peak temperatures
SOLAR SHADING				
- no solar shading	(++)	Increases flow rates substantially	--	Increases peak temperatures substantially
- large external	(--)	Decreases flow rates substantially	++	Decreases peak temperatures substantially
WINDOW TYPE				
- simple single glazing	0	No apparent effect	0	No apparent effect
- advanced triple energy glazing	0	No apparent effect	0	No apparent effect

Conclusions

Some general conclusions regarding natural ventilation design are:

- It is important to carefully study the conditions for a building before deciding on the ventilation system.
- It is essential that some key parameters are kept under adequate control if natural ventilation should be an alternative.
- Under certain circumstances e.g. high thermal loads, natural ventilation is not an acceptable system.

From this specific study, it can be concluded that these factors are beneficial for natural ventilation performance

- Higher buildings
- Airtight buildings *)
- Well insulated envelopes
- High thermal mass
- Large area of (adjustable) facade vents
- Limited internal heat loads
- Night ventilation
- Minimised window area
- Active use of windows
- Effective solar shading

*) The leakage of the envelope is a parameter that must be handled with care. Although the results of the parameter study indicates more favourable performance regarding both ventilation rate and indoor temperatures with increasing leakage, it must be kept in mind that the leakier the envelope is the more difficult it is to control the ventilation air flow rates and more energy is wasted during the heating season. Thus, the optimum strategy must be to build tight and ventilate right – i.e. to give opportunities to control the airflow rates by means of facade vents and window opening.

Acknowledgements

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The Air Infiltration and Ventilation Centre provides technical support in air infiltration and ventilation research and application. The aim is to provide an understanding of the complex behaviour of the air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

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