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Nr 180 THE USE OF TRACER GAS FOR DETERMINING VENTILATION EFFICIENCY

Mats Sandberg and Anders Svensson

When premises are to be ventilated, two problems are often the most interesting. These are:

- creating a sufficient ventilation flow with regard to cooling or heating requirements or the amount of pollution, and
- side-effects resulting from flow generating locally high air velocities.

From energy and comfort viewpoints it is of greatest importance that ventilation is available in those places in a building where it is required. In this respect different types of ventilation systems have different efficiency (ventilation efficiency). When designing a ventilation system it is therefore important for the designer to know how the actual air change rate in different parts of an occupied zone varies according to different parameters and operational conditions.

The detailed measurement of air movements in ventilated premises is technically extremely difficult and only after complicated calculations can measurements be used for the determination of ventilation efficiency. From a practical point of view the use of tracer gas is therefore a more attractive method. But even this method involves a number of difficulties. The greatest problem is the interpretation of concentration curves. The slope of these curves cannot generally be taken as a measurement of 'air change rates'.

At the National Swedish Institute for Building Research (SIB) a project is in progress which primarily aims to determine the relationship between ventilation design and ventilation efficiency. This project aims to:

- determine variations in air change rates between different parts of an occupied zone and to express these as the degree of efficiency or an efficiency factor, and to
- determine what proportion of supply air directly disappears with exhaust air (the short-circuiting effect).

Factors affecting the flow pattern in a room will also affect the air change rates in the occupied zone and the short-circuiting air flow. Such factors are:

- the type and positioning of supply and exhaust air terminals,
- the supply air temperature and flow, and
- the shape and size of the room.

This article discusses the various definitions of ventilation efficiency. This is followed by a study of the interpretation of concentration curves by the detailed examination of a theoretical model case. The implications of definitions of ventilation efficiency are examined by applying them to the results of the model case. In conclusion, preliminary results are presented of measurements of ventilation efficiency tests carried out at full scale in the laboratory.

Definitions of ventilation efficiency

The definition used shall express the system's ability to evacuate pollution from a particular source in a room. In this context pollution is interpreted in general terms and therefore includes e.g. smells, radon and overtemperature (although the source of the latter may lie outside the particular room being investigated, this will not affect the ensuing discussion). For reasons of convenience it is assumed in the following that pollution is something that can be expressed in units of kg/m^3 or ppm. The definition of ventilation efficiency should meet two important criteria:

- it must be *directly* linked to the characteristics of the system described, and
- it must be *opera*
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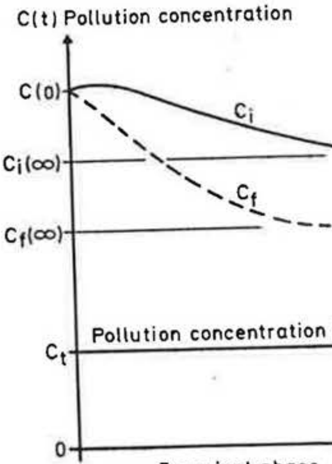

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- it must be *operational* i.e. it must indicate how efficiency is to be measured.

The definition of ventilation efficiency can be based preliminary on two characteristics of the ventilation system to be described:

- the *relative ventilation efficiency*, which expresses how the system's ventilation capability varies between different places in a room, and
- the *absolute ventilation efficiency*, which expresses the ability of the ventilation system to reduce a pollution concentration in relation to the feasible theoretical maximum.

What this involves is shown in figure 1.

Figure 1 shows the development of concentrations in different parts of a room when at time t = 0 there is an even distribution of pollution concentration in a room, and when there is a homogenous source of pollution, i.e. the production of pollution is constant.

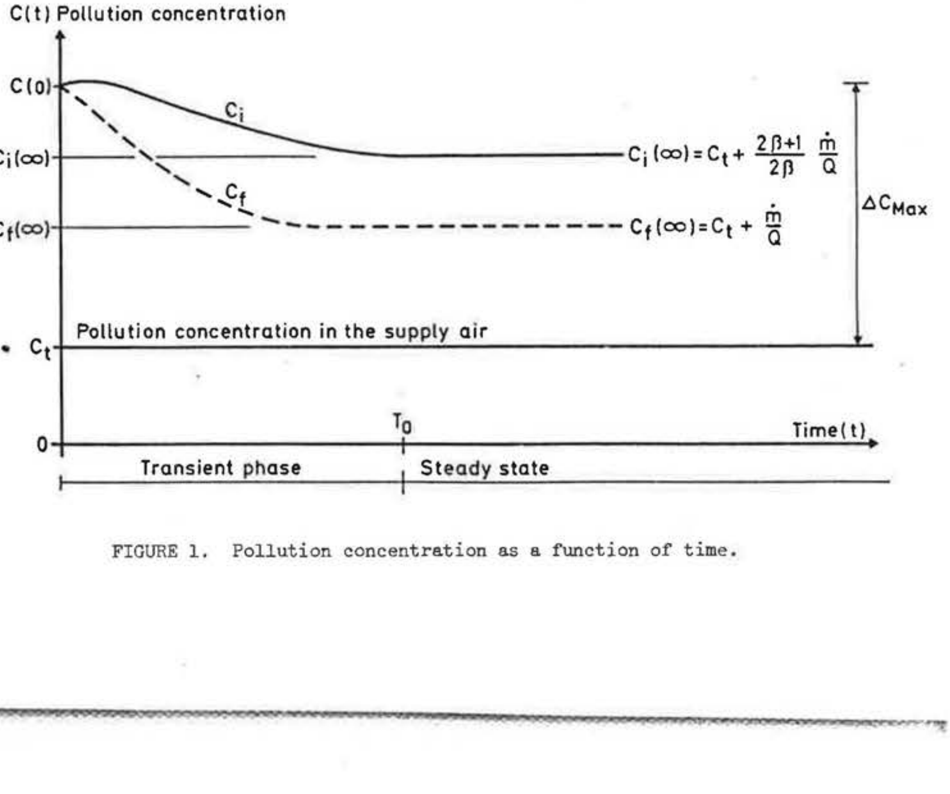

FIGURE 1. Pollution concentration as a function of time.

Using values of the *steady state condition* the *relative ventilation efficiency* (ε_i^r) can be defined as

$$\varepsilon_i^r = \frac{C_f - C_t}{C_i - C_t} = 1 + \frac{C_f - C_i}{C_i - C_t} \qquad (1)$$

When the pollution content of the supply air is equal to zero then

$$\varepsilon_i^r = \frac{C_f}{C_i} \qquad (2)$$

When $C_i = C_f$ then $\varepsilon_i^r = 1$

The relative ventilation efficiency is always positive and can be greater than 1 in local parts of the room.

Definitions (1) and (2) relate to the definition of ventilation efficiency given by Rydberg (2).

The *absolute ventilation efficiency* (ε_i^a) is defined as

$$\varepsilon_i^a = \frac{C(0) - C_i}{\Delta C_{max}} = \frac{C(0) - C_i}{C(0) - C_t} \qquad (3)$$

When $C_i = C_t$ then $\varepsilon_i^a = 1$.

The absolute ventilation efficiency is always less than 1 and can be negative.
The above definitions are primarily a measurement of the ventilation system's efficiency over a 'long-term' period and therefore a poorer measurement of its ability to remove 'transients'.

The absolute ventilation efficiency is the most interesting measurement of a system's performance. The motive for introducing the relative ventilation efficiency is that it can also be measured during non-stationary conditions. This is illustrated later.

Ventilation efficiency can be determined primarily in two ways:

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The direct method. An experimental arrangement is set up to imitate the actual conditions under which the ventilation system is to operate. The stationary pollution concentrations are quantified and inserted in definitions (1) to (3). In this context it may be advisable to design a number of standard cases.

The indirect method. Attempts are made to identify quantifiable 'system parameters' which characterize the system's ability to evacuate pollution. These parameters can be used to calculate (estimate) ventilation efficiency as defined in (1) to (3).

The most common system parameter, which many attempt to introduce (see e.g. 3), is the so-called local air change rate. It is demonstrated in the example below that there is generally no *unequivocal* local air change rate which characterizes a system. The problem of defining a local air change rate is related to the fact that a room is generally part of an interlinked system. This means that changes in pollution concentration (tracer gas concentration) depend on the dilution process in other parts of the room. It is only under ideal conditions, i.e. complete admixture, that a local air change rate exists.

The interpretation of concentration curves by comparison with a model case

By studying a simplified, although realistic, model case of the dilution process in a room, a closer idea can be obtained of the quantities giving rise to the concentration curves studied. Figure 2 illustrates a room divided into two equal control volumes V. The lower volume represents the occupied zone while the other represents the exhaust and supply air zones which in the following are referred to solely as the exhaust air zone. In both the supply and exhaust air zones there are ideal stagnation zones, i.e. areas where air change with the surroundings does not take place. This means that the active volume, V_1 respectively V_2, is less than that of the control volume.

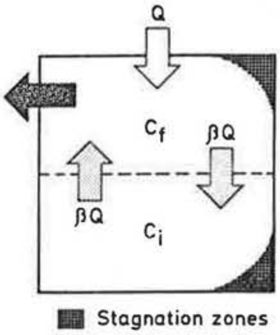


FIGURE 2. The room in the model case.

For each active volume admixture is complete whereas the mixing of the two volumes can vary from zero ($\beta = 0$) to complete mixture ($\beta \to \infty$).

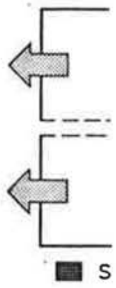
The pollution concentration in the supply air, C_t, is assumed to be equal to zero and the pollution source is assumed to be homogenous. Concentration as a function of time, t, is then given by the equations:

$$\begin{aligned} C_f(t) &= C_f^{tr}(t) + \frac{\dot{m}}{Q} \\ C_i(t) &= C_i^{tr}(t) + \frac{\dot{m}}{\frac{2\beta}{2\beta+1}Q} \end{aligned} \tag{4}$$

When C_f^{tr} and C_i^{tr} stand for the *transient parts* of the solution, i.e. when $t \to \infty$ then

$$\left.\begin{aligned} C_f^{tr}(t) &\to 0 \\ C_i^{tr}(t) &\to 0 \end{aligned}\right\} \text{which means that} \left\{\begin{aligned} C_f(t) &\to \frac{\dot{m}}{Q} \\ C_i(t) &\to \frac{\dot{m}}{\frac{2\beta}{2\beta+1}Q} \end{aligned}\right. \tag{5}$$

The flow $\frac{2\beta}{2\beta+1}Q$ can be interpreted as the amount of fresh air drawn into the occupied zone in the 'long-term'. In other words, if the exhaust air and occupied zones are provided with fresh air flows Q and $\frac{2\beta}{2\beta+1}Q$ respectively, see



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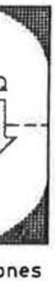

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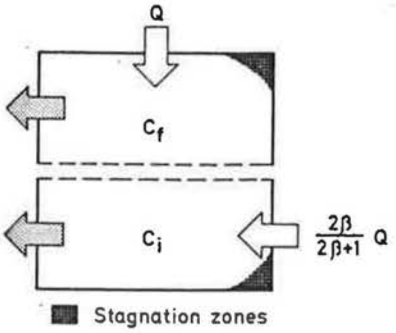

FIGURE 3. Equivalent model at steady state.

figure 3, then the stationary concentration will be the same as equation (5). Factor $\frac{2\beta}{2\beta + 1}$ will correspond, in accordance with this model, to the admixture factor γ as defined in reference (1). However, the transient behaviour of the tracer gas curves is considerably more complicated, see equations (6) and (7) below, than what is given by the latter model. Only the stationary conditions are the same.

The transient solutions are equal to

$$\begin{aligned} C_f^{tr}(t) &= K_1\, a_{11}\, e^{-\lambda_1 t} + K_2\, a_{12}\, e^{-\lambda_2 t} \\ C_i^{tr}(t) &= K_1\, a_{21}\, e^{-\lambda_1 t} + K_2\, a_{22}\, e^{-\lambda_2 t} \end{aligned} \qquad (6)$$

Where

K_1, K_2 = Constants dependent on the initial concentration i.e. C(0)

$\lambda_1(\frac{Q}{V_1}, \frac{Q}{V_2}, \beta)$, $\lambda_2(\frac{V}{V_1}, \frac{V}{V_2}, \beta)$ = The eigenvalue to the equation system

$a_{11}(\frac{V}{V_1}, \frac{V}{V_2}, \beta)$, $a_{21}(\frac{V}{V_1}, \frac{V}{V_2}, \beta)$ = The components in the eigenvector to λ_1

$$a_{12}(\frac{V}{V_1}, \frac{V}{V_2}, \beta), \quad a_{22}(\frac{V}{V_1}, \frac{V}{V_2}, \beta) = \text{The components in the eigenvector to } \lambda_2$$

The eigenvalues are equal to

$$\lambda_{1.2} = n_0\{+(\frac{V}{V_1} + \beta(\frac{V}{V_1} + \frac{V}{V_2})) \mp \sqrt{(\frac{V}{V_1} + \beta(\frac{V}{V_1} + \frac{V}{V_2})^2 - 4\beta\frac{V^2}{V_1V_2})}\} \tag{7}$$

where $n_0 = \frac{Q}{2V} = \frac{Q}{V_r}$ is the *nominal air change rate.*

Please note that both the eigenvalues and the eigenvectors are functions of the linking factor β.

From the above expression for λ_1 and λ_2 it appears that

$$|\lambda_1| < |\lambda_2|$$

This means that when a sufficient period of time has elapsed, say $t > T_0$, the other terms in (6) can be neglected, and we then obtain for $t > T_0$:

$$\begin{aligned} c_f^{tr}(t) &\sim K_1\ a_{11}\ e^{-\lambda_1 t} \\ c_i^{tr}(t) &\sim K_1\ a_{21}\ e^{-\lambda_1 t} \end{aligned} \tag{8}$$

i.e.

$$\frac{c_f^{tr}(t)}{c_i^{tr}(t)} \sim \frac{a_{11}}{a_{12}} \equiv b \tag{9}$$

The ratio b is *constant* and equal to

$$b = \frac{2\beta\ \frac{V}{V_1}}{\frac{V}{V_1} + \beta(\frac{V}{V_1} - \frac{V}{V_2}) + \sqrt{(\frac{V}{V_1} + \beta(\frac{V}{V_1} + \frac{V}{V_2}))^2 - 4\beta\frac{V^2}{V_1V_2}}} \tag{10}$$

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$$\frac{V}{V_1} + \frac{V}{V_2}))^2 - 4\beta\frac{V^2}{V_1V_2} \qquad (10)$$

In spite of the curves separating, the ratio is constant irrespective of the value of β, i.e. independent of the differing 'changes' in both volumes, see figure 4.

From figure 4 and the foregoing discussion, the following conclusions can be drawn about a linked system (which is what one always has in practice):

- For all $\beta \to \infty$, i.e. independent of what 'air change rates' has taken place in different parts of a room, the slope of the curves will vary until they are parallel.
- When the curves become parallel the slope is determined by the *slowest* component in the system.
- A certain minimum period of time, T_0, is required before the system's ventilation capability has completely taken effect.

From this it follows that the slope of a curve cannot be interpreted as a measure of the 'air change rate' at the point in question.

In this particular case there is nothing that *unequivocally* can be defined as a local air change rate.

On the other hand, in the case of *complete admixture* ($\beta \to \infty$) then

$$\beta \to \infty \Rightarrow \begin{cases} -\lambda_1 \to -\dfrac{Q}{V_1 + V_2} = -n_0\dfrac{2V}{V_1 + V_2} \\ -\lambda_2 \to -\infty \\ b \to 1 \end{cases}$$

The process can be characterized unequivocally by the slope of the curves and a local air change rate can be defined, which in this case is equal to the nominal air change rate.

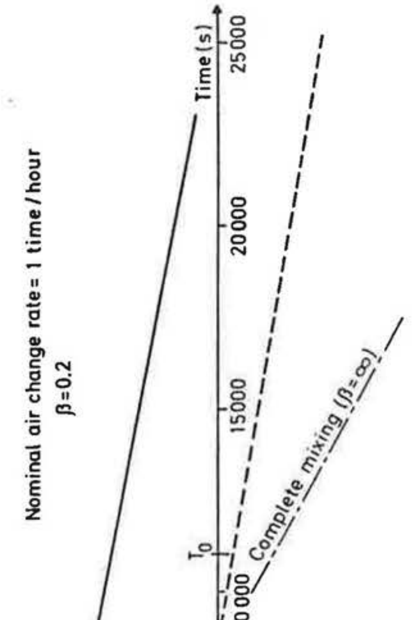

Fig. 4a

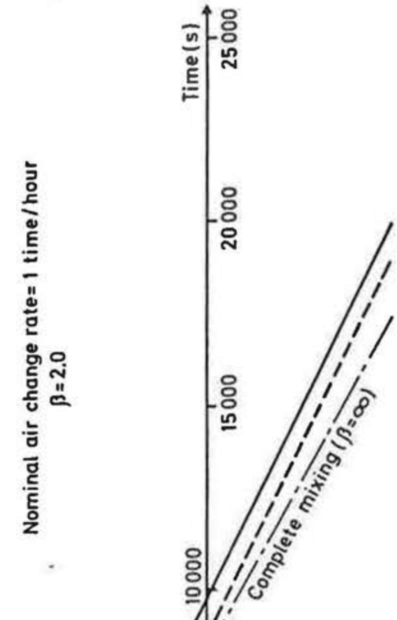

Fig. 4b

FIGURE 4. The transient solution as a function of time for different β.

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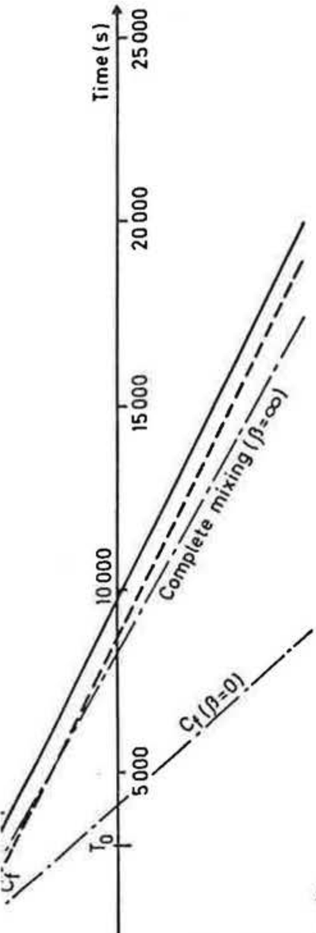

Fig. 4b

FIGURE 4. The transient solution as a function of time for different β.

The stationary concentration values according to equation (5) inserted in the definition of *relative ventilation efficiency* gives

$$\varepsilon_i^r = \frac{C_f}{C_i} = \frac{2\beta}{2\beta + 1} = \frac{1}{1 + \frac{1}{2\beta}} \qquad (11)$$

and the absolute ventilation efficiency in the occupied zone will be

$$\varepsilon_i^a = \frac{C(0) - C_i}{C(0)} = 1 - \frac{C_i}{C(0)} = 1 - \frac{\dot{m}}{Q \cdot C(0)} \left(\frac{2\beta + 1}{2\beta}\right) \qquad (12)$$

In the case of complete admixture ($\beta \to \infty$), (10) and (11) will be

$$\varepsilon_i^r = 1 \quad \text{and} \quad \varepsilon_i^a = 1 - \frac{\dot{m}}{Q \cdot C(0)}$$

That a certain amount of time, T_0, is required before the ventilation system's capability has completely taken effect, gives rise to measurement consequences. When admixture is incomplete the period over which measurements are carried out must be longer than T_0. However, where complete admixture has occurred then it is sufficient to carry out measurements until a statistically acceptable estimation of the curve's slope has been obtained.

Methods for the determination of ventilation efficiency

In the discussion of the advantages and drawbacks of the different methods of measurement, the *model case* in the foregoing sections is here used as a *reference case*, i.e. the demands placed on the measurement method are that it will provide a 'measurement' of ventilation efficiency that, preferably, is equal to ratios (11) and (12) or is, at least, proportional to these. In order to simplify the expression for ratio b according to equation (10), we assume here that no stagnation zones exist in the model case.

Two methods will be discussed in greater detail.

Method A (The 'decay method')

- The room is filled with tracer gas. With the aid of fans the gas is mixed to an even concentration C(0).
- The fans are turned off and the decay of the tracer gas concentration is continuously recorded.

Method B (The 'source method')

- A constant flow of tracer gas is admitted to the supply air duct.
- The growth of the tracer gas concentration is recorded at different points.

Discussion of methods

Methods A and B are theoretically the same. Concentration curves are given by the transient equations (6). Only the initial conditions are different. This means that only the constants K_1 and K_2 are dependent on the method used. See figure 5 for the relationship between method A and B.

In practice, however, there is a difference between these methods. For the experimental method A there is an initial oscillation process which gives rise to error.

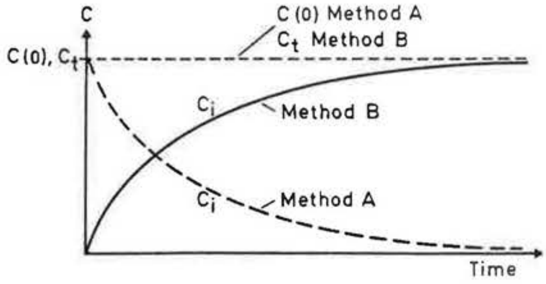

FIGURE 5. The relationship between method A and B.

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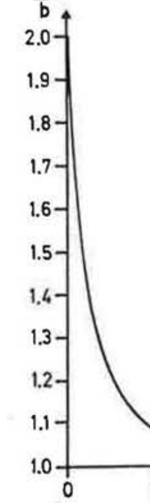

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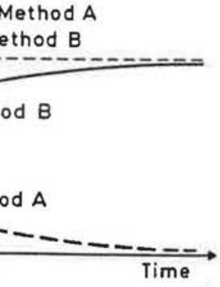

en method A and B.

Apart from this difference the methods are identical. This means that conclusions about the transient process, discussed above, involve both methods. Thus the ratios for $t > T_0$ will be

Method A

$$\frac{C_f}{C_i} = b \;; \quad \text{i.e. constant}$$

Method B

$$\frac{C_t - C_f}{C_t - C_i} = b \;; \quad \text{i.e. constant}$$

These ratios can be used in estimates of relative ventilation efficiency since in the reference case

$$b = \frac{2\beta}{1 + \sqrt{1 + 4\beta^2}} \quad \text{and} \quad \varepsilon_i^r = \frac{2\beta}{2\beta + 1}$$

for *small* β the ratio b will be: $b \approx \beta < \varepsilon_i^r$

for *large* β the ratio b will be: $b \approx \frac{2\beta}{1 + 2\beta} = \varepsilon_i^r$

This means that for small β the fixed ratio b gives an *underestimate*, (see figure 6), of the relative ventilation

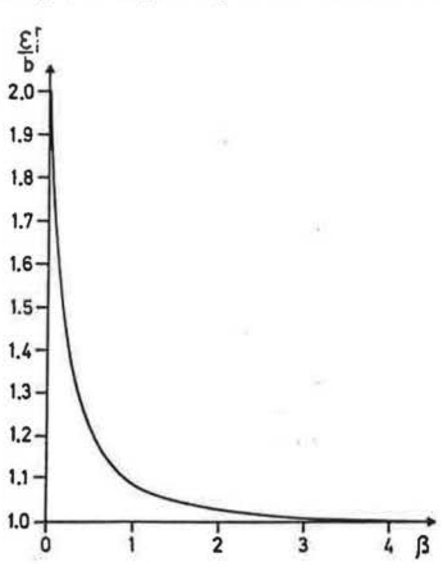

FIGURE 6. The ratio $\frac{\varepsilon_i^r}{b}$ as a function of β.

efficiency in the operational case presented in figure 1. However, ratio b is proportional to ventilation efficiency, as defined previously, and is in other words an *index of ventilation efficiency*. If on the other hand, the operational case had been the same as the experimental one, then ratio b would, by definition, give an 'exact' estimate of relative ventilation efficiency.

Measurements

Measurements have been carried out in a room measuring (width x length x height) = 3.6 x 4.2 x 2.7 m (see figures 7 and 8). The supply air terminal was mounted centrally in the ceiling.

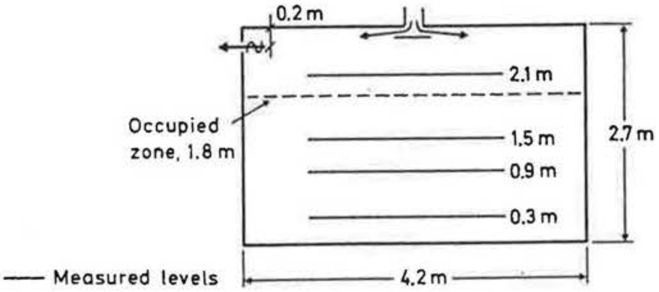

FIGURE 7. Sketch of test room.

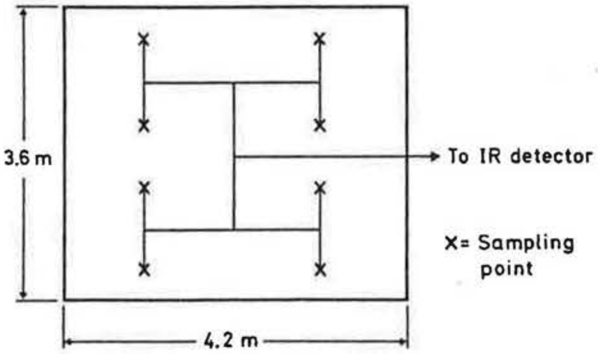

FIGURE 8. Vizualisation of the measurement arrangements at different levels above the floor (0.3, 0.9, 1.5 and 2.1 m).

The exhaust air termi
beneath the ceiling. T
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3, 6 or 9 m/s.

A number of sensors w
0.9 m 1.5 m and 2.1 m
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The supply air flow i
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The investigation was
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Examples of the resu
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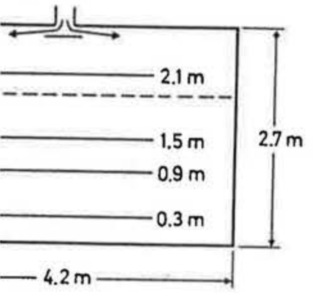
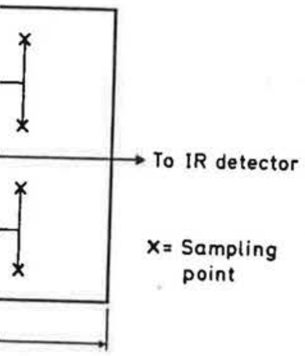




easurement arrangements
ove the floor (0.3, 0.9,

The exhaust air terminal was placed above the door and 0.2 m beneath the ceiling. The slot height of the supply air terminal was varied so that the velocity of supply air was approx. 3, 6 or 9 m/s.

A number of sensors were placed in the room at levels 0.3 m, 0.9 m 1.5 m and 2.1 m above the floor, and one was placed in the exhaust air duct. A sketch of the room is shown in figure 7.

The supply air flow into the test room was varied to correspond to 1, 2, 3 or 4 air changes. The supply air temperature was also varied to obtain overtemperatures of 0, 2, 4 or 8°C.

The investigation was based on measurements of tracer gas at these different levels in order to determine the tracer gas concentrations. The tracer gas used was 'laughing gas' (N_2O). Measurements of the tracer gas concentrations at these levels and in the exhaust air duct were made alternatingly at the electric valves in the tubes connected to the measurement points. With the aid of a computer these valves were regulated so that one at a time was linked to the tracer gas analyzer. Air was sucked through the others by a pump and dispersed outside. With this arrangement a continuous flow from all measurement points was obtained, see figure 8.

Examples of the results obtained can be found in tables 1 and 2. These tables present the recorded tracer gas concentrations at different levels in the room at the commencement of steady state conditions. The tables are exemplified by a dilution curve in figure 9.

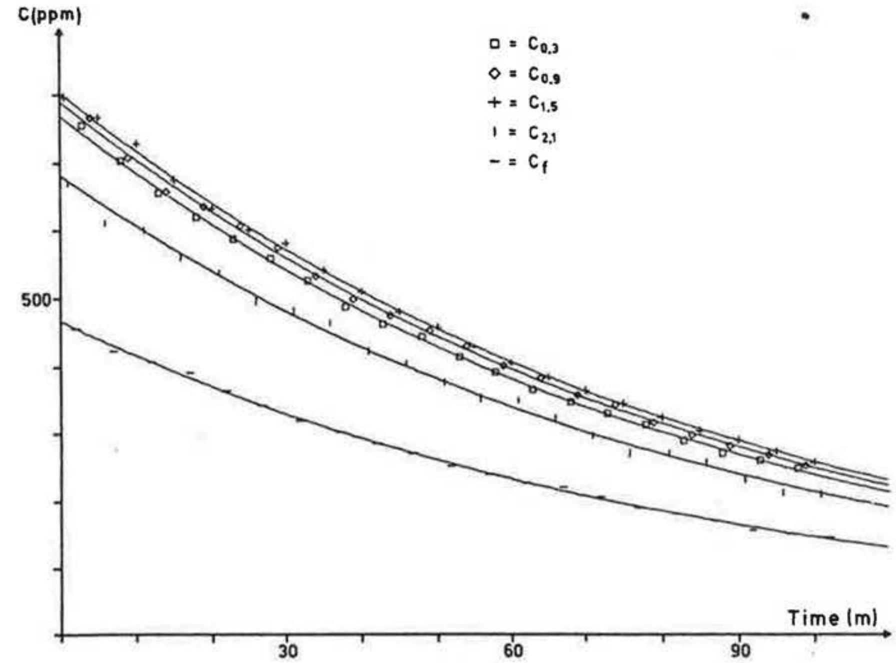


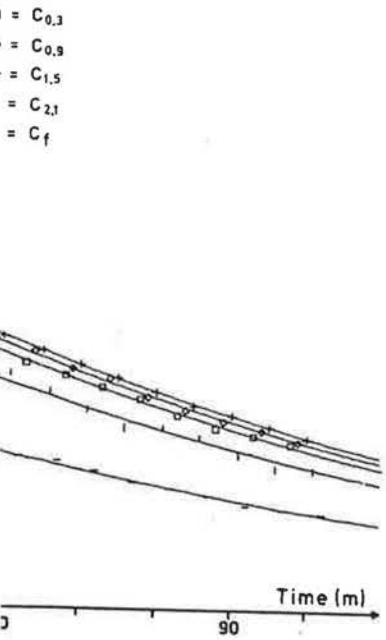
FIGURE 9. Dilution process at different levels in the room during steady state conditions.

Table 1. Measurement method
in ppm at different
state conditions. S

Test nr	Nominal air change rate n_0	$C_{0.3}$
001	1	738
002	1	756
003	1	770
004	1	769
005	2	721
006	2	671
007	2	735
008	2	739
009	3	832
010	3	796
011	3	718
012	3	777
013	4	818
014	4	836
015	4	654
016	4	846
017	1	780

Table 2. Measurement method
in ppm at different
state conditions. S
10 mm.

Test nr.	Nominal air change rate n_0 h^{-1}	$C_{0.3}$
019	4	747
020	4	781
021	4	773
022	4	755
023	3	788
024	3	768



fferent levels in the room
nditions.

Table 1. Measurement method A, primary values. Concentrations in ppm at different levels during initial steady state conditions. Slot width of supply air terminal 24 mm.

Test nr	Nominal air change rate n_0	$c_{0.3}$	$c_{0.9}$	$c_{1.5}$	$c_{2.1}$	c_f	Supply air overtemp. °C	Notes
001	1	738	543	546	325	288	0	Unstable flow process
002	1	756	797	819	778	622	2	
003	1	770	791	804	681	466	4	See fig. 8
004	1	769	796	811	695	459	8	
005	2	721	804	875	619	413	8	See fig. 9
006	2	671	752	821	533	395	4	
007	2	735	735	829	680	525	2	
008	2	739	736	790	701	630	0	
009	3	832	802	796	778	712	0	See fig. 10
010	3	796	903	866	646	527	2	
011	3	718	792	818	405	327	4	
012	3	777	836	873	400	318	8	
013	4	818	883	903	460	387	8	
014	4	836	928	682	474	401	4	
015	4	654	784	647	544	478	2	See fig. 11
016	4	846	848	851	865	789	0	
017	1	780	556	467	414	465	0	Same test data as 001

Table 2. Measurement method A, primary values. Concentrations in ppm at different levels during initial steady state conditions. Slot width of supply air terminal 10 mm.

Test nr.	Nominal air change rate n_0 h^{-1}	$c_{0.3}$	$c_{0.9}$	$c_{1.5}$	$c_{2.1}$	c_f	Supply air overtemp. °C
019	4	747	883	647	564	483	8
020	4	781	699	667	657	594	4
021	4	773	766	764	762	764	2
022	4	755	764	765	773	727	0
023	3	788	787	786	785	784	0
024	3	768	812	703	673	606	2

On the basis of this material the relative ventilation efficiency ε_i^r, according to equation (1), can now be calculated. These calculations are presented in tables 3 - 4.

Table 3. Processed primary values showing the relative ventilation efficiency ε_i^r according to equation (2)

Test no.	Nominal air change rate n_0 h^{-1}	Relative ventilation efficiency %				Supply air overtemp. ^oC
		$\varepsilon_{0.3}^r$	$\varepsilon_{0.9}^r$	$\varepsilon_{1.5}^r$	$\varepsilon_{2.1}^r$	
001	1	39.0	53.0	52.7	88.6	0
002	1	82.3	78.0	75.9	79.9	2
003	1	60.5	58.9	58.0	68.4	4
004	1	59.7	57.7	56.6	66.0	8
005	2	57.3	51.4	47.2	66.7	8
006	2	58.0	52.5	48.1	74.1	4
007	2	71.4	71.4	63.3	77.2	2
008	2	85.3	85.6	79.7	89.9	0
009	3	85.6	88.8	89.4	91.5	0
010	3	66.2	58.4	60.9	81.6	2
011	3	45.5	41.3	40.0	80.7	4
012	3	40.9	38.0	36.4	79.5	8
013	4	47.3	43.8	42.9	84.1	8
014	4	48.0	43.2	58.8	84.6	4
015	4	73.1	61.0	73.9	78.9	2
016	4	93.3	93.0	92.7	91.2	0
017	1	59.6	83.6	99.6	112.3	0

Table 4. Processed primary values showing the relative ventilation efficiency ε_i^r according to equation (2)

Test no.	Nominal air change rate n_0 h^{-1}	Relative ventilation efficiency %				Supply air overtemp. ^oC
		$\varepsilon_{0.3}^r$	$\varepsilon_{0.9}^r$	$\varepsilon_{1.5}^r$	$\varepsilon_{2.1}^r$	
019	4	64.7	54.7	74.7	85.6	8
020	4	76.1	85.0	89.1	90.4	4
021	4	98.9	99.7	100.0	100.3	2
022	4	96.3	95.2	95.0	94.0	0
023	3	99.5	99.6	99.7	99.9	0
024	3	78.9	74.6	86.2	90.0	2

Figures 10 - 15 pro
tive ventilation ef
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of supply air and at
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One conclusion to be
is that the savings
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uation (1), can now be calcula-
esented in tables 3 - 4.

lues showing the relative ven-
ε^r_i according to equation (2)

tilation efficiency %		Supply air overtemp. °C
$\varepsilon^r_{1.5}$	$\varepsilon^r_{2.1}$	
52.7	88.6	0
75.9	79.9	2
58.0	68.4	4
56.6	66.0	8
47.2	66.7	8
48.1	74.1	4
63.3	77.2	2
79.7	89.9	0
89.4	91.5	0
60.9	81.6	2
40.0	80.7	4
36.4	79.5	8
42.9	84.1	8
58.8	84.6	4
73.9	78.9	2
92.7	91.2	0
99.6	112.3	0

ues showing the relative ven-
^r_i according to equation (2)

ilation efficiency %		Supply air overtemp. °C
$\varepsilon^r_{1.5}$	$\varepsilon^r_{2.1}$	
74.7	85.6	8
89.1	90.4	4
100.0	100.3	2
95.0	94.0	0
99.7	99.9	0
86.2	90.0	2

Figures 10 - 15 provide a more overall view of how the relative ventilation efficiency varies with the nominal air change rates, with different values for the overtemperature of supply air and at different levels above the floor (0.3, 0.9 and 1.5 m).

These initial results show a heavy reduction in the relative ventilation efficiency with an increase in the supply air temperature. The nominal air change, n_0, also considerably affects ventilation efficiency. Thus in the case tested, the relative ventilation efficiency at level 0.9 m decreased from 93% under isothermal conditions to 43% when the temperature of air entering the room exceeded the room's temperature by 4°C and when the nominal air change rate was 4 times per hour. One way of reducing the effect of the increased temperature of the room in this particular case, is to reduce the slot opening of the supply air terminal. The relative ventilation efficiency then drops from 95% during isothermic conditions to 85% at an overtemperature of 4°C (slot opening 10 mm). By adjusting the supply air terminal we have therefore, in this particular case, been able to improve ventilation efficiency by a factor of 2, i.e. the air change rate at this level (0.9 m) has doubled in comparison to what it was previously.

These conditions can, of course, greatly affect operational efficiency. Sometimes, in order to increase ventilation efficiency, it has been felt necessary to increase the supply air flow (the nominal air change rate). In addition to raising operational costs this can also result, at least in some cases, in the reduction of the relative ventilation efficiency as is shown in figures 14 - 15.

One conclusion to be drawn from these preliminary measurements is that the savings potential is probably quite large. By using a well-dimensioned ventilation installation it ought to be possible to increase ventilation efficiency and reduce ventilation air flows in relation to those currently in use.

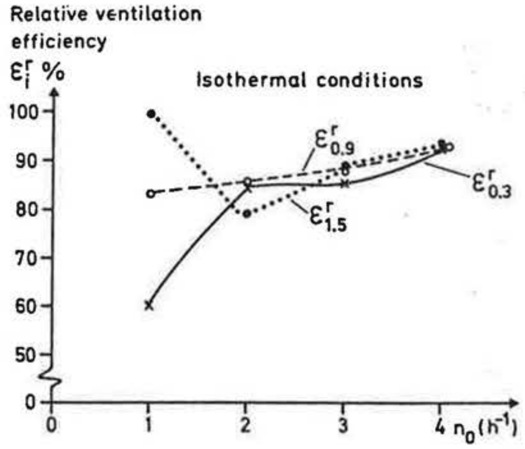


FIGURE 10. The relative ventilation efficiency as a function of the nominal air change rate (n_0). Supply air terminal's slot opening 24 mm.

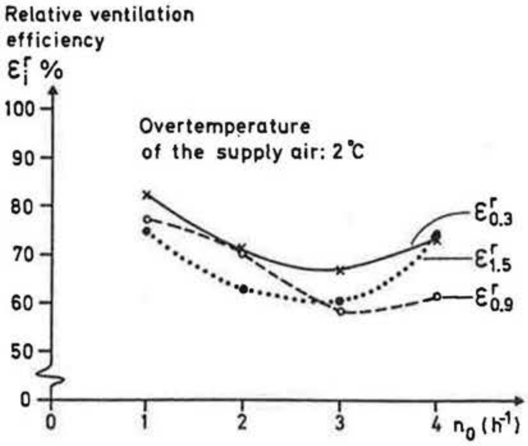
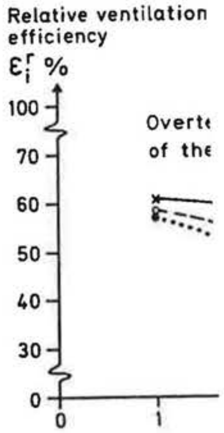


FIGURE 11. The relative ventilation efficiency as a function of the nominal air change rate (n_0). Supply air terminal's slot opening 24 mm.



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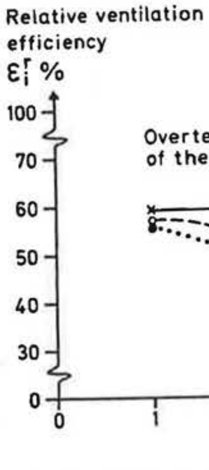

FIGURE 13. The relative
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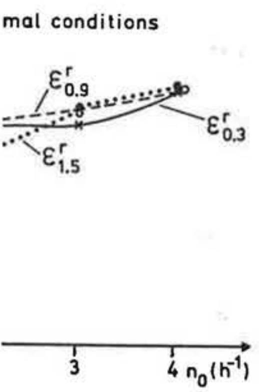

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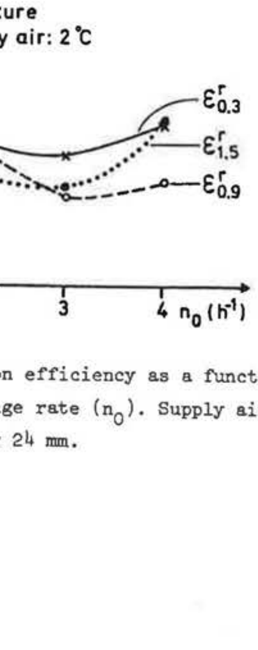

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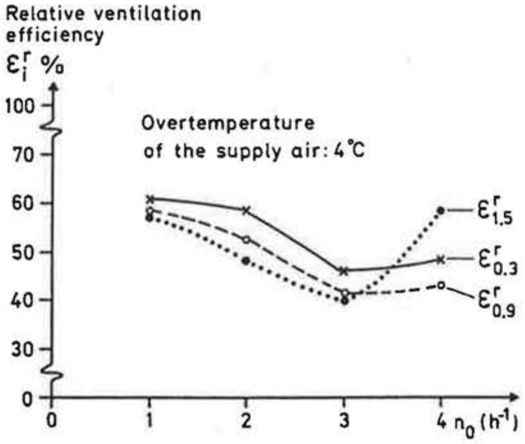

FIGURE 12. The relative ventilation efficiency as a function of the nominal air change rate (n_0). Supply air terminal's slot opening 24 mm.

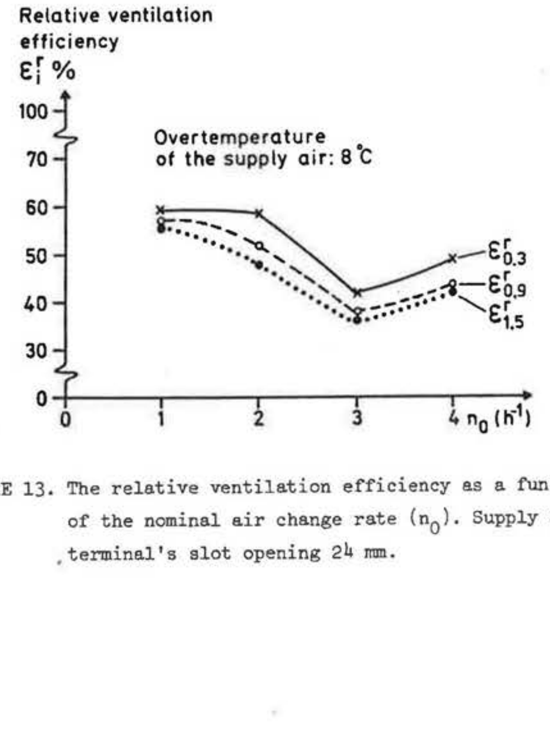

FIGURE 13. The relative ventilation efficiency as a function of the nominal air change rate (n_0). Supply air terminal's slot opening 24 mm.

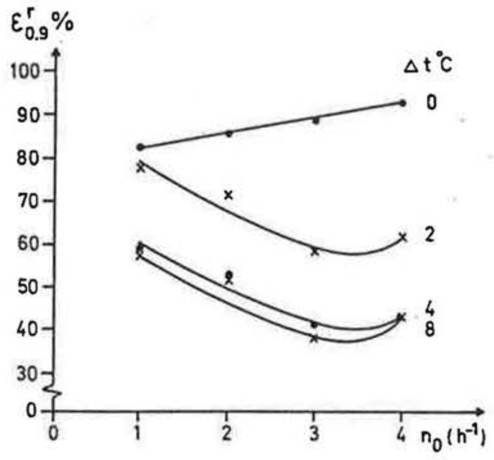

FIGURE 14. The relative ventilation efficiency at level 1.5 m above the floor as a function of the nominal air change rate (n_0) in the room and the overtemperature of the supply air (Δt).

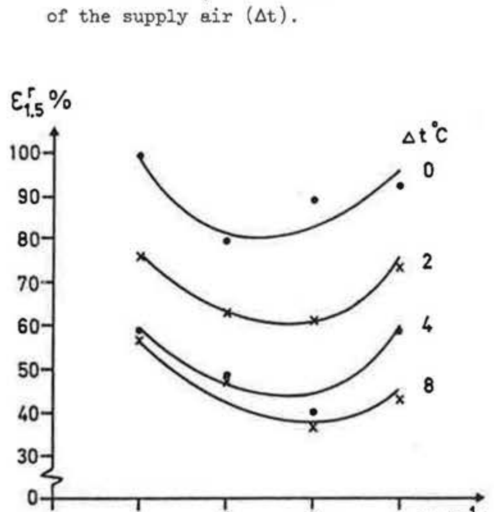

FIGURE 15. The relative ventilation efficiency at level 0.9 m above the floor as a function of the nominal air change rate (n_0) in the room and the overtemperature of the supply air (Δt).

SUMMARY

Results from an ongoing
Institute for Building
efficiency", will provi
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Determining the variati
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It is planned that the
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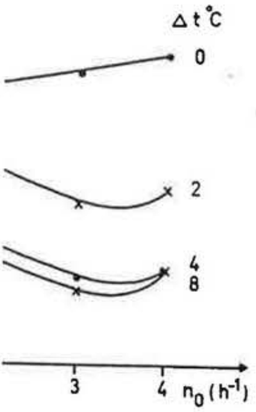

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a function of the nominal air
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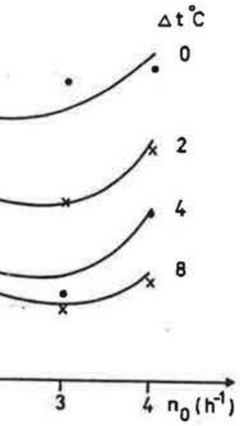

ation efficiency at level 0.9 m
a function of the nominal air
the room and the overtemperature
Δt).

SUMMARY

Results from an ongoing project at the National Swedish Institute for Building Research, and entitled "Ventilation efficiency", will provide information about previously little known conditions concerning the distribution of ventilation air in a room.

Determining the variations in the air change rates occuring in a room by measurements of air movements, can only be carried out with the aid of complicated calculations. From a practical viewpoint the use of tracer gas is a more attractive method. However, the article points out that the interpretation of concentration curves demands a certain amount of care. The slope of the curves cannot generally be taken as a direct measurement of the air change rate.

The findings presented indicate that, in some cases, only a small proportion of the ventilation air flow is utilized in the occupied zone. This naturally has a considerable impact on operational economy. Because of low ventilation efficiency, it is sometimes necessary to increase the ventilation air flow in order to raise the air change rate in the occupied zone. Operating costs can also be reduced be designing the ventilation system to achieve the maximum possible ventilation efficiency. Ventilation air flows can then be reduced, in relation to those currently in use, without air quality being impaired.

It is planned that the project will continue during the next two years, and during this period interim reports will be published.

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Nomenclature

a_{11}, a_{21} = the components in the eigenvector to λ_1

a_{12}, a_{22} = the components in the eigenvector to λ_2

b = ratio between concentrations

C = concentration of pollution

C(0) = the initial concentration

C(∞) = the steady state concentration

C^{tr} = the transient concentration

K_1, K_2 = constants

$\dot{m}$ = the source of pollution

n_0 = nominal air change rate $(= \frac{Q}{V_r})$

Q = the ventilation air flow

t = time

T_0 = specified time interval

V = the volume of each zone

V_r = the room's total volume (= 2V)

V_1 = the active volume in zone 1 $(V_1 \leq V)$

V_2 = the active volume in zone 2 $(V_2 \leq V)$

Greek symbols

β =

γ =

Δt =

ε^a =

ε^r =

λ_1, λ_2, =

Suffixes

i =

f =

t =

$\left.\begin{matrix}0.3\\0.9\\1.5\\2.1\end{matrix}\right\}$ =

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e components in the eigenvector to λ_1
e components in the eigenvector to λ_2
io between concentrations
centration of pollution
initial concentration
steady state concentration
transient concentration
stants
source of pollution
inal air change rate (= $\frac{Q}{V_r}$)
ventilation air flow
e
cified time interval
volume of each zone
room's total volume (= 2V)
active volume in zone 1 ($V_1 \leq V$)
active volume in zone 2 ($V_2 \leq V$)

Greek symbols

β = linking (coupling) factor
γ = admixture factor
Δt = overtemperature (°C)
ε^a = the absolute ventilation efficiency
ε^r = the relative ventilation efficiency
λ_1, λ_2, = eigenvalues

Suffixes

i = occupied zone
f = exhaust air
t = supply air
0.3, 0.9, 1.5, 2.1 = measuring level