Summary The *local* ventilation efficiency of a mechanical ventilation system may in general terms be defined as 'providing air in those parts of a room where it is required'. In this paper different definitions of the local efficiency and methods for measuring it are discussed. Finally, results from measurements of the ventilation efficiency are presented. These indicate that, in some cases, only a minor proportion of the ventilation air flow is utilised in the occupied zone. Furthermore it is demonstrated that it is normally not advisable to use the slope of the tracer gas curves in a linear logarithmic plot as a measure of the 'local ventilation rate'. The slopes will normally, after a certain period of time has elapsed, become equal. Then they reflect only the overall ventilation rate and not local differences.

Measurements of ventilation efficiency by using the tracer gas technique

M. SANDBERG and A. SVENSSON

List of symbols

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^r	=	the relative ventilation efficiency index per cent	
C	=	concentration of pollution	ppm
C(O)	=	the initial concentration	ppm
$C(\infty)$	=	the steady state concentration	ppm
C^{tr}	=	the transient concentration	ppm
K_1, K_2	=	constants	
$\dot{m}$	=	the source of pollution	m²/sec.
n_0	=	nominal ir change rate($= \frac{Q}{V_r}$)	h^{-1}
Q	=	the ventilation air flow	
t	=	time	sec, min, h.
T_0	=	specified time interval	sec.
V	=	the volume of each zone	m^3
V_r	=	the room's total volume (=2V)	m^3

Greek symbols

β	=	nondimensional coupling factor	
γ	=	nondimensional mixing factor	
Δt	=	over temperature	°C
η_a	=	the absolute ventilation efficiency at steady state	per cent
η_r	=	the relative ventilation efficiency at steady state	per cent
λ_1, λ_2,	=	eigenvalues	h

Suffixes

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

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1 Introduction

When premises are to be ventilated, two problems are often important. These are:
(1) creating a sufficient ventilation flow with regard to cooling or heating requirements or the amount of pollution, and
(2) side-effects resulting from flow generating locally high air velocities.
From the point of view of energy and comfort it is of the greatest importance that ventilation air is available in those places in a room where it is required. In this respect different types of ventilation systems have different efficiencies (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 'the local 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. The project aims are to:

(1) 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

(2) 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:

(1) the type and positioning of supply and exhaust air terminals,

(2) the supply air temperature and flow,

(3) the shape and size of the room.

This paper discusses the various definitions of ventilation efficiency, 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.

2 Definitions of ventilation efficiency

The definition used expresses the ability of the system to evacuate pollution from a particular source in a room. In this context pollution is interpreted in general terms and therefore includes e.g. odours, radon and over temperature (excessive heat causing temperatures higher than required). For reasons of convenience it is assumed in the following that pollution is something that can be expressed in units of ppm (parts per million). The definition of ventilation should meet two important criteria:

(1) it must be directly linked to characteristics of the system described,

(2) it must be operational i.e. it must indicate how efficiency is to be measured.

The definition of ventilation efficiency can be based on two characteristics of a ventilation system:

(a) the *relative ventilation efficiency,* which expresses how the ventilation capability of the system varies between different places in a room, and

(b) the *absolute ventilation efficiency,* which expresses the ability of the ventilation system to reduce a pollution concentration in relation to the feasible theoretical maximum reduction.

Fig. 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 independent of time and equal everywhere in the whole room.

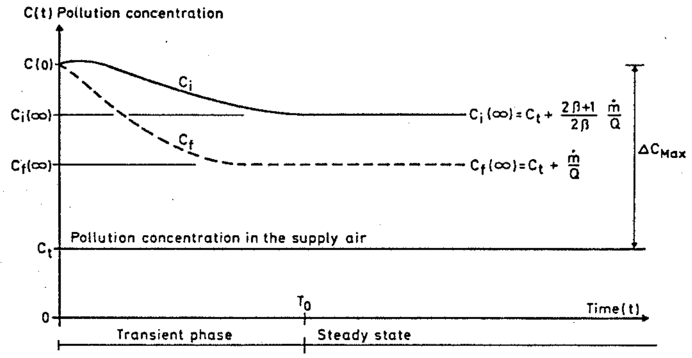

Fig. 1. Pollution concentration as a function of time.

Using values of the steady state condition the *relative ventilation efficiency* (n_{ri}) can be defined as

$$\eta_{ri} = \frac{C_f - C_t}{C_i - C_t} = 1 + \frac{C_f - C_i}{C_i - C_t} \qquad (1)$$

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

$$\eta_{ri} = \frac{C_f}{C_i} \qquad (2)$$

When $C_i = C_f$ then $\eta_{ri} = 1$.

The relative ventilation efficiency is always positive and can be greater than 1.

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

The *absolute ventilation efficiency* (n_{ai}) is defined as:

$$\eta_{ai} \frac{C(0) - C_i}{\Delta C_{max}} = \frac{C(0) - C_i}{C(0) - C_t} \qquad (3)$$

When
$C_i = C_t$ then $\eta_{ai} = 1$

The absolute ventilation efficiency is always less than 1 and can be negative.

The above definitions are primarily a measurement of the efficiency of the system over a long-term period and are not necessarily a good measurement of its ability to remove 'transients'.

2.1 Determining the ventilation efficiency

Both the relative and the absolute ventilation efficiency can be determined primarily in two ways:

(1) *The direct method.*

An experimental arrangement is used to imitate the actual conditions under which the ventilation system is to operate. The steady state 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.

(2) *The indirect method.*

This involves attempts to identify quantifiable 'system parameters' which characterize the system's ability to evacuate pollution. The system parameters are normally quantities measured at non-stationary conditions (transient analysis). These parameters can be used to calculate or estimate ventilation efficiency defined at stationary conditions in definitions (1) to (3).

One common system parameter, which many attempt to introduce, (see e.g. [4]), is a so-called local air exchange rate defined from the slope of tracer gas curves in a linear-logarithmic plot. This is at a first glance a natural definition because, when the mixing is complete in the whole room, the slope in a linear logarithmic plot is equal to the *nominal air-exchange rate* $n_o = \frac{Q}{V}$, when Q is the amount of outdoor air ('fresh' air) supplied to the room and V is the total volume of the room. But when the mixing is incomplete then there is no such simple physical interpretation of the slope of the curves in a linear logarithmic plot. This will be demonstrated in the example below, where it is shown that generally no unequivocal local air change rate can be defined from the slope.

The problem of defining a 'local air change rate' (ventilation rate) from the slope 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 mixing, that an unequivocal local air change rate can be defined from the slope.

3 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. Fig. 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.

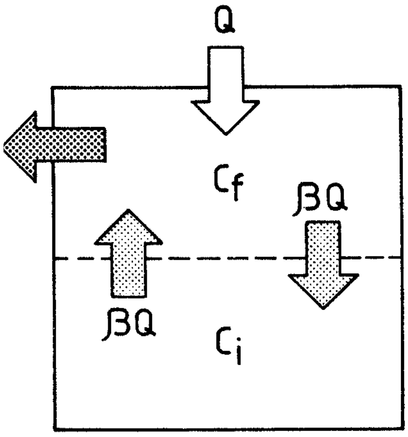

Fig. 2. The room in the model case.

For each volume the mixing is assumed complete. The mixing between the two volumes is expressed by a parameter ($\beta = 0$) to complete mixing ($\beta \rightarrow \infty$). This parameter is a pure mathematical concept and the 'flow' βQ is not a physically measureable quantity, the parameter β is merely a measure of the degree of mixing or coupling between the zones.

On the assumption that the pollutant concentration in the supply air, C_t, is zero, the mass balance gives the following differential equation for the concentrations C_f and C_j in the exhaust air zones respectively the occupied zone:

$$V\frac{dC_f}{dt} = -(1+\beta)\,Q \cdot C_f + \beta Q\, C_j + \frac{\dot{m}}{2}$$
$$V\frac{dC_j}{dt} = \beta Q\, C_f - \beta Q\, C_j + \frac{\dot{m}}{2} \tag{4}$$

when $\dot{m}$ represents the rate of pollution generation from a homogeneous pollution source, that fills up the whole room. The solution of the mass balance equations gives the following expressions for the concentrations as a function of time, t.

$$C_f(t) = C_f^{tr}(t) + \frac{\dot{m}}{Q}$$
$$C_i(t) = C_i^{tr}(t) + \dot{m}/\left(\frac{2\beta}{2\beta+1}Q\right) \tag{5}$$

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

$$C_f^{tr}(t) \rightarrow 0 \text{ which means that } C_f(t) \rightarrow \frac{\dot{m}}{Q} \tag{6}$$

and

$$C_i^{tr}(t) \rightarrow 0 \text{ which means that } C_i(t) \rightarrow +\dot{m}/\left(\frac{2\beta}{2\beta+1}Q\right)$$

The 'flow' $\frac{2\beta}{2\beta+1}Q$ can be interpreted as the amount of fresh air drawn into the occupied zone over 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 Fig. 3, then the steady state concentration will be the same as (6).

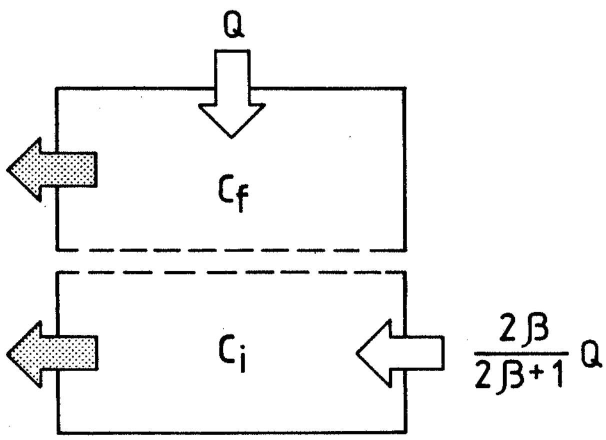

Fig. 3. Equivalent model at steady state.

The transient solutions of Equation (4) can be expressed as follows (see e.g. any textbook on ordinary differential equations):

$$C_f^{tr}(t) = K_1\, a_{11}\, e^{-\lambda_1 t} + K_2\, a_{12}\, e^{-\lambda_2 t}$$
$$C_i^{tr}(t) = K_1\, a_{21}\, e^{-\lambda_1 t} + K_2\, a_{22}\, e^{-\lambda_2 t} \tag{7}$$

Where

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

$\lambda_1(\frac{Q}{V},\beta)$, $\lambda_2(\frac{Q}{V},\beta)$ = Eigenvalues

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

$a_{12}(\beta)$, $a_{22}(\beta)$ = The components in the eigenvector to λ_2

The eigenvalues are equal to

$$\lambda_{1,2} = n_0\{-(2\beta+1) \pm \sqrt{(1+4\beta^2)}\} \tag{8}$$

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

Note that both the eigenvalues and the eigenvectors are functions of the coupling factor β.

Form the above expression for $\lambda_1$and λ_2 it appears that

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

This means that when a sufficient period of time has elapsed, say $t > T_o$, the other terms in (7) can be neglected, and we then obtain for $t > T_0$:

$$C_f^{tr}(t) \sim K_1\, a_{11} e^{-\lambda_1 t}$$
$$C_i^{tr}(t) \sim K_1\, a_{21} e^{-\lambda_1 t} \tag{9}$$

i.e.

$$\frac{C_f^{tr}(t)}{C_i^{tr}(t)} \sim \frac{a_{11}}{a_{12}} \equiv b \tag{10}$$

The ratio b is constant and equal to

$$b = \frac{2\beta}{1+\sqrt{(1+4\beta^2)}} \tag{11}$$

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

The eigenvalue λ_1 i.e. the slope in a linear-logarithmic diagram is then equal to [3].

$$\lambda_1 = n_0 \frac{C_f^{tr}}{C^{tr}} \tag{12}$$

where C^{tr} is the mean concentration in the room.

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

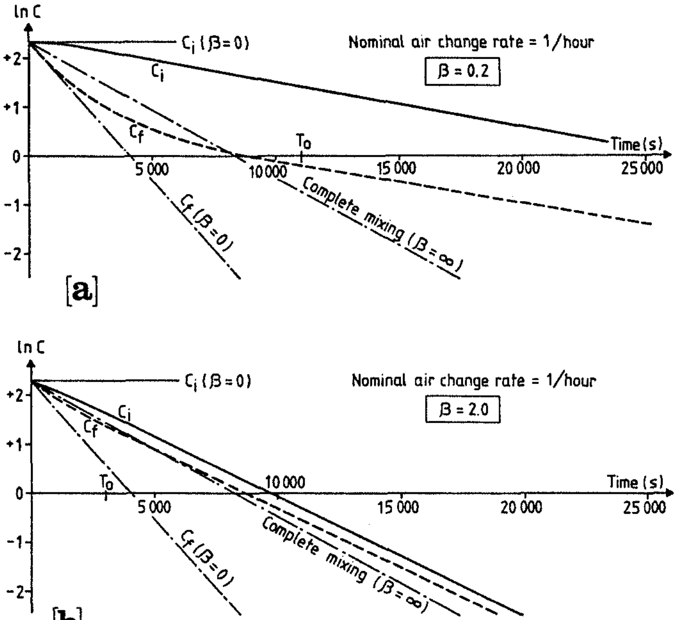

Fig. 4. The transient solution as a function of time for different β. (coupling factor)

(1) For all $0 < \beta \leq \infty$, i.e. independent of what 'air change rates' has been taken place in different parts of a room, the slope of the curves will vary until they are parallel in a linear-logarithmic plot, see Fig. 4.

(2) When the curves become parallel the slope is determined by the over-all ventilation rate.

(3) A certain minimum period of time, T_0, is required before the local differences in 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 'local air change rate' at the point in question.

On the other hand, in the case of complete mixing ($\beta \rightarrow \infty$) then

$$\beta \rightarrow \infty \Rightarrow \begin{cases} -\lambda_1 \doteq -\dfrac{Q}{V_r} \\ -\lambda_2 \rightarrow -\infty \\ b \rightarrow 1 \end{cases}$$

The whole 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.

One way of defining a local 'air change rate' for the whole process would be to fit, to the measured transient concentration, an expression of the type

$$C(t) \approx C(0)e^{-\gamma n_0 t} \tag{13}$$

The parameter γ is commonly referred to as the mixing factor[1]. It is not advisable, however to use an expression such as (13) to define the 'local air change rate' at different points. The reason for this is twofold.

(1) The mixing factor γ is dependent on the measuring time. When the measuring time approaches "infinity", the parameter γ approaches $\frac{C_f^{tr}}{C^{tr}}$, see Equation (12), in all points in the room.

(2) Local differences will be underestimated.

The stationary concentration values according to Equation (6) inserted in the definition of relative ventilation efficiency gives:

$$\eta_{ri} = \frac{C_f}{Ci} = \frac{2\beta}{2\beta + 1} = \frac{1}{1 + \frac{1}{2\beta}} \tag{14}$$

and the absolute ventilation efficiency in the occupied zone will be

$$\eta_{ai} = \frac{C(0) - C_i}{C(0)} = 1 - \frac{C_i}{C(0)} = 1 - \frac{\dot{m}}{Q \cdot C(0)} \left(\frac{2\beta + 1}{2\beta}\right) \tag{15}$$

In the case of complete mixing ($\beta \rightarrow \infty$), (14) and (15) be

$$\eta_{ri} = 1 \text{ and } \eta_{ai} = 1 - \frac{\dot{m}}{Q \cdot C(0)}$$

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

4 Methods for the determination of ventilation efficiency

In the discussion of the advantages and drawbacks of the different methods of measurement, the model case described in the foregoing sections is used here 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.

Two methods will be discussed in greater detail.

4.1 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, i.e. C_t = constant.

The growth of the tracer gas concentration is recorded at different points.

4.3 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

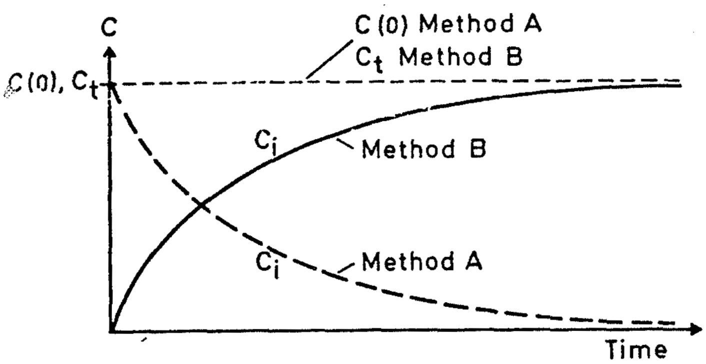

Fig. 5. The relationship between method A and B.

means that only the constants K_1 and K_2 are dependent on the method used. See Fig. 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.

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

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

Method B

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

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

$$b^r = \frac{2\beta}{1 + \sqrt{(1 + 4\beta^2)}} \text{ and } \eta_{ri} = \frac{2\beta}{2\beta + 1}$$

for small β the ratio b will be: $b^r \approx \beta < \eta_{ri}$

for large β the ratio b will be: $b^r \approx \dfrac{2\beta}{1 + 2\beta} = \eta_{ri}$

This means that for small β the fixed ratio b gives an *underestimate,* (see Fig. 6), of the relative ventilation efficiency in the operational case presented in Fig. 1. However, ratio b^r is proportional to ventilating efficiency, as defined previously, and is in other words *an index of ventilation efficiency.* It is shown[3] that generally the relative ventilation efficiency is overestimated or underestimated depending on the type of the ventilation system and the location of the pollution source. The only exceptional case is in the ideal case of complete mixing. Other measures of the ventilation efficiency, as e.g. the area under the tracer gas curves, are also given[3] which are thoroughly discussed both from a theoretical and practical point of view.

5 Measurements

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

The exhaust air terminal was placed above the door and 0.2 m beneath the ceiling. The slot height of the supply

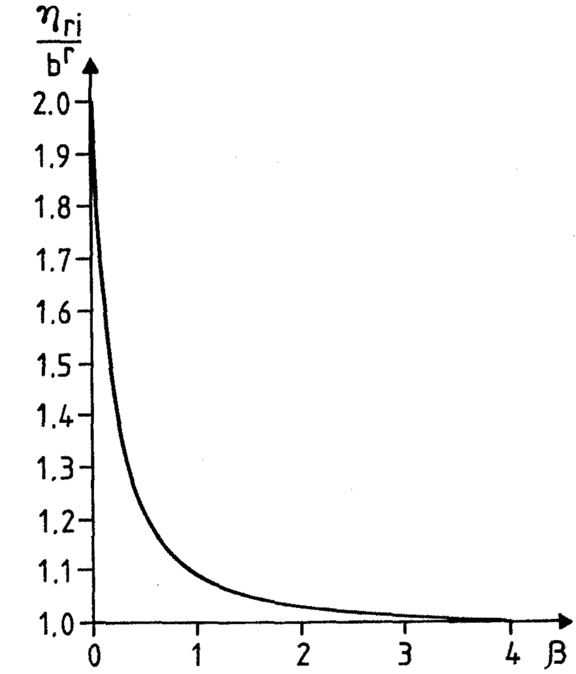

Fig. 6. The ratio $\frac{\eta_{ri}}{b^r}$ as a function of β.

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 Fig. 7.

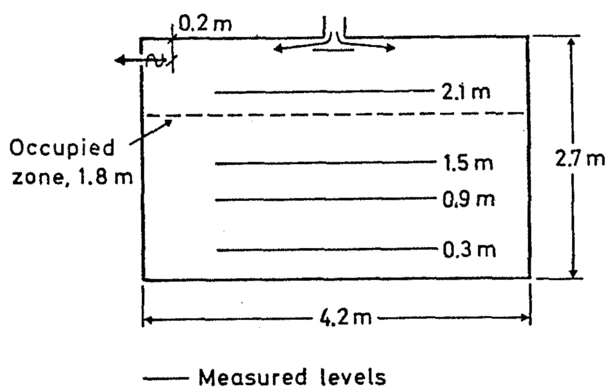

Fig. 7. Sketch of test room.

The supply air flow into the test room was varied to correspond to 1, 2, 3 or 4 air changes per hour. 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 nitrous oxide (N_2O). Measurements of the tracer gas concentrations at these levels and in the exhaust air duct were made using consecutively the electrically-operated 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 analyser. Air was drawn in through the others by a pump and dispersed outside. With this arrangement a continuous flow from all measurement points was obtained, see Fig. 8.

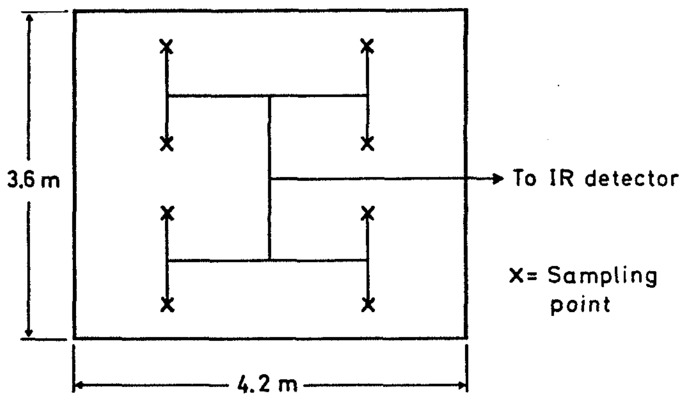

Fig. 8. Visualisation of the measurement arrangements at different levels above the floor (0.3, 0.9, 1.5 and 2.1m).

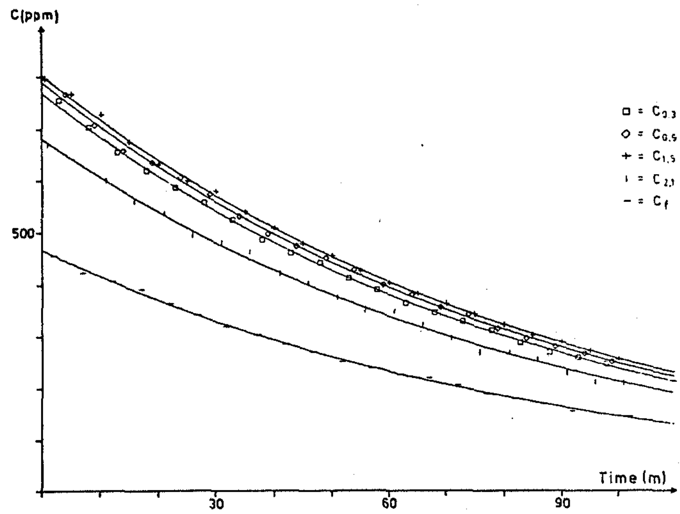

Fig. 9. Dilution process at different levels in the room during steady state conditions.

Examples of the results obtained by using the decay method 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 constant ratios between the concentrations. The tables are exemplified by a dilution curve in Fig. 9.

Table 1. Measurement method A, primary values. Concentrations in ppm at different levels during initial constant ratio between concentrations. Slot width of supply air terminal 24mm.

Test no	Nominal air change rate n_0 h^{-1}	$C_{0.3}$	$C_{0.9}$	$C_{1.5}$	$C_{2.1}$	C_f	Supply air over-temperature °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 constant ratio between concentrations. Slot width of supply air terminal 10mm.

Test no	Nominal air change rate n_0 h^{-1}	$C_{0.3}$	$C_{0.9}$	$C_{1.5}$	$C_{2.1}$	C_f	Supply air over-temperature °C	Notes
0.9	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 η_{ri}, according to Equation (1), can now be assessed by using the constant ratio between the concentration C_f in the exhaust air terminal and in the concentration C_i at the actual level. These calculations are presented in Tables 3 and 4. Figs. 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).

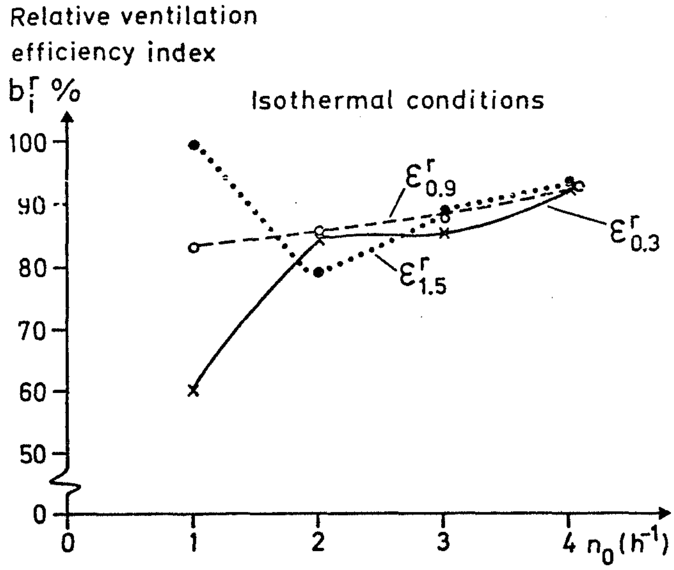

Fig. 10. The relative efficiency as a function of the nominal air change rate (n_0). Slot opening of supply air terminal, 24mm.

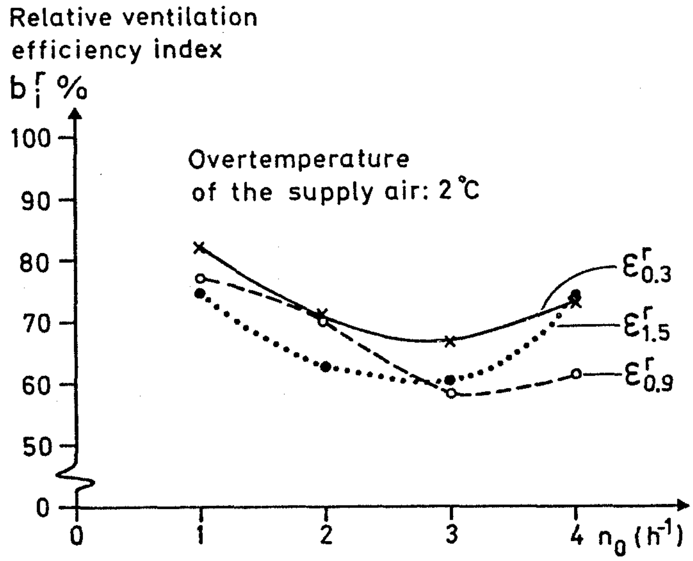

Fig. 11. The relative ventilation efficiency index b^r as a function of the nominal air change rate (n_0). Slot opening of supply air terminal, 24mm.

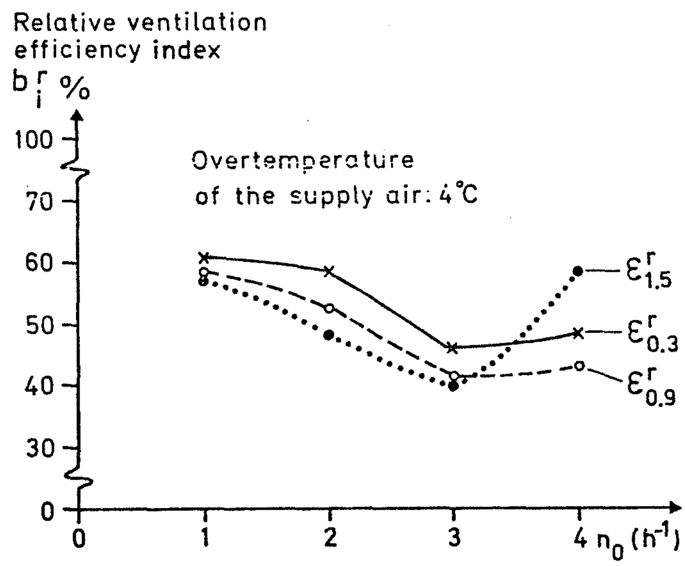

Fig. 12. The relative ventilation efficiency index b^r as a function of the nominal air change rate (n_0). Slot opening, 24mm.

Table 3. Processed primary values showing the relative ventilation efficiency index b_i^r.

Test no	Nominal air change rate n_0 h^{-1}	Relative ventilation efficiency per cent $b_{0.3}^r$	$b_{0.9}^r$	$b_{1.5}^r$	$b_{2.1}^r$	Supply air over-temperature °C
001		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 index b_i^r.

Test no	Nominal air change rate n_0 h^{-1}	Relative ventilation efficiency per cent $b_{0.3}^r$	$b_{0.9}^r$	$b_{1.5}^r$	$b_{2.1}^r$	Supply air over-temperature °C
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

These initial results show a heavy reduction in the relative ventilation efficiency with an increase in the supply air temperature. The nominal air change rate, n_0, also considerably affects the ventilation efficiency. Thus in the case tested, the relative ventilation efficiency at level 0.9 m decreased from 93 per cent under isothermal conditions to 43 per cent when the temperature of air entering the room exceeded the temperature required in the room 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 per cent during isothermal conditions to 85 per cent at an overtemperature of 4°C (slot opening 10 mm). By adjusting the supply air terminal in this particular case, it is therefore possible to improve ventilation efficiency by a factor of 2.

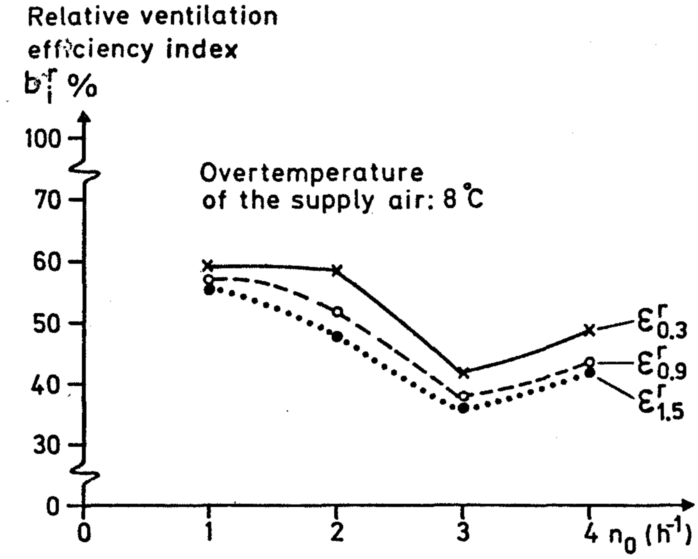

Fig. 13. The relative ventilation efficiency index b^r as a function of the nominal air change rate (n_0). Slot opening, 24mm.

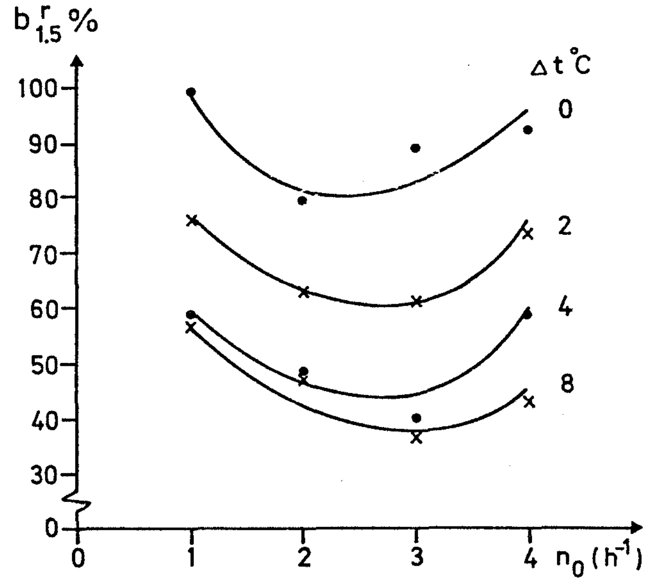

Fig. 14. The relative ventilation efficiency index b^r at level 1.5m above the floor as a function of the nominal air change rate (n_0) in the room and the over temperature of the supply air (Δt).

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 Figs. 14 and 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.

6 Conclusions

Results from a project in progress at the National Swedish Institute for Building Research, and entitled "Ventilation efficiency", will provide information about

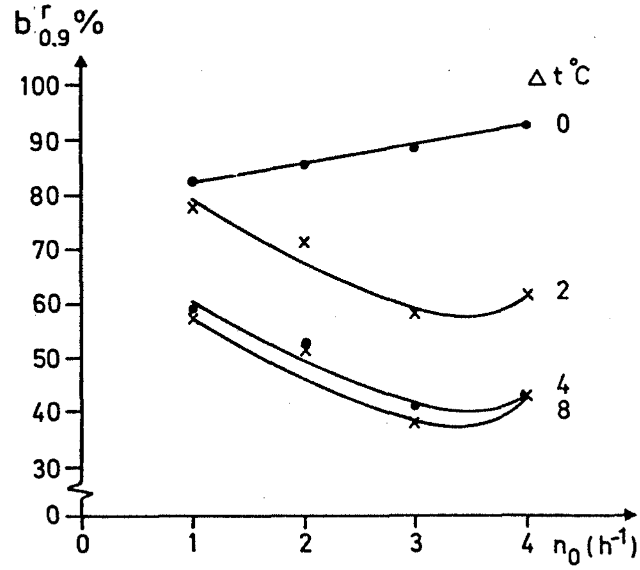

Fig. 15. The relative ventilation efficiency index b^r at level 0.9m above the floor as a function of the nominal air change rate (n_0) in the room and the over temperature of the supply air (Δt).

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 paper 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 local air change rate.

The findings presented indicate that, in some cases, only a small proportion of the ventilation air flow is utilised 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 local air change rate in the occupied zone. Operating costs can also be reduced by 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.

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