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EXTRACT

This report is an article that has been presented in Environment Industrial.

A two zone mixing model is used to describe the concept(s) of and to define the effectiveness of ventilation. Tests using this principle, with air supply in the upper zone and exhaust either in the same zone or in the other zone near the floor, are reviewed. Hot air supply and exhaust opening in the upper zone gives short-circuit with poor efficiency. With the exhaust near the floor the efficiency with regard to the air quality in the lower zone equals to complete mixing. These tests also predict promising result for a system with air supply in the lower zone and exhaust in the upper.

3 INDEXING TERMS: NORWEGIAN / ENGLISH

NORWEGIAN	ENGLISH
Ventilasjonseffektivitet	Ventilation Effectiveness
Lagdelte luftmasser	Air stratification
Kortslutning	Short-circuit

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PREFACE

This report is an article that was presented in Environment International, Vol. 8, 1982.

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VENTILATION EFFICIENCY

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Research indicates that ventilating systems can be designed for higher ventilation efficiency in the zone of occupation than systems designed for complete mixing. Expressions for ventilation efficiency are derived using a two-box theoretic model. These definitions of ventilation efficiency can be used for practical measurements, and also seem to be valid for multibox schemes. Measurements reviewed show that diagonal schemes are the most efficient. Short-circuiting schemes, with warm air supply along the ceiling and high wall exhaust, produce very low efficiencies. The mathematical model predicts high efficiencies using diffuse air supply directly to the zone of occupation, if the air is not used for heating.

Introduction

There is a tendency in Scandinavia, the USA, and many other countries to reduce ventilation in order to save energy and lower the cost for heating, ventilating and air conditioning. There are conflicting interests between hygiene and comfort on the one hand, and energy economy on the other hand. This situation has led to an investigation of ventilation in terms of efficiency. We believe that ventilation can be done more efficiently, indicating that decreased ventilation does not necessarily mean decreased air quality. In fact, if one looks at the zone of occupancy, it is possible to increase the ventilation air change rate in this zone without increasing the air change rate for the room in question.

The research project reviewed by this paper has the objective of establishing (1) expressions for ventilation efficiency; (2) methods for measuring ventilation efficiency; and (3) rules for achieving efficient ventilation.

Results

Expressions for ventilation efficiency

The objective of the ventilation process is to maintain a certain air quality in the zone of occupation, i.e., to control contamination levels according to set standards. Heat and cold can in this respect also be treated as contaminations. Contamination levels are expressed as concentrations. Concentrations of chemical substances are usually in ppm or mg/m^3; concentrations of heat, and cold are in kJ/m^3 (ϱCpT). Almost all ventilation processes are governed by turbulent diffusion of contaminations, i.e., contaminations are spread by the air currents in the room. The turbulent diffusion constants are roughly the same for both heat and respirable airborne chemicals.

The conservation equation for steady state is

$$(C_E - C_S)\dot{V} = \dot{Q}, \tag{1}$$

where $\dot{Q}$ = source production rate;
$\dot{V}$ = ventilation air flow rate;
C_E = steady-state concentration in the exhaust air;
C_S = steady-state concentration in the supply air.

The source production rate is the net rate. All local exhausts are subtracted. For heating/cooling $\dot{Q}$ is the net heating/cooling load handled by the general ventilation system. It is convenient to separate the zone of occupancy, or the working zone, from the rest of the room; accordingly, Eq. (1) is transformed introducing C_W, the concentration in the working zone:

$$[(C_E - C_W) + (C_W - C_S)]\dot{V} = \dot{Q}. \tag{2}$$

The part $(C_E - C_W)\dot{V}$ might now be regarded either as an additional local exhaust, or as an additional contamination source, depending on the sign. Required ventilation air flow rate is

$$\dot{V} = \dot{Q}/[(C_E - C_W) + (C_W - C_S)]. \tag{3}$$

A convenient reference process is complete mixing. C_E is then equal to C_W. Required ventilation air flow rate for complete mixing is

$$\dot{V}_{cm} = \dot{Q}/(C_W - C_S); \tag{4}$$

$\dot{Q}$, C_W, and C_S are equal and it is obvious that $\dot{V}_{cm}/\dot{V}$ could be called a steady-state ventilation efficiency. The notation ϵ_{II} is used for this efficiency:

$$\epsilon_{II} = \dot{V}_{cm}/\dot{V} = [(C_E - C_W) + (C_W - C_S)]/(C_W - C_S)$$
$$= 1 + (C_E - C_W)/(C_W - C_S). \qquad (5)$$

It is easy to see that this efficiency (depending on the system layout) might become greater than 1. A high wall/ceiling supply and exhaust ventilation scheme that is run as a warm-air heating system develops thermal stratification. Then $C_E > C_W$ and $C_S > C_W$. It is easy to see that ϵ_{II} then is less than 1, and accordingly the ventilation system is not efficient. However, it is not obvious from the efficiency expression to conclude that the system is inefficient for every type of chemical contamination. It depends on the location of the source of contamination and the density of the contamination compared to the density of the room air. If the chemical contamination is similar to the "thermal contamination," the efficiency is also equal. It is obvious that steady-state efficiency is strongly dependent on the characteristics of the source and the ventilation process, but, in general, systems with positive $(C_E - C_W)/(C_W - C_S)$ are favourable and should be looked for.

In actual situations, transient state is equally as important as steady state, and for this reason there is a need for defining a transient ventilation efficiency as well. Our experience has been that one often gets thermal stratification which behaves very much like a two-box model with complete mixing within each box. For this reason a two-box model is used to arrive at expressions for transient ventilation efficiency (Mathisen and Skåret, 1981). Others have also used this approach (Malmström and Östrøm, 1980; Sandberg, 1981).

The exchange of air between the boxes is influenced by the temperature and turbulence conditions and the location of the supply and exhaust terminals. If contamination is produced in a room with two or more zones of air currents, there is reason to believe that the concentrations will be unequally distributed between the zones, due to the unequal supply of clean air. If a room is ventilated by jets, air will be entrained into the jets.

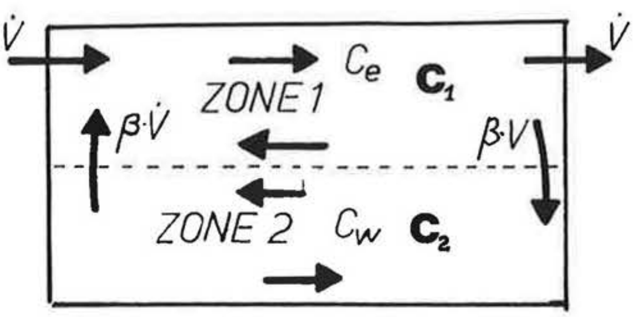

Fig. 1. Air currents in a confined room. The mixing between air zone is indicated by β.

After the jets have been dissolved, the air is transported back again and into the jets. In this way a circulating zone is established.

Two main factors determine whether the whole room is directly influenced by the air currents from the jets: (a) the size of the room, and (b) the temperature of the supply air. A high wall or ceiling supply and exhaust ventilation scheme may establish circulation zones as shown in Fig. 1. Assuming the room is filled with a tracer gas, the decay of the tracer gas is studied.

The differential equations for the decay will be

$$C_E\beta \cdot \dot{V} - C_W\beta\dot{V} = (1 - x) \cdot V \cdot \frac{dC_W}{dt} \longrightarrow \frac{dC_2}{dt}$$
$$= a_{21}C_1(t) + a_{22}C_2(t), \qquad (6)$$

$$C_W\beta \cdot \dot{V} - (1 + \beta)\, C_E \cdot V = x \cdot V\frac{dC_e}{dt} \longrightarrow \frac{dC_1}{dt}$$
$$= a_{11}C_1(t) + a_{12}C_2(t), \qquad (7)$$

where $C_1 = C_E$, the concentration in the exhaust air;
$C_2 = C_W$, the concentration in the working zone;
β = exchange factor indicating the mixing between the zones;
$\dot{V}$ = air flow rate;
$x = V_1/V$, the ratio between the volume of zone I and the complete volume of the room;
t = time.

The differential Eqs. (6) and (7) have a rather simple analytical solution. The general differential equations have a form

$$\frac{dy_1}{dt} = a_{11}y_1 + a_{12}y_2 \quad y_1 = C_E(t)$$

$$a_{11} = -\frac{1 + \beta}{x}n, \; a_{12} = \frac{\beta}{x}n,$$

$$\frac{dy_2}{dt} = a_{21}y_1 + a_{22}y_2 \quad y_2 = C_W(t)$$

$$a_{21} = \frac{\beta}{1 - x}n, \; a_{22} = -\frac{\beta}{1 - x}n.$$

These equations have a solution if

$$y_1 = k_1 e^{\lambda t},$$
$$y_2 = k_2 e^{\lambda t},$$
$$n = \text{nominal air change rate} = \frac{\dot{V}}{V}.$$

The analytical solution gives

$$\lambda = \frac{a_{11} + a_{22} \pm \sqrt{(a_{11} + a_{22})^2 - 4(a_{11} \cdot a_{22} - a_{12} \cdot a_{21})}}{2}.$$

This in turn yields two values for each k:

$$\frac{k_2}{k_1} = - \frac{a_{11} + a_{21} - \lambda}{a_{12} + a_{22} - \lambda}.$$

However, k_1 can be made equal to 1, and if the starting concentrations $C_E(t) = C_W(t) = C(0)$ for $t = 0$, a final solution is

$$C_E(t) = C(0) \frac{1 - {}^2k_2}{{}^1k_2 - {}^2k_2} e^{\lambda_1 \cdot t} \left[1 - \frac{1 - {}^1k_2}{1 - {}^2k_2} e^{(\lambda_2 - \lambda_1) \cdot t} \right], \tag{8}$$

$$C_W(t) = C(0) \frac{1 - {}^2k_2}{{}^1k_2 - {}^2k_2} \cdot {}^1k_2 \cdot e^{\lambda_1 \cdot t} \left[1 - \frac{1 - {}^1k_2}{1 - {}^2k_2} \cdot \frac{{}^2k_2}{{}^1k_2} e^{(\lambda_2 - \lambda_1) \cdot t} \right], \tag{9}$$

where ${}^1k_2$, ${}^2k_2$, and λ_1, λ_2 are functions of x, n, and β. With complete mixing, Eq. (3) and (4) simply will reduce to

$$C_{cm}(t) = C(0)e^{-nt}; \tag{10}$$

$|\lambda_2| > |\lambda_1|$, indicating that after a certain time, which usually has an order of magnitude close to the nominal time constant for the room, the concentration decay has the same rate for each box. The concentrations, however, are different for each box, normally lowest for the "supply box" (depending on the thermal characteristics of the supply air, the nominal "supply box" is not always the actual supply box). The concentration difference is strongly dependent on β and so are the λ's and

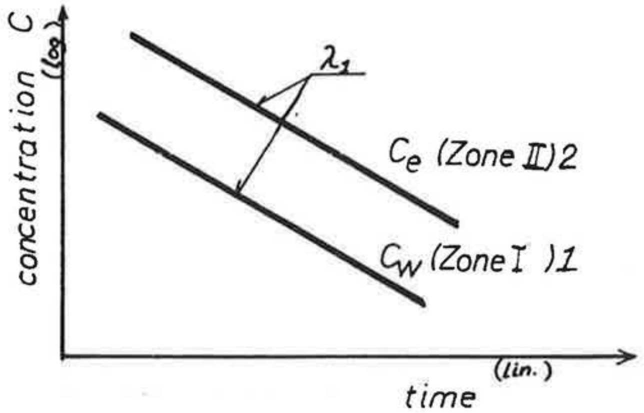

Fig. 2. Decay tracer gas in a room with two zones. Single logarithmic diagram.

the k's. In a lin-log plot (lin time, log concentration) of the concentration decay, λ_1 is the slope of decay curve, indicating how fast contaminations are diluted, Fig. 2. The dilution rate is the same for the whole room except for a constant ratio between the concentration in each box. It is obvious that the box with the lowest concentration in total has the best ventilation. Nevertheless, λ_1 is a measure for the overall performance of the ventilation system and should be compared with n, the nominal air exchange rate for the room. The ratio $-\lambda_1/n$ is adopted as a transient ventilation efficiency, ϵ_1:

$$\epsilon_I = - \lambda_1/n. \tag{11}$$

Malmström and Östrøm (1980) have shown that ϵ_1 is equal to the ratio between the concentration in the exhaust air, C_E, and the mean concentration for the room in question, C_R, during transient conditions, which is the same as the definition of ventilation efficiency used by Rydberg and Kulmar (1947).

The efficiency could also be defined as the ratio between the concentration in the exhaust air and the working zone, similar to steady state:

$$\epsilon_{IV} = \frac{C_E(t)}{C_W(t)} = 1/{}^1k_2 \left(\frac{1 - \frac{1 - {}^1k_2}{1 - {}^2k_2} e^{(\lambda_2 - \lambda_1) \cdot t}}{1 - \frac{1 - {}^1k_2}{1 - {}^2k_2} \cdot \frac{{}^2k_2}{{}^1k_2} \cdot e^{(\lambda_2 - \lambda_1) \cdot t}} \right). \tag{12}$$

After the period of stabilization, Eq. (12) can be written as

$$\epsilon_{IV} = 1/{}^1k_2. \tag{13}$$

Since this ratio does not relate to any reference condition and only is simply an expression for the concentration distribution in the room, it is not a useful tool to describe ventilation efficiency. However, ϵ_{IV} and ϵ_I are closely connected.

If the exhaust opening is moved to box 2-low (see Fig. 3), a "diagonal" ventilation scheme is produced. The same scheme is produced also if supply opening is moved to box 2-low instead of the exhaust. This process is governed by the differential Eqs. (6) and (7):

$$\left.\begin{matrix}\text{S in 1} \\ \text{E in 2}\end{matrix}\right\} \quad \begin{matrix} a_{11} = - \dfrac{1 + \beta}{x} n, \; a_{12} = \dfrac{\beta}{x} n \\ a_{21} = \dfrac{1 + \beta}{1 - x} n, \; a_{22} = - \dfrac{1 + \beta}{1 - x} n \end{matrix}$$

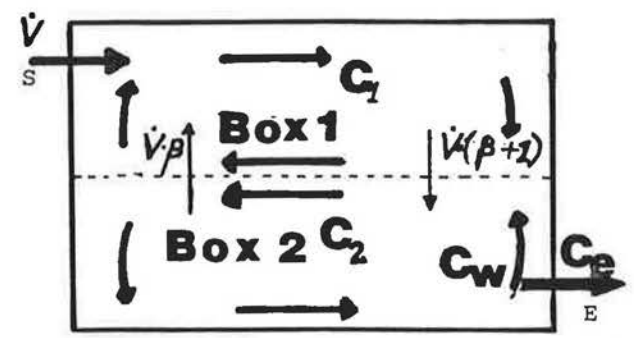

Fig. 3. Flow patterns for a two-box model, supply and exhaust in different boxes.

$$\left.\begin{array}{l}\text{S in 2}\\ \text{E in 1}\end{array}\right\} \quad a_{11} = -\frac{1+\beta}{x}n,\; a_{12} = \frac{1+\beta}{x}n$$

$$a_{21} = \frac{\beta}{1-x}n,\; a_{22} = -\frac{1+\beta}{1-x}n.$$

Figure 4 shows the calculated efficiency ϵ_I and ϵ_{II} as a function of β for different ventilation and contamination source schemes. Note that the diagonal ventilation scheme, supply and exhaust in different boxes, is the most efficient. Furthermore, complete mixing is the worst situation for the diagonal scheme, while best for the "conventional scheme." This indicates that complete mixing is not what one always should aim for, and that it should be possible to improve ventilation efficiency by simply changing to diagonal solutions. The best results are achieved if one supports the natural tendency for thermal stratification and uses diagonal bottom fed ventilation schemes. This is what we successfully have done in industrial plants in Norway during the last few years.

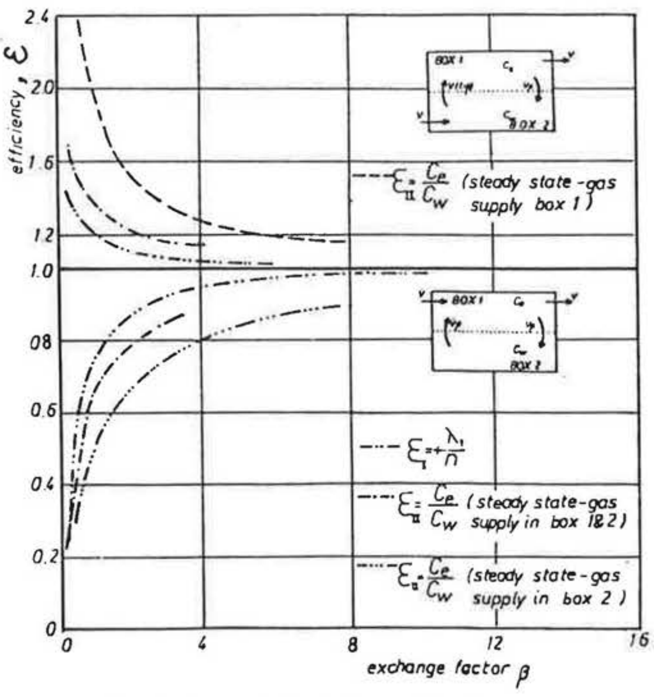

Fig. 4. Different efficiencies as a function of β.

It is, of course, important to avoid supplying air with higher temperature than the air in the working zone. Other conclusions from Fig. 4 are first that ϵ_I does not depend on the source, contrary to ϵ_{II}, and second, it is not a prerequisite to do tests with uniform initial concentrations in the room.

Laboratory Tests

Experimental technique

Tests have been carried out in a room as shown in Fig. 5. Four different locations of the air exhaust were used. The air supply was a slot 20-mm wide mounted just beneath the ceiling. To simulate a contaminated convection current, the tracer gas was supplied near a heat source of 60 W, as shown in Fig. 5. A number of probes for tracer gas measurements were placed on a column (rig) which could be placed at different positions in the room. The probes were placed in five different levels, 0.12, 0.57, 1.07, 1.60 and 2.0 m above the floor. One probe was placed in the exhaust duct. The column was placed at six different positions, of which only three are referred to in this paper. At each position a number of samplings were taken at the different levels. The sampling, analyzing of measurements and moving of the rig were done automatically, based on a microprocessor.

The studies started with a "clean" room. The tracer gas supply and the sampling were started at the same time. The gas concentration increased until the steady state was obtained. In the steady-state situation the column was moved to the five other positions. Then it returned to the starting position, and the tracer gas supply was turned off before starting the transient measurements.

N_2O was used as a tracer gas. The measurements were done with an infrared gas analyzer type URAS 1 (Hartman and Braun, Germany). Multipoint measurements with one analyzer were conducted by means of an auxiliary pumping system, as shown in Fig. 6. Test air for testing was continuously sucked through the six tubes from the measuring points. The solenoid valves were located close to the analyzer and the pump and the analyzer sucked the gas through each tube in sequence. The valve switching was controlled by the microproces-

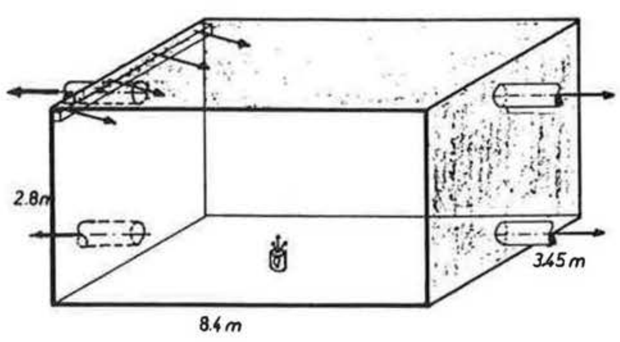

Fig. 5. The model used for the laboratory investigations.

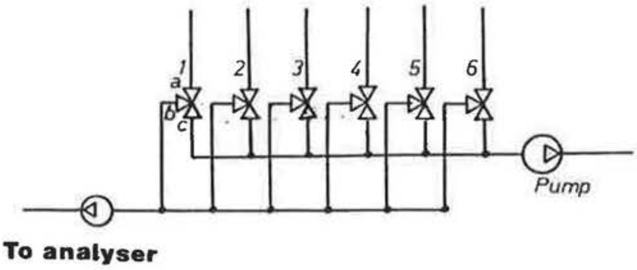

Fig. 6. Tubing and valve arrangements for auxiliary suction from measuring points.

sor. After each switching there was a certain time lag due to a short distance between the solenoid manifold and the analyzer.

Test results

Figure 7 shows a typical test record. The ventilation scheme for this record is high wall supply and exhaust ("short-circuiting scheme"). The concentrations increase gradually from zero up to steady-state level. The measured concentrations are fluctuating very much, except for the exhaust air measurements. This is due to the source characteristics which, together with the ventilation system, creates much turbulence.

The fluctuations continue for the same reason through the steady-state period. In the decay period (tracer gas supply is shut off), the concentrations decrease gradually to zero. During this period the measurements are not fluctuating. The reason for this is that the concentrations are distributed more evenly when the source is shut off. The curve-fitting regression coefficients for all measurements are satisfactory in spite of the fluctuations. It should be mentioned that the heat source is not switched off when the tracer gas supply is cut.

In Fig. 8 a lin–log plot of the concentrations is made for the decay period. Six main features in these measurements should be noted:

1. Concentration levels are different for different levels in the room.
2. The log-lin plot of the decay curves in concentration–time diagram tends to become parallel straight lines.
3. The difference in concentration between various locations for steady state and transient state are not equal.
4. There are great fluctuations in concentration levels when the concentration source is present.
5. Transient-state decay does not depend on the source, steady-state does.
6. Efficiencies are lower than unity.

These findings correspond reasonably well with the two-box model.

Measurements of Malmström and Ahlgren (1981) show that, during the growing transient period the slope

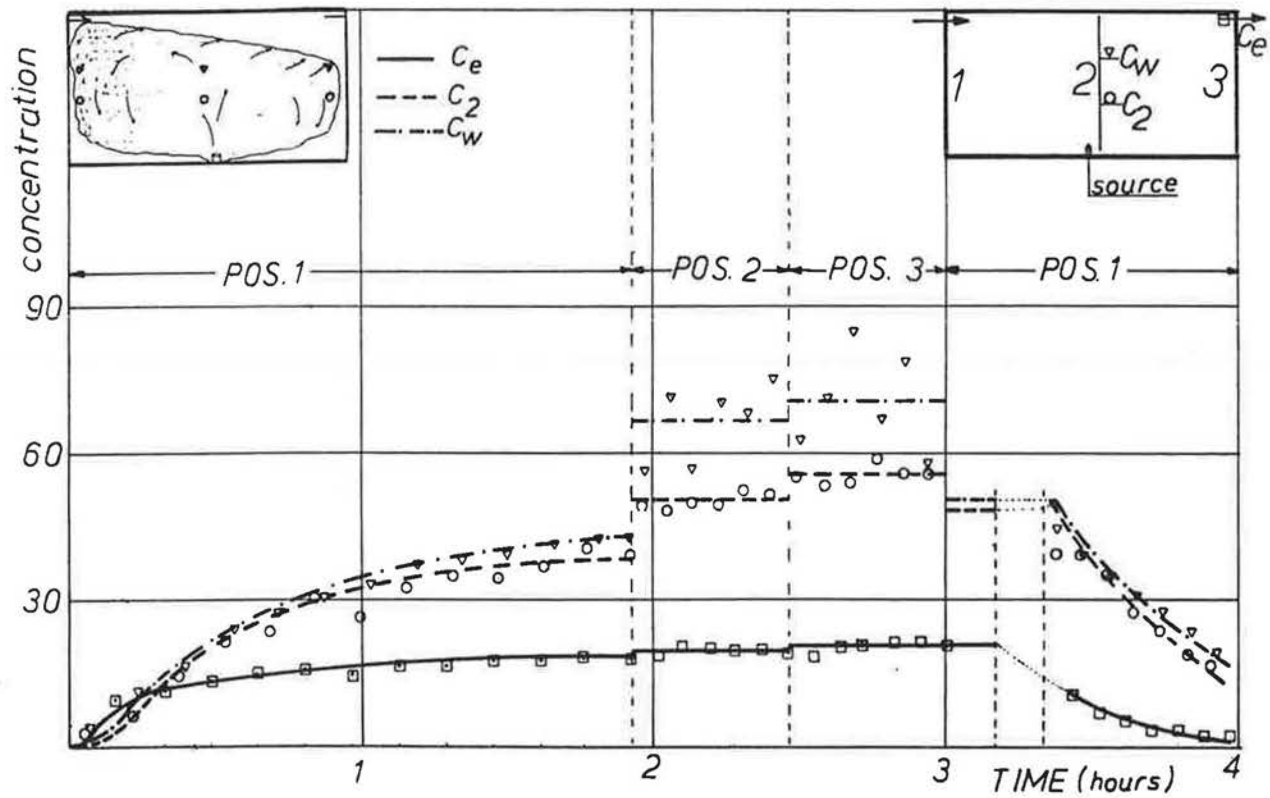

Fig. 7. Tracer gas test record, "short-circuiting" scheme. $\Delta T = +9\,°C$, $n = 6$ ach.

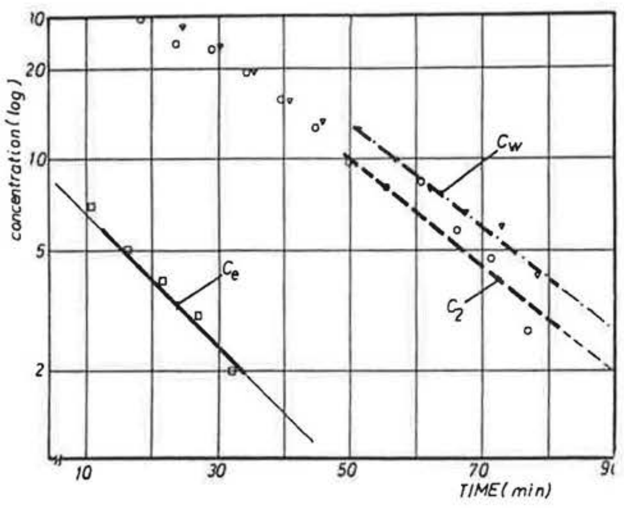

Fig. 8. Log-lin plot of decay period in Fig. 7.

of the lin-log curves, taking the difference between steady-state concentration levels and the measured concentrations for each point, are equal to the slope during the decay period. This means that ϵ_I might be calculated from the first period. We have not done that because of the great fluctuations. Our calculations of ϵ_I are based on the decay period.

A still more pronounced difference between transient and steady state is shown in Fig. 9. This is a test record from a test with a "diagonal scheme" as shown in the figure. During the transient decay period there is little difference in concentration levels between the zone of occupation and the exhaust air. The slope of the decay curve is much steeper than for the test shown in Fig. 7. The nominal air change rate is the same for those two tests. This corresponds again very well with the theoretical two-box model predictions.

Some important tests are tabulated in Table 1. Tests shown are carried out with supply air temperature above exhaust air temperature.

"Diagonal scheme"

Calculation of ϵ_I and ϵ_{II} are shown in Fig. 10. ϵ_{II} is the arithmetic mean value from measurements in positions 1, 2, and 3. This is done because of the large local variations caused by the source. We believe that the mean values give a reasonable representation of the overall conditions. The local variations can be read from Table 1.

It seems that the transient ventilation efficiency is rather independent of whether the exhaust is located on the same or on the opposite side as the supply side. The efficiencies are greater than or equal to 1, which corresponds with the predictions. For the lower air change rates the steady-state efficiencies decrease with increasing temperature difference. The reasons for this are unclear. One explanation could be that the W levels for the lower temperature differences are partly located in the "high box," a situation which gradually changes toward a more complete "low box" level for the higher temperature differences. For the higher air change rates, the W levels are all located in the "high box." The change in transient efficiency can be explained by a change in the relative size of the "box" volumes. In general the flow conditions are complicated, and convection currents on the walls play an important but still unquantified role.

It should be mentioned that all tests are run with a slot width that produces a momentum flux which generates appropriate air velocities for comfort reasons at isothermal conditions. Figures 11 and 12 show photographs of smoke tests supplying smoke at the heat and tracer gas source. The pictures are taken some time after the start of supply, showing the shape of the circulation boxes. Note the shift in the ability of the source to penetrate into the high box when the temperature difference increases.

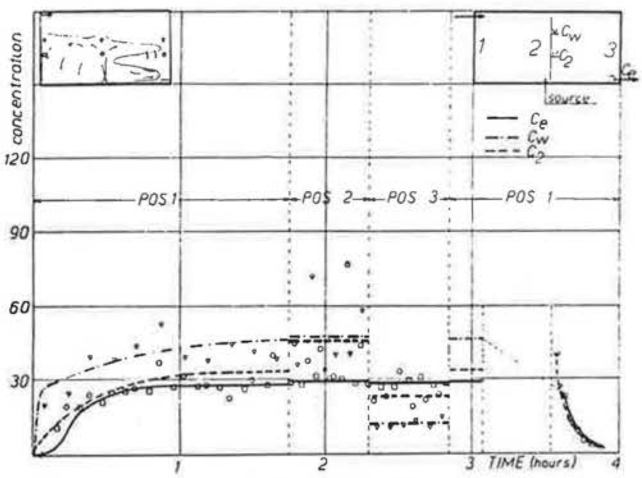

Fig. 9. Tracer gas test record, "diagonal" scheme. $\Delta T = +9\,°C$, $n = 6$ ach.

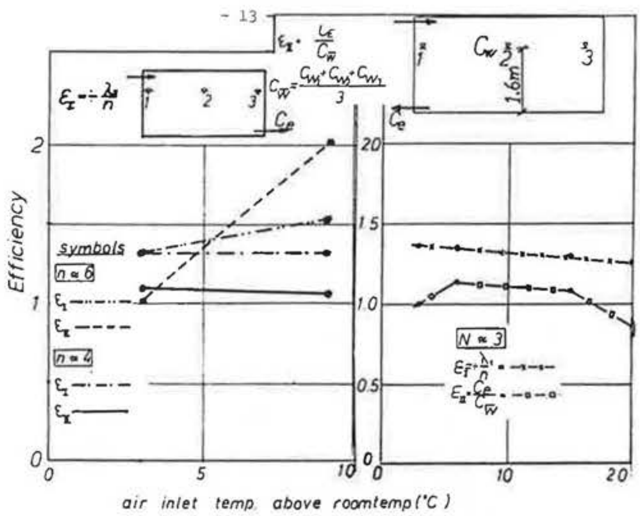

Fig. 10. Calculations of ϵ_I and ϵ_{II} from measurements as a function of ΔT. Diagonal scheme with two different locations of exhaust opening.

Table 1. Summary of test results.

Test No.	Test Conditions: Air Exchange Rate	Test Conditions: Supply Temp./ Temp. Above Exhaust Temp.	Air Inlet/Outlet Positions: b	a	c	d	Efficiency, Steady State: ϵ_{IIWI}	ϵ_{IIWII}	ϵ_{IIWIII}	ϵ_{II2I}	ϵ_{II2II}	ϵ_{II2III}	Mean Value ϵ_{II}	Efficiency, Transient $\epsilon_I = -\frac{\lambda_1}{n}$
1	6	42/10	x				0.38	0.33	0.32	0.42	0.39	0.37	0.37	0.33
2	6.81	40.6/ 9		x			0.68	1.03	3.20	0.88	0.71	1.54	1.34	1.66
3	6	29.5/ 3	x				0.91	1.06	1.16	1.07	1.26	1.36	1.14	0.53
4	6.44	23.9/ 3		x			0.86	1.11	1.08	0.90	1.06	1.12	1.02	1.16
5	3.96	29.4/ 9		x			0.29	0.63	1.66	0.51	0.64	1.22	0.83	1.29
6	4.01	36.0/ 9	x				0.26	0.28	0.30	0.25	0.28	0.30	0.29	0.37
7	4.02	23.9/ 3		x			1.00	1.09	1.15	1.02	1.02	1.02	1.05	1.31
8	4.44	24.4/ 3	x				0.83	0.95	0.97	0.81	0.94	0.95	0.91	0.47
9	4.08	38.4/15		x			0.54	1.32	2.76	0.71	0.69	1.20	1.20	1.38
10	3.3	23.2/ 3			x		0.99	1.03	0.99	0.98	1.04	1.03	1.01	1.27
11	3.0	25.9/ 6			x		0.98	1.26	1.18	1.02	1.13	1.11	1.11	1.40
13	3.18	38.7/15			x		0.80	0.95	2.38	0.44	0.83	0.86	1.03	1.28
14	3.36	44.3/20			x		0.74	0.93	1.21	0.55	0.77	0.71	0.82	1.20
15	2.94	47.3/15				x	0.31	0.29	0.29	0.33	0.33	0.31	0.31	0.45
16	2.94	- / 9				x	0.40	0.49	0.70	0.65	0.75	0.69	0.61	0.51
17	3.48	23.3/ 3				x	1.12	1.27	1.29	1.23	1.24	1.28	1.24	1.16[a]
17A[b]	3.12					x	-	-	-	-	-	-	-	0.58
12	3.12	30.6/ 9			x		0.63	1.12	1.10	0.83	0.98	1.18	0.93	1.59[a]
12A					x									1.56
18	3.18	21 /−6				x	0.98	0.91	1.01	0.97	0.94	1.00	0.97	1.04

[a]Under influence of convection from bulb.
[b]Same conditions as No. 17, but bulb switched off.

"Short-circuiting scheme"

ϵ_I and ϵ_{II} are shown in Fig. 13. There is a marked drop in efficiencies compared to the "diagonal" scheme. When exhaust and supply is located on the same side, the drop does not occur until the temperature difference exceeds 5 °C. This is because the contamination plume, due to the direction of the recirculating currents, enters more or less concentrated and stratified into the exhaust opening. This effect disappears when stratification increases, because the temperature of the plume is too low to allow the plume to enter the "high" box. Because the heat source is not switched off, during the decay period, this in fact works as though exhaust takes place from the low box, which in turn produces transient efficiency larger than 1.

Figures 14 and 15 show photos from smoke tests, which visualize the shape of the circulation boxes. In addition to confirming the applicability of the two box model, the tests using a short-circuiting scheme demonstrate how sensible the behaviour of a ventilation system can be to the mutual layout of the location of supply and exhaust openings, and the location and strength of the heat sources. Note here too the same shift in flow patterns with increasing temperature difference as for Figs. 11 and 12. If both heat and tracer gas supply are shut off, transient efficiency should drop. Figure 16 shows a plot in which this is done. As we can see, the predicted drop actually takes place.

General comments

Tests with supply air temperature lower than the room air temperature (cooling) are not shown here. A few tests are run of which one is tabulated in Table 1 (test No. 18). The results are indicating that a short-cir-

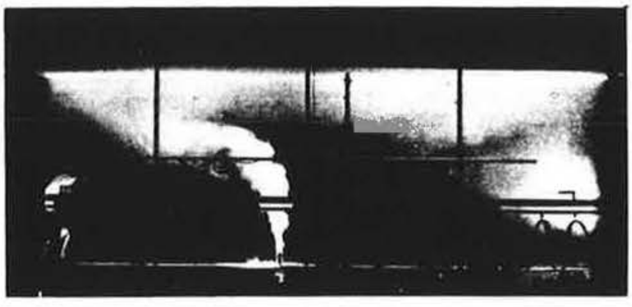

Fig. 11. "Diagonal scheme." $\Delta T = +3$ °C, $n = 3$. Plume penetrates into the "high box."

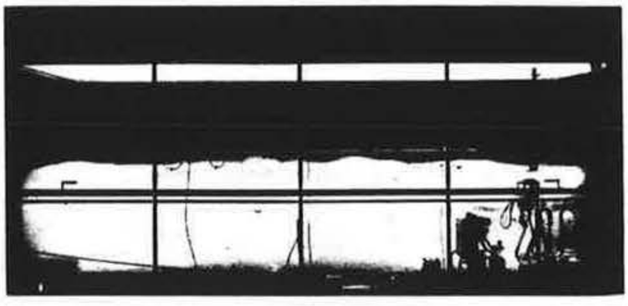

Fig. 12. "Diagonal scheme." $\Delta T = +15$ °C, $n = 3$. Plume does not penetrate into the "high box."

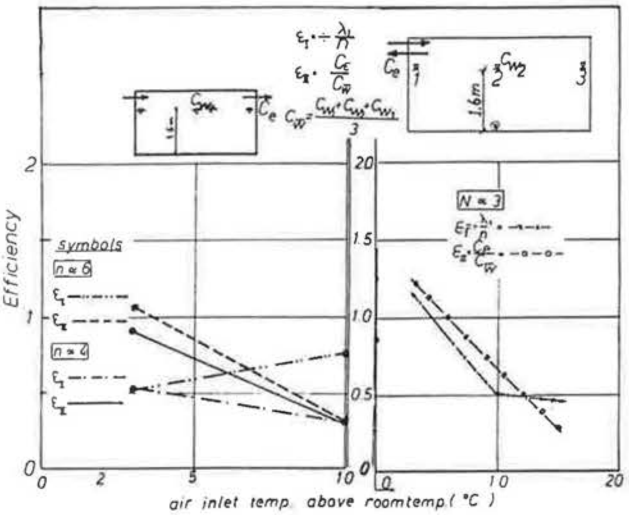

Fig. 13. Calculation of ϵ_I and ϵ_{II} from measurements as a function of ΔT. "Short-circuiting scheme" with two different locations of exhaust opening.

cuiting scheme tends to behave like a diagonal scheme, and vice versa with high wall/ceiling air supply. Moving air supply to floor level, it is obvious that supplying air with a temperature lower than the room air temperature produces similar results as with high supply of heated air. However, we expect that efficiencies for a diagonal scheme will be higher for all contaminations having a density lower than the room air. All contaminations having "normal" concentrations will become lighter than the room air when heated only some tenth of 1 °C.

Conclusions

Expressions for ventilation efficiency are derived using a two-box theoretical model. The definition of ventilation efficiency arrived at in this manner can be used for practical measurements, and seems to be valid also for multibox schemes. Different methods of measuring and defining ventilation efficiency give essential differences in the results. ϵ_I, the ratio between λ_I and n,

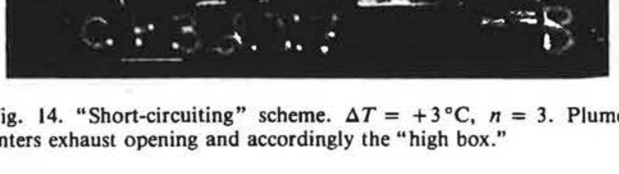

Fig. 14. "Short-circuiting" scheme. $\Delta T = +3\,°C$, $n = 3$. Plume enters exhaust opening and accordingly the "high box."

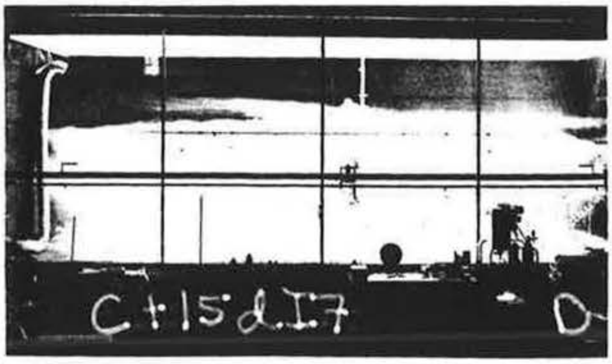

Fig. 15. "Short-circuiting" scheme. $\Delta T = +15\,°C$, $n = 3$. Plume does not penetrate into the "high box."

gives a measure of the average speed at which a ventilating system dilutes contaminations brought into a room compared to complete mixing dilution rate.

The ratio between the concentration in the exhaust air and the working zone at steady state, ϵ_{II}, gives a measure of the ability of the ventilating system to remove contaminants from the zone of occupation. The latter efficiency is the most interesting one at normal conditions. ϵ_{IV}, the ratio between the concentration in the exhaust air and the working zone at transient condi-

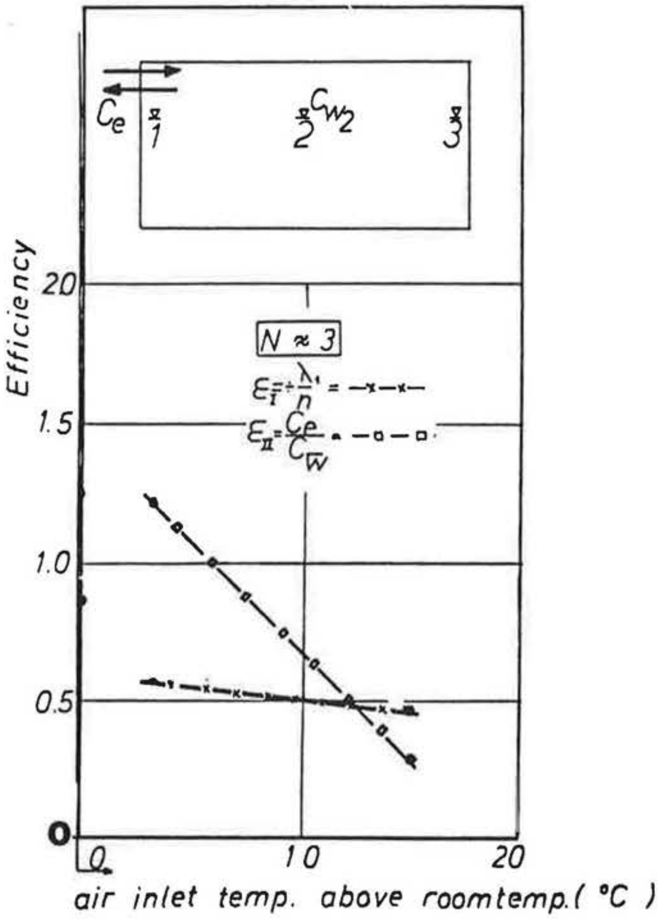

Fig. 16. Results with the "short-circuiting" scheme when the heat source is switched off.

tions, does not tell much about how well a ventilating system removes contaminations from the working zone. The transient methods do not consider the fact that the contaminations are usually supplied from single sources.

The tests show that the efficiencies vary considerably between different ventilating systems. One conclusion so far states that with the air supply located just beneath the ceiling and the air exhaust near the floor, we will obtain the best system for warm air ventilation (air heating). Then it should be obvious that a system with air supply near the floor and exhaust beneath the ceiling should be the best system for cool air ventilation (cooling). That is, "diagonal schemes" seem to be the most efficient, and more efficient than having complete mixing.

It should be mentioned that diagonal schemes with diffuse air supply (not applied for heating) in the working zone have successfully been used in industrial plants in Scandinavia in the last few years. In future research, we will explore more thoroughly "diagonal schemes" with low-level air supply, both for small and large rooms. We will also carry out field measurements to determine how efficient ventilation is in existing buildings, and of course explore the usefulness of the described methods of measurement.

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