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EXTRACT This report is a paper that was presented at the ASHRAE Annual Meeting 1983. A two-zone mixing model is used to describe the concept(s) of and to define the effectiveness of ventilation. Tests using this principle, displacement with stratification, in an office room for one to three persons (28 m^2, 2.8 m ceiling height), are reviewed. Stratification was secured by supplying the ventilation air with a temperature always lower than the air temperature in the zone of occupation. Exhaust openings were placed in the upper zone. For winter conditions, the stratification was more or less impaired by the convective air circulation created by the panel heaters. However, the system behaved as a displacement flow type system. For summer conditions, effectiveness was usually heigher than for winter conditions. The required supply-air temperature for cooling is higher when using the described principle instead of complete mixing. Ventilation effectiveness was higher for this ventilation design than complete mixing effectiveness in all tests.

3 INDEXING TERMS:

NORWEGIAN	ENGLISH
Ventilasjonseffektivitet	Ventilation Effectiveness
Lagdelte luftmasser	Air stratification
Fortrengning	Displacement

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PREFACE

This report is a paper that was presented at the ASHRAE Annual Meeting 1983. It has been published in the 1983 ASHRAE Transactions Vol. 89 2A.

The work presented was sponsored by Royal Norwegian Council for Scientific and Industrial Research (NTNF) and is a part in the project Ventilation Efficiency (Ventilasjonssystemers effektivitet).

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Ventilation Efficiency— A Guide to Efficient Ventilation

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ABSTRACT

A two-zone mixing model is used to describe the concept(s) of and to define the effectiveness of ventilation. Multiroom aspects and procedures for measuring ventilation effectiveness are dealt with.

The simple two zone model, experimentally verified by laboratory tests, predicts generally high effectiveness for ventilating systems using the displacement principle, taking advantage of stratification. Tests using this principle in an office room for one to three persons (28 m^2, 2.8 m ceiling height), are reviewed.

Stratification was secured by supplying the ventilation air with a temperature always lower than the air temperature in the zone of occupation. Necessary heating of the room was provided for by using panel heaters under the windows; the air supply was located at the opposite wall.

For winter conditions, the stratification was weak and also more or less impaired by the convective air circulation created by the panel heaters. However, the system behaved as a displacement flow type system.

For summer conditions, effectiveness was usually higher than for winter conditions but was dependent on how the surplus heat entered the room. The lowest effectiveness occurred when the only heat sources were persons and solar-heated floors. The required supply-air temperature for cooling is higher when using the described principle instead of complete mixing.

Ventilation effectiveness was higher for this ventilation design than complete mixing effectiveness in all tests.

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INTRODUCTION

The primary aims for ventilation are to remove pollution generated in an occupied space and to supply fresh (clean) air in order to maintain air quality in agreement with applied standards, which normally means to eliminate any health risk related to air quality in the zone of occupation. In addition, ventilation is also needed for eliminating fire hazards and building damage resulting from flammable gases, humidity, and corrosive gases.

Requirements are often expressed as air-exchange rates for the different rooms. Unfortunately, this is not satisfactory for assigning air quality for the zone of occupation. Since there is a tendency to decrease air-exchange rates for the purpose of saving energy and reducing the cost of heating and ventilating, we are now approaching the safety limits for air quality, and the effectiveness of the ventilating system becomes very important.

The ventilation process is a complicated flow problem. However, the concepts of ventilation efficiency, or ventilation effectiveness, can be used as a guide for designing more efficient ventilating systems with respect to both energy usage and air quality. This paper describes these concepts and uses them to discuss appropriate research work carried out in Norway and Sweden. The term ventilation effectiveness is used in this paper rather than ventilation efficiency. It is thought that this is more meaningful and is in full agreement with the notation used in Scandinavia.

VENTILATION EFFECTIVENESS

The equation for conservation of mass for a single room can be written as follows:

$$V_R \frac{d\overline{C}_R}{dt} = \dot{Q} - \left[\sum_{i=1}^{n} (\overline{C}_E - C_S)\, \dot{V}_i + \sum_{j=1}^{m} \dot{E}_j \right] \qquad (1)$$

where

$\overline{C}_R$ = Spatial mean concentration (mass/unit volume)
$\overline{C}_E$ = Concentration of contamination in the exhausted air
C_S = Concentration of contamination in the supply air
$\dot{E}_j$ = Elimination rate by removal mechanism j
$\dot{Q}$ = Contamination production rate (mass/unit time)
$\dot{V}_i$ = Flow rate through exhaust device i (equal amount of air is to be supplied)
V_R = Total room volume
t = Time

The left-hand side of the equation represents the concentration change if contaminant generation and removal are not in balance. The conditions for the zone of occupation do not enter directly into this equation but are proportional to either $\overline{C}_R$ or $\overline{C}_E$. However, the constant of proportionality differ between systems because of different flow patterns and turbulence level. The conditions are largely influenced by the thermal conditions of the ventilating system and the room and also the occupational characteristics and the characteristics of the pollution sources. As can be seen it is not possible to control the air quality by controlling the mass balance parameters as inlet/outlet conditions and ventilation airflow rate. In other words, it is not possible to design for air quality by using the general mass balance equation.

A better approach is to develop relations between the conditions for the zone of occupation and the parameters in the mass balance equation. It is appropriate to call these relations performance parameters or efficiencies. If the elimination rate is subtracted from the production rate, one may, for the steady state situation, write the mass balance equation as follows:

$$\dot{V}_{cm} = \frac{\dot{Q} - \sum_{j=1}^{m} \dot{E}_j}{C_{cm} - C_S} = \text{complete mixing flow rate} \qquad (2)$$

where

C_{cm} = complete mixing concentration
= volumetric mean concentration for the exhausted air

Without going into details it is interesting, at this point, to state that ventilation airflow rates can be decreased by using three different approaches.

1. Reduce contaminant generation.
2. Increase local elimination.
3. Increase concentration differences between supply and exhaust.

The first two approaches are obvious and the practical implications are also obvious. The third one seems to oppose air quality. The concept of ventilation efficiency will show that this is not the case. If the concentration in the zone of occupation, C_0, is measured an apparent airflow rate for the room can be calculated.

$$\dot{V}_a = \frac{\dot{Q} - \sum_{j=1}^{m} \dot{E}_j}{C_0 - C_S} \tag{3}$$

$\dot{V}_a$ = Apparent ventilation air flow rate (m^3/s)

Now, the ventilation effectiveness can be defined as the ratio between $\dot{V}_a$ and $\dot{V}_{cm}$.

$$\epsilon_{st} = \frac{\dot{V}_a}{\dot{V}_{cm}} = \frac{C_{cm} - C_S}{C_0 - C_S} = 1 + \frac{C_{cm} - C_0}{C_0 - C_S} \tag{4}$$

C_{cm} is equal to the mean exhaust air concentration, $\overline{C}_E$, and is not a direct measure of the air quality. Through the ventilation effectiveness the concentration in the zone of occupation (i.e. the air quality) can be determined. Designing systems having $C_{cm} > C_0$ is efficient. This means that increased concentration difference between supply and exhaust does not necessarily impair air quality. It also means that aiming at complete mixing is not the best guide to efficient ventilation. These conclusions are arrived at in a rather simple and obvious way without having any theoretical flow model for the room.

To expand this concept a two-zone single-room mixing model is introduced (figure 1). This model is chosen not only because it is simple but also because practical experience has indicated that ventilated rooms tend to behave like this. The two zones are the supply zone and the rest of the room. For simplicity, the supply-air concentration is taken to be zero and the net production rate, Q_n, is introduced.

$$\dot{Q}_n = (\dot{Q} - \sum_{j=1}^{m} \dot{E}_j)$$

$$\frac{d\overline{C}_1}{dt} = \frac{\phi \dot{Q}_n}{k\,V} - \frac{y + \beta_{12}}{k}\, n\overline{C}_1 + \frac{(y - x) + \beta_{12}}{k}\, n\overline{C}_2 \tag{5}$$

$$\frac{d\overline{C}_2}{dt} = \frac{(1 - \phi)}{(1 - k)} \frac{\dot{Q}_n}{V} + \frac{\beta_{12}}{1 - k}\, nC_1 - \frac{1 - x + \beta_{12}}{1 - k}\, n\overline{C}_2$$

where

$\overline{C}_1$ = Mean concentration in zone 1
$\overline{C}_2$ = Mean concentration in zone 2
k = Ratio of volume of zone 1 to total volume
V = Total room volume
ϕ = Relative production rate in zone 1
$\dot{Q}_n$ = Total net production rate of contaminants
β_{12} = Interzonal mixing parameter
n = Nominal air-exchange rate ($\dot{V}/V$)
x = Relative supply airflow rate for zone 1
y = Relative exhaust airflow rate for zone 1

Each zone is assumed to be well-mixed. These equations can be rewritten:

$$\frac{d\overline{C}_1}{dt} = a_{10} + a_{11}\overline{C}_1 + a_{12}\overline{C}_2 \tag{6}$$

$$\frac{d\overline{C}_2}{dt} = a_{20} + a_{21}\overline{C}_1 + a_{22}\overline{C}_2$$

$$a_{10} = \frac{\phi\dot{Q}_n}{kV}; \quad a_{11} = -\frac{y + \beta_{12}}{k} n; \quad a_{12} = \frac{y - x + \beta_{12}}{k} n$$

$$a_{20} = \frac{(1 - \phi)\dot{Q}_n}{(1 - k)V}; \quad a_{21} = \frac{\beta_{12}}{1 - k} n; \quad a_{22} = -\frac{1 - x + \beta_{12}}{1 - k} n$$

The steady-state solution is obtained by letting the time derivatives become zero. The steady-state solution is:

$$\overline{C}_1 = \frac{a_{10}a_{22} - a_{20}a_{12}}{a_{12}a_{21} - a_{11}a_{22}} \tag{7}$$

$$\overline{C}_2 = \frac{a_{20}a_{11} - a_{10}a_{21}}{a_{12}a_{21} - a_{11}a_{22}}$$

The steady-state ventilation effectiveness, letting zone 2 be the zone of occupation, is:

$$\epsilon_{st} = \frac{C_{cm}}{\overline{C}_2} = \frac{k}{n\phi} \quad \frac{a_{12}a_{21} - a_{11}a_{22}}{\frac{a_{20}}{a_{10}} a_{11} - a_{21}} = \frac{\overline{C}_E}{C_0} \tag{8}$$

$$C_{cm} = \frac{\dot{Q}_n}{V} = a_{10} \cdot \frac{k}{n\,\phi} = a_{20} \cdot \frac{1-k}{n(1-\phi)}$$

New information is obtained by solving the equations when $a_{10} = a_{20} = 0$ in Eq. 6 ,i.e., solving for the step response.

$$\overline{C}_1 = K_1 e^{\lambda_1} + K_2 e^{\lambda_2} \tag{9}$$

$$\overline{C}_2 = K_1 \cdot {}^1k_2 e^{\lambda_1} + K_2 \cdot {}^2k_2 e^{\lambda_2}$$

$$\lambda_1 = \frac{1}{2}(a_{11} + a_{22}) + \frac{1}{2}\sqrt{(a_{11} - a_{22})^2 + 4a_{12}a_{21}}$$

$$\lambda_2 = \frac{1}{2}(a_{11} + a_{22}) - \frac{1}{2}\sqrt{(a_{11} - a_{22})^2 + 4a_{12}a_{21}}$$

$${}^1k_2 = \frac{\lambda_1 - a_{11}}{a_{12}}$$

$${}^2k_2 = \frac{\lambda_2 - a_{11}}{a_{12}}$$

λ_1 and λ_2 are the eigen values. See Sinden (1978) or Rektorys (1969) for the complete solution.

a_{11} and a_{22} are negative and $|\lambda_1| << |\lambda_2|$, which means that the decay curves have the same time-constant after a sufficient time, approximately $3\cdot(1/\lambda_2)$. The characteristics of the decay shown in figure 2, are:

1. The system is controlled by the longest time-constant, $1/\lambda_1$, indicating that the decay curves become parallel using a semilog graph where the time scale is linear.

2. The slope of the decay curves during the stablized decay period, $t > 3(1/\lambda_2)$, is less than for a complete mixed decay if supply and exhaust are located in the same zone (short-circuiting). The slope is steeper if supply and exhaust are located in opposite zones (displacement).

3. The concentration is lowest in the supply zone.

The slope of the parallel curves, λ_1, omitting the sign, is an apparent air exchange rate, characterizing the whole room, not specifically the zone of occupation. The ratio $-\lambda_1/n$ is a transient ventilation effectiveness, ϵ_{tr}, and indicates, first of all, the existence of short-circuiting. The influence on air quality depends on whether the zone of occupation is in the supply zone or not. Since this effectiveness index is not dependent on the source, it is not a good air quality indicator. However, if the short-circuiting takes place outside the zone of occupation, the index is proportional to the steady-state ventilation effectiveness and further, as a time-constant indicator, it tells how fast the zone can be emptied of contaminants if the generation stops, compared to complete mixing.

The areas under the curves are a measure of the exposures. These can be used for assigning ventilation effectiveness as shown by Sandberg (1981). If the room is completely mixed at zero time, the ratio between the initial concentration and the area under the concentration curve measured at an arbitrary point is a local air exchange rate because, inverted, it gives the local time-constant for the contaminants and also the air at that point. A similar time-constant can be calculated from the following expression (Sandberg 1981, 1982): using a pulse injection.

$$\frac{1}{n^o} = \frac{\int_o^\infty C(t)t\,dt}{\int_o^\infty C(t)\,dt}$$

Comparing this local air exchange rate with the nominal rate gives all information about the dilution process at this specific location. The local transient ventilation effectiveness is, for this reason, calculated as $\epsilon^o_{tr} = n^o/n$. However, this effectiveness index is not a complete air quality indicator, because it does not depend on any type of contamination source, except one. It coincides with the steady-state ventilation effectiveness for a homogeneous source. The three effectiveness expressions, ϵ_{st}, ϵ_{tr}, and ϵ^o_{tr} describe together the ventilating system in a room very well. Theoretical calculations based on the two zone model can be used to develop a design guideline for ventilating systems. However, the model does not reflect details for concentrated sources, but the qualitative result is correct.

Scandinavian research work has verified the validity of the two zone model (Sandberg 1981, 1982; Skaaret and Mathisen 1981; Mathisen and Skaaret 1981; Malmstrom and Ahlgren 1981).

Figures 3 and 4 show calculations of expected effectiveness as a function of the mixing parameter for four different schemes of ventilation. In these figures, β_{12} expresses the interzonal mixing intensity, which indicates complete separation when $\beta_{12} = 0$ and complete mixing when $\beta_{12} = \infty$. It is readily seen that interzonal circulation (mixing) has to be at least 15 times the nominal ventilation airflow rate in order to characterize the space as completely mixed. ϕ expresses the ratio between the production rate of contaminants in the upper zone and the total production rate. One can see from the graphs that the short-circuiting types of systems require intense circulation in order to achieve proper ventilation effectiveness. The effectiveness decreases sharply when the mixing parameter falls below 4 for the ceiling type of short circuiting schemes. These types of systems are, unfortunately, widely used. The most interesting feature to see is the very good performance of a vertical-up displacement type of system. The curve denoted $\phi = 1$ indicates promising results. In practice this can be achieved if contaminant sources and ventilating system can be matched in a way that a plug flow type of contaminant flow is obtained from zone 2 to zone 1. Experience from industry indicates that this is possible when there are point sources present that emit contaminants that have a density lower than the room air. It is realistic here to achieve values for β_{12} lower than 1. Pure plug flow is difficult to achieve in practice, however. The only way to create this is to supply air through a porous floor and exhaust it through a porous ceiling and have a rather high airflow rate (displacement velocity higher than 0.25 m/s), which for a room height of 2.5 m gives 360 air changes per hour. When utilizing thermal stratification, i.e., supplying air with a temperature not higher than the temperature in the zone of occupation and with low velocity, practical experience shows that the displacement flow thus created has properties close to plug flow. As pointed out, the two zone model is verified through laboratory tests and experience in

industrial ventilation. The displacement ventilation principle is, apart from in industry, also used in concert halls, theaters, sporting halls, and office type spaces with good results.

For some room types, especially large rooms, more than two zones are developed. However, the predictions resulting from the two zone models are still valid. Short-circuiting is defined in the same way, but displacement should be defined as exhaust from the zone that is farthest away from the supply zone.

Ventilation of office rooms equipped with a vertical-up displacement system has not been investigated earlier. Research work in Norway during the last year has been devoted to this type of system.

ASPECTS OF MEASURING VENTILATION EFFECTIVENESS

Single-Room Building or Sealed Rooms.

Steady-state ventilation effectiveness can be measured by measuring the concentrations of the actual contaminants at significant locations in the zone of occupation and in all exhaust ducts, together with measuring the airflow rates. Total airflow rates are determined by carrying out a completely mixed decay preferably using a tracer gas. The complete mixed concentration is determined either by calculating an air flow-weighted average concentration for the exhaust airflows or, if in/exfiltration is a significant part of the total ventilation, at certain intervals to artificially mix the air in the premises without influencing infiltration, disturbing neither the sources nor the elimination devices and measure the new steady state concentration, which will occur after a time period of 3 nominal time constants.

Contaminant sources can also be simulated by using an appropriate tracer gas, which disperses similarly to the actual contaminants.

Using a tracer gas is an appropriate procedure for measuring transient effectiveness. Local and overall effectiveness are determined from the same tests by starting with a well-mixed room or using a pulse injection technique. Note that it is not necessary to start with a well-mixed room for determining the slope of the curves. The nominal air-exchange rate is determined from a well-mixed decay as previously stated.

Multiroom Buildings

For multiroom buildings it is necessary to differentiate between single-room effectiveness and distribution effectiveness. The communication between the rooms has to enter into the mass balance equation, which for each room takes the following form:

$$\dot{V}_i \frac{d\overline{C}_i}{dt} = \left[\dot{Q}_n + \left(\sum_{j=0}^{r} \overline{C}_j \dot{V}_{ji} - \sum_{j=1}^{r} \overline{C}_i \dot{V}_{ij}\right)\right] - \overline{C}_i \dot{V}_{io} \qquad (10)$$

where

$\dot{Q}_n$ = Net generation rate
i = room i
r = number of rooms in the building
o = outside

The number of rooms communicating with room i is normally less than r. Maldonado (1982) denotes the expression within the square brackets as the net zonal generation rate. The local ventilation airflow rate or better, the local time constant, is determined from the area under the curve method, starting from a well-mixed building. This method takes into account the fact that the air is used before entering another room. The slope of the curve method involves the same procedure. The distribution index is determined by comparing the local air-exchange rate (time-constant) with the nominal rate (time-constant) measured by using a total building well-mixed decay. Single-room effectiveness can be determined by comparing with a well-mixed decay within each room but not between rooms. The amount of outside air to each room and the flow between the rooms can be determined by solving the complete mass balance equations from the tracer gas decay (Sinden 1978). The flow of outside air to each room may be determined by using a steady-state tracer gas method keeping the concentrations in each room constant and equal and measuring the generation rate for each room (Freman, Gale, and Sandberg 1982). The decay method will be more successful having one specific tracer gas for each room and measuring the simultaneous decay for each tracer gas. The transient effectiveness can now be determined from the area under the curve method and the calculated exchange rates of outdoor air. Steady-state effectiveness are more difficult to determine. However, measuring the concentrations of actual

contaminants as described for the single room is the most accurate procedure. It is also possible to use a room-by-room tracer-gas procedure. In addition an exposure index can be determined by using a room-by-room pulse supply and measuring the response for each room. The ratios between each area and the area for the reference (supply) room is a relative exposure index (Sandberg 1981; Maldonado 1982), which can be used to account for contaminant generation in other rooms. This pulse technique can also be used to calculate local time constants by the integration procedure. Each contaminant source can be simulated simultaneously in each room using tracer-gas and having the correct relative strengths. It is, of course, necessary to differentiate between the different contaminants.

Details regarding instrumentation will be omitted here, except to point out that the choice of tracer gas is dependent on many considerations, such as molecular weight, toxicity , flammability, absorbity, instrument cost, etc. For large rooms, it seems also to be necessary to use tracer gases that can be detected at low concentrations to limit the amount of tracer gas needed.

DISPLACEMENT VENTILATION VERTICAL-UP FOR SMALL OFFICE ROOMS

A characteristic feature of this type of ventilating system is that air is supplied with a velocity complying with the normal theoretical comfort range. The turbulence intensity is kept as low as possible in order to promote displacement flow. The ventilation air is not used for heating, consequently, separate heating systems are required. All types of convective and radiant heating can be used, but radiant heating seems to be the most appropriate.

Tests have been carried out in a room approximately 78 m^3 with a floor area of 28 m^2 (figure 5). Air was supplied through an adjustable opening in the short wall opposite the window wall. Supply velocity was below 10 cm/s. Both size and geometry of the supply opening were varied as well as supply airflow rate and temperature. Air was exhausted under the ceiling at the same side of the room as the supply device. The room was heated by electric convective heaters under the windows. The windows were simulated by water-cooled panels. Initial tests revealed that in order to avoid cold draft along the floor coming down behind the convectors, it was necessary to have a window sill to deflect the cold draft into the convective air currents above the heaters. Persons in the room were simulated by heated cylinders having a surface area and heat output equivalent to a sedentary average adult person. The contaminations from the persons were simulated by using N_2O as a tracer gas. Ventilation effectiveness was measured by measuring the concentration of this tracer gas with an IR analyzer. Air temperatures and air velocities were also measured. Data were collected and processed by using a microprocessor-based data-logging system. Other heat sources simulated were lighting and solar radiation. Solar heat input was simulated by using an electric heated film, covering parts of the floor, simulating direct insolation through the windows. In addition, the windows were kept at a temperature that accounted for the absorption of solar radiation. There was no furniture in the room; the data acquisition system did not allow for that. Several tests were carried out simulating winter and summer conditions.

Winter Conditions

There was no cooling load. The transmission loss was covered by the internal heat loads and the heaters. The supply air temperature was kept slightly below the temperature in the zone of occupation. The resultant temperature increase for the air through the room was approximately 1 K, which means that the thermal stratification was not pronounced. The resultant effectiveness, measured 1.6 m above the floor level, are shown in table 1.

TABLE 1

Air exch. rate (h^{-1})	1	2	3
$\epsilon_{st} = C_{cm}/C_o$	1.15	1.3	1.9
$\epsilon_{tr} = -\lambda/n$	1.10	1.25	1.8

These tests show that there are at least two circulation zones in the room, in spite of the small temperature differences, and also that a displacement flow type exists. Note that ϵ_{tr} is not dependent on the source characteristics. It is obvious that this system performs better than conventional systems in spite of the circulation generated by the heaters, the persons, and the convection currents along the room surfaces.

Summer Conditions

The effectiveness of removing heat load is now of additional interest. The conservation equation for mass is also valid for heat, because the dispersion of heat in air is identical to the dispersion of mass when conditions are turbulent. For this reason the effectiveness expressions developed for mass transport can be used, using temperatures instead of concentrations (in reality it is (ρCpT) that replaces the concentrations, but (ρCp) may be considered constant for the temperature range in question). The calculated effectiveness are called temperature effectiveness. In reality it is ventilation effectiveness with respect to energy. The steady-state temperature effectiveness cannot be expected to be equal to the concentration effectiveness because of the different source and removal characteristics (heat is, e.g. absorbed, transmitted, or emitted by the surrounding surfaces.) Transient temperature effectiveness was not measured.

Figures 6 to 9 show some of the tests carried out. As one can see the results vary with the heat source configurations, which is logical, because they have a significant influence on the turbulence structure and intensity. Solar heat load alone causes the lowest effectiveness. The reason for this is probably the generation of strong convection currents in the middle of the room causing rather good mixing. However, increasing the airflow rate increases the effectiveness. The lowest transient ventilation effectiveness is 1.2, which clearly indicates the existence of a displacement type of flow patterns for the room with two or more zones. The steady state ventilation effectiveness ranges between 1 and 15 in the zone of occupation. The steady-state temperature effectiveness in the zone of occupation ranges between 1.15 and 1.5 indicating that cooling energy can be saved using this principle. Another feature is that the supply air temperature will be higher than for conventional systems, indicating that free cooling can be used at higher outdoor air temperatures. Finally, mechanical cooling ratings are lowered and COPs for the machinery are increased.

Discussion

The system is not optimized through these tests. A better matching of supply air location and diffuser geometry to the heat and contamination sources might improve the effectiveness. The main information from the tests is that the displacement principle described, is working in a normal office room. The comfort implications need also to be mentioned. All results show temperature differences from floor level to 1.5 m above floor level less than 2 K which indicates that comfort conditions are not violated.

CONCLUSIONS

A two-zone mathematical model for studying contamination dispersal in a room is described. From this model, concepts and definitions of ventilation effectiveness are derived. Steady-state ventilation effectiveness is defined as the ratio between the complete mixing concentration and the actual concentration in the zone of occupation. Overall transient effectiveness is defined as the ratio between the nominal time-constant and the longest actual time-constant for the room. Local ventilation effectiveness is defined as the ratio between the nominal time-constant and the local time-constant.

Using these definitions and the theoretical two-zone model, it is shown that the vertical-up displacement ventilation principle is more efficient than the complete mixing principle.

Tests carried out in a 78 m^3 test room show that the vertical-up displacement principle is practical and performs as predicted by the theoretical model. Steady-state ventilation effectiveness ranged between 1 and 15, and transient overall effectiveness between 1.2 and 3.0. Vertical temperature gradients in the zone of occupation were within the thermal comfort limits i.e., less than 2 K.

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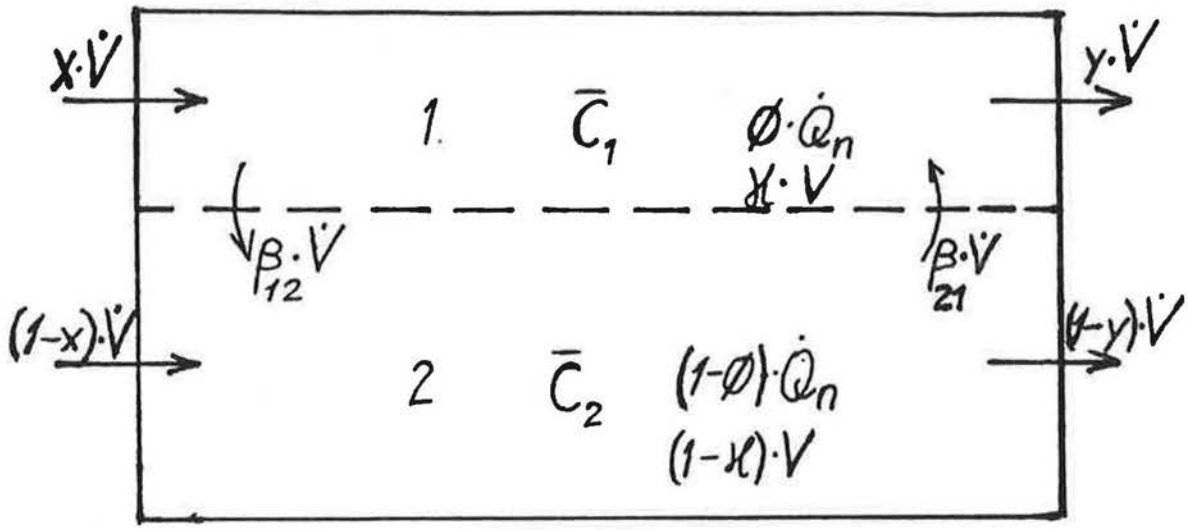

Figure 1. Two-zone mixing model with no external recirculation. Each zone has good mixing.

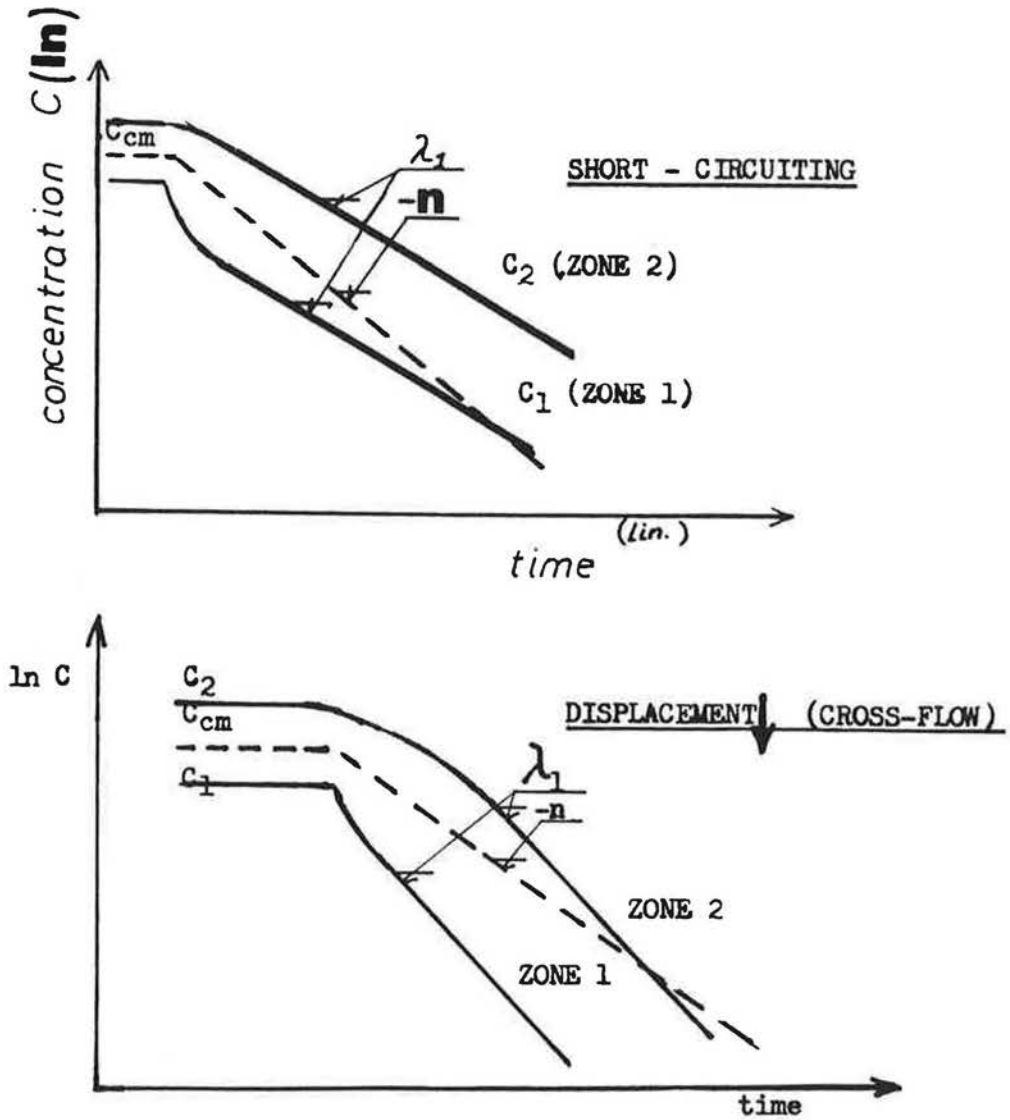

Figure 2. Decay characteristics for a two-zone mixing model.

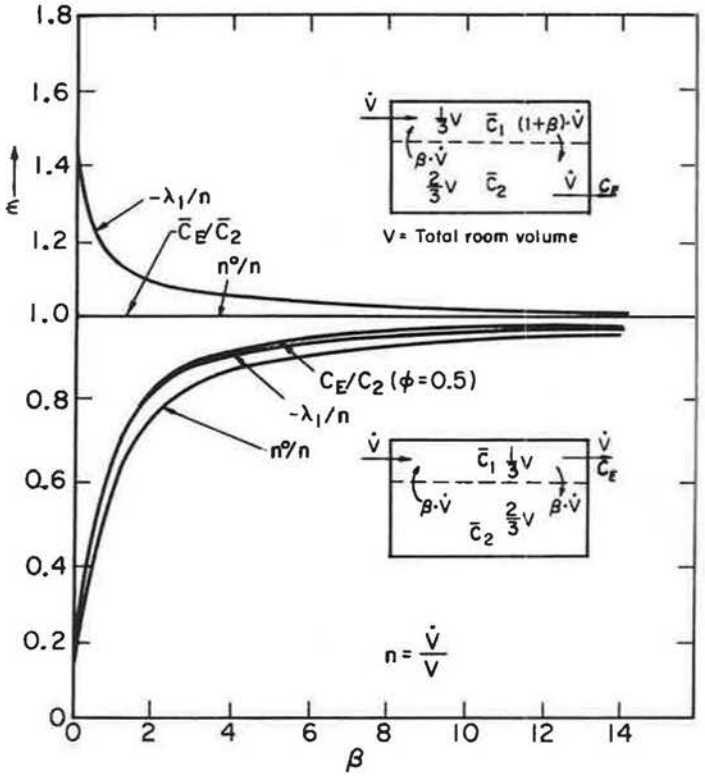

Figure 3. Calculated ventilation effectiveness by using a two-zone mixing model. Short-circuiting + displacement vertical-down.

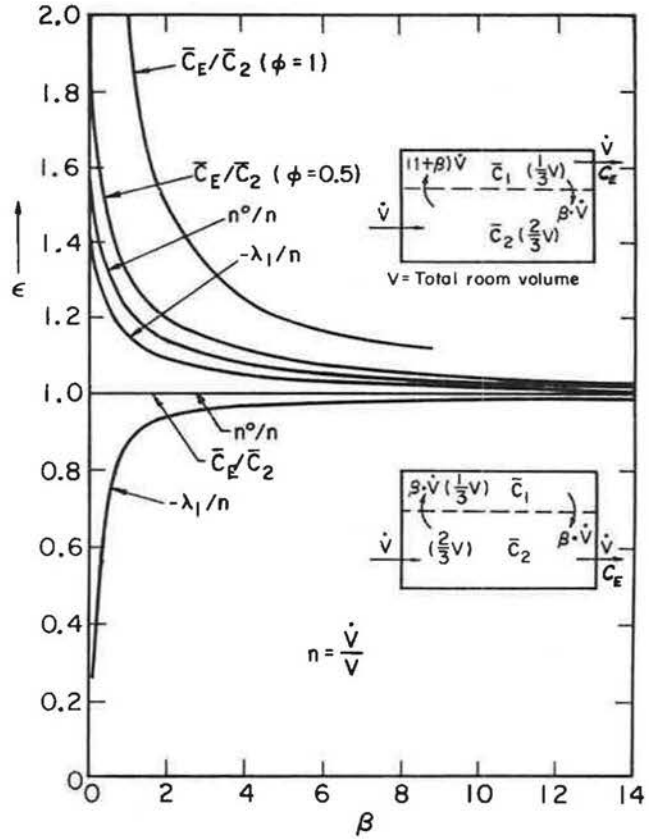

Figure 4. Calculated ventilation effectiveness by using a two-zone mixing model. Short-circuiting + displacement vertical-up.

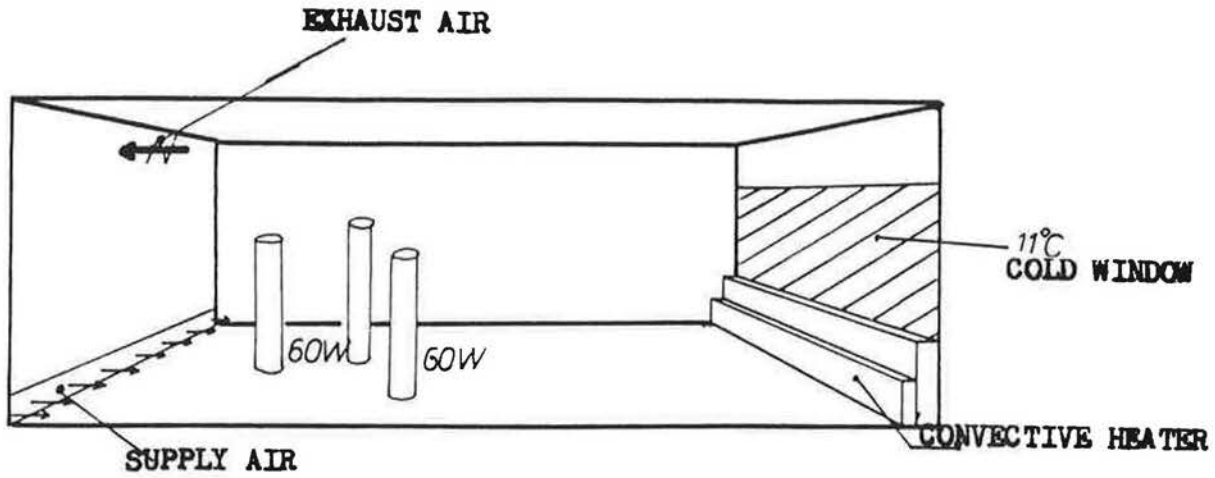

Figure 5. Test room simulating an office room, schematic.

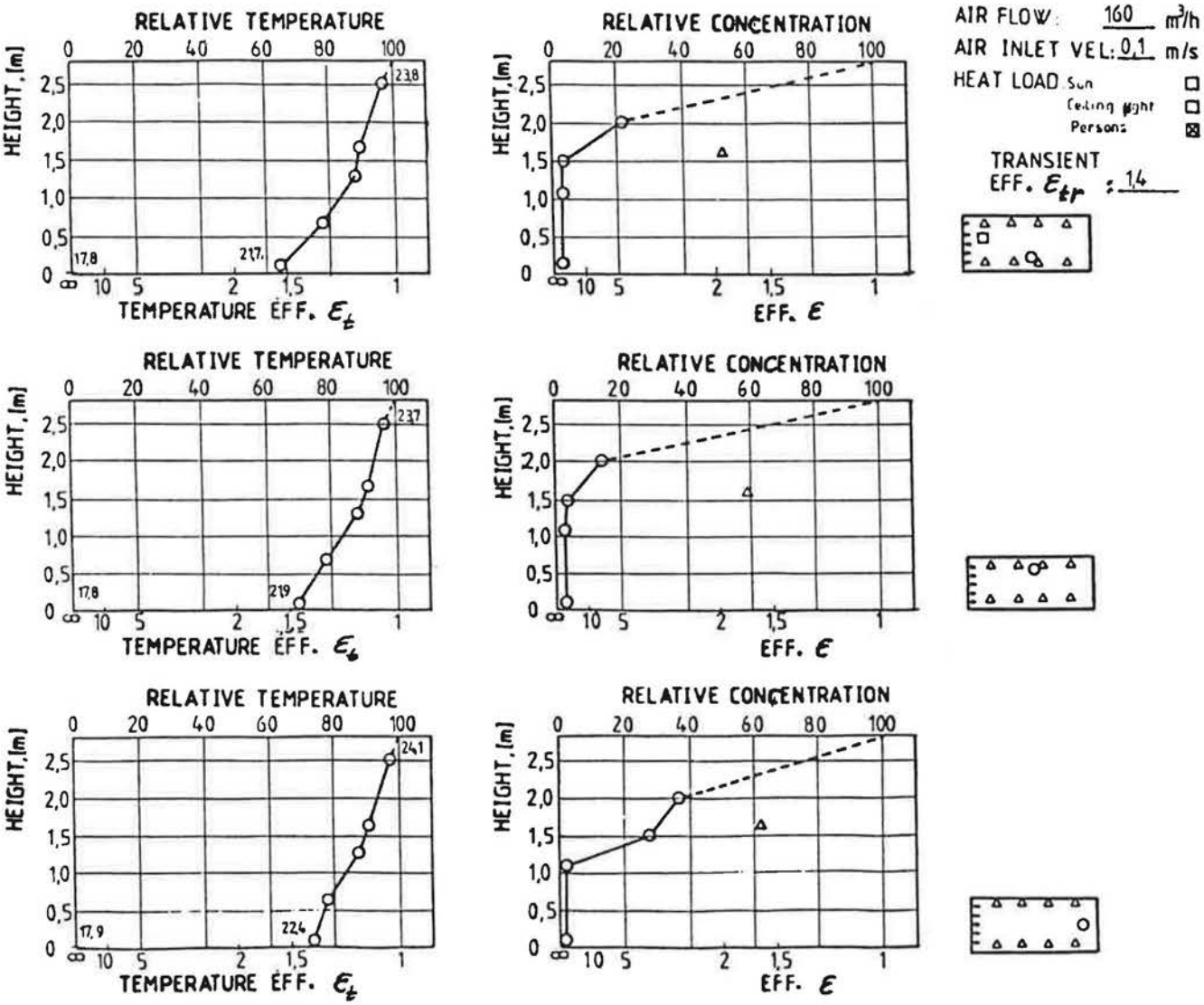

Figure 6. Measured concentration and temperature effectiveness in the test room. Upper scale shows relative concentration (temperature) profiles. Lower scale shows effectiveness profiles. Excess heat given off by persons: 2ach.

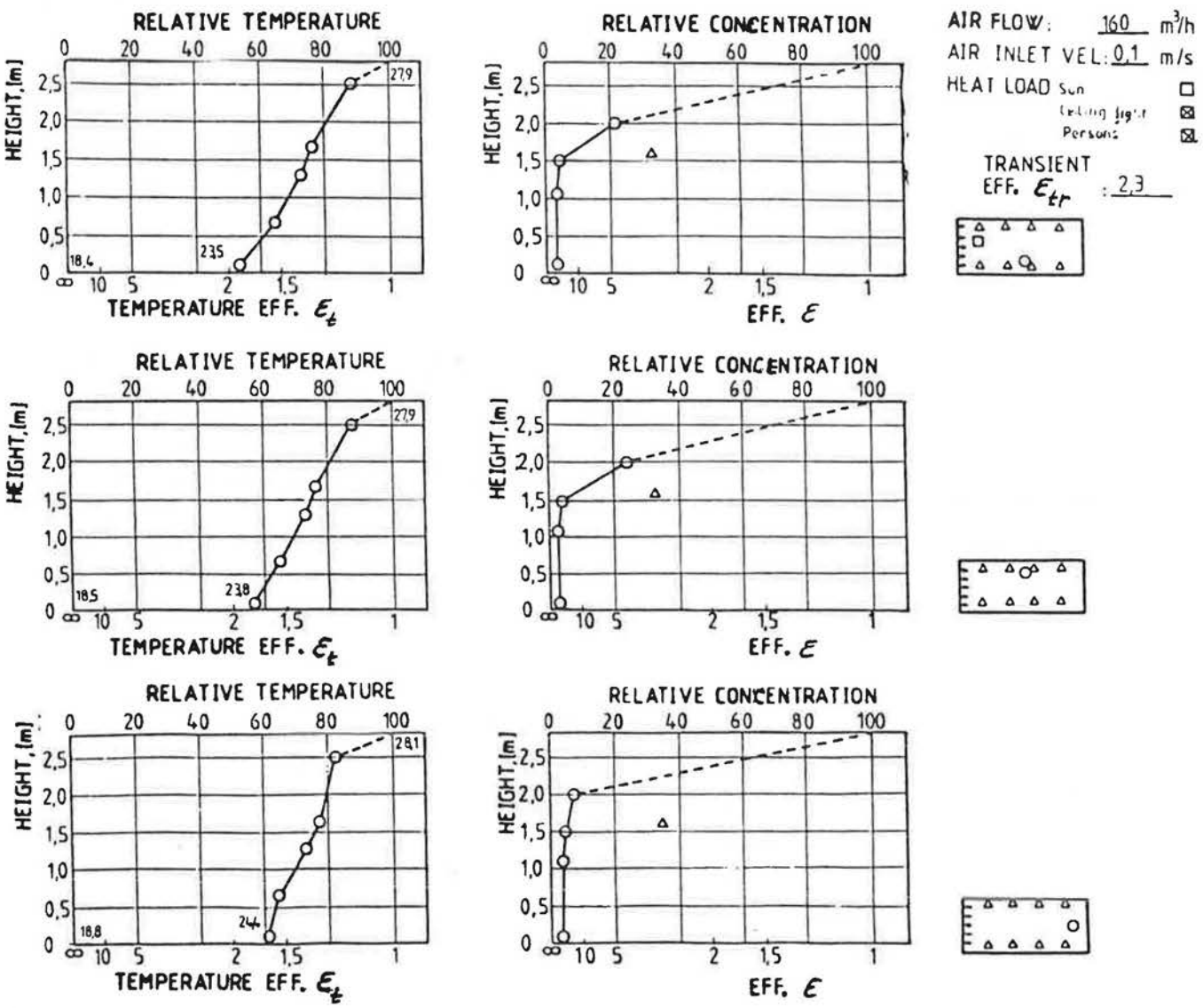

Figure 7. Measured temperature and concentration effectiveness. Scales are the same as figure 6. Excess heat given off by lighting and persons: 2 ach.

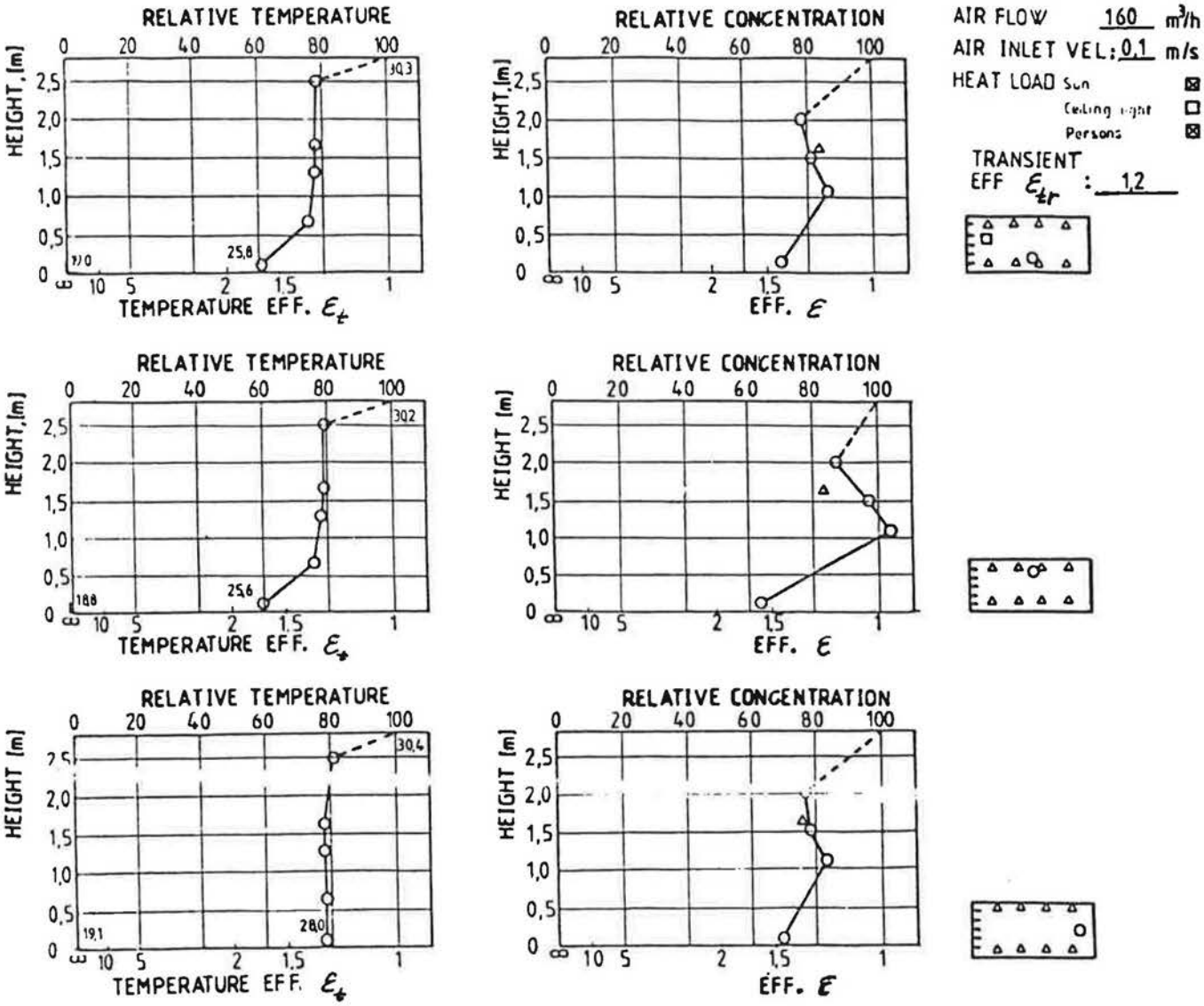

Figure 8. Measured temperature and concentration effectiveness scales are the same as figure 6. Excess heat caused by insolation and persons: 2 ach.

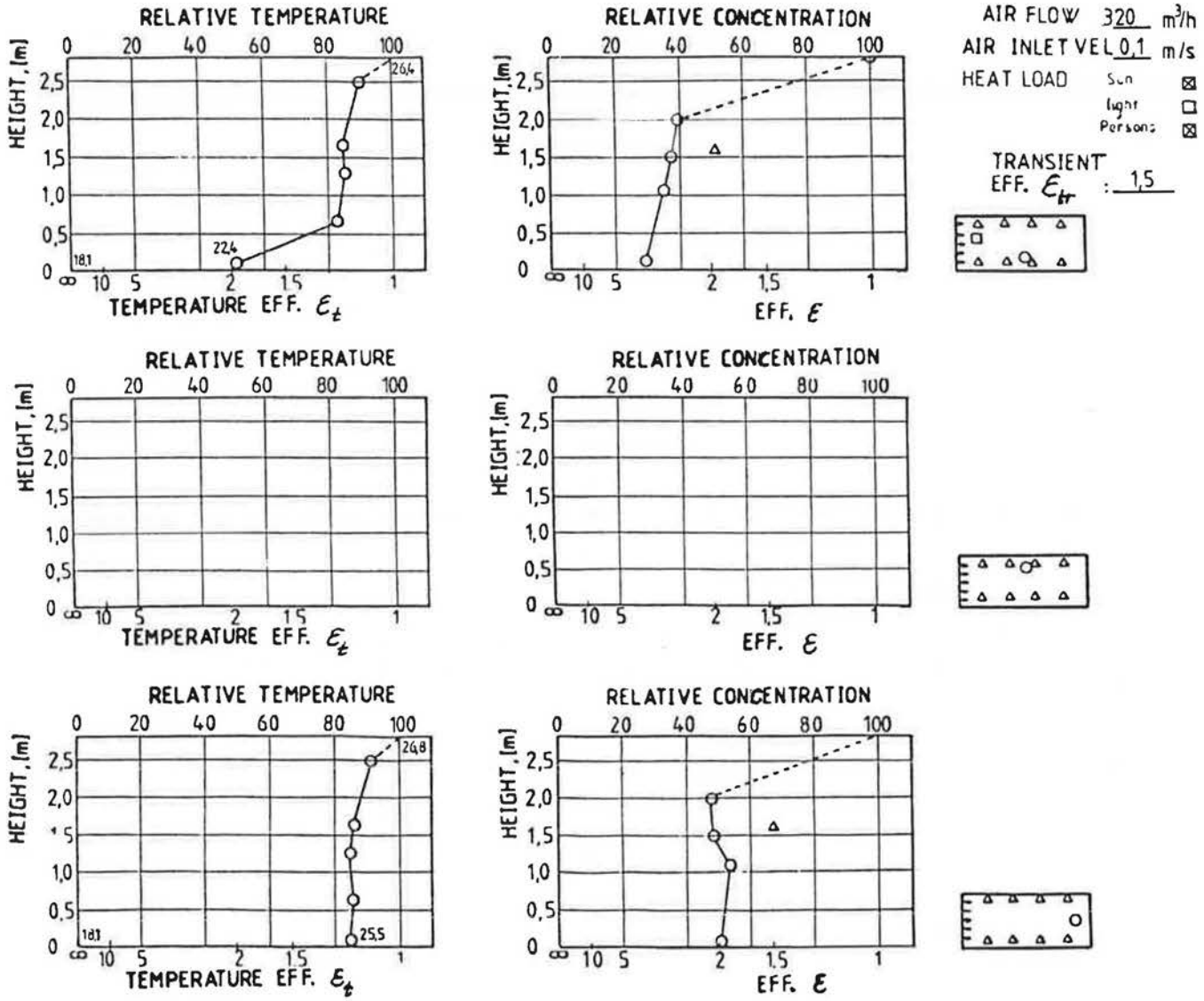

Figure 9. Measured temperand concentration effectiveness. Excess heat from solar radiation and persons: 4 ach. Scales are the same as figure 6.

DISCUSSION

J. E. JANSSEN, Honeywell, Inc., St. Paul, MN: Ventilation efficiency has an important implication with respect to the Ventilation Standard 62-1981. You have shown a correlation between efficiency and mixing factor, β . Do you think it would be possible to predict values of β for different orientations of the supply and return registers and the air velocities so that efficiency could be predicted?

E. SKARET: The value of β is dependant on many factors, including the temperature differences between supply and exhaust air, room geometry, surface temperature and occupancy. The β factor used in the paper was convenient to show the difference in effectiveness between different system lay-outs. In principle it should be possible to determine β experimentally. However, I do not think that it is appropriate to use β for design purposes, i.e., to decide on the air flow rates in detail. For design purpose an indication of the lowest expected effectiveness for different types of systems should rather be used. Much experimental work is needed to establish this information, and there is no doubt that this is possible.